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Combustion Process of the Compound Supply CNG Engine

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Abstract: Objective: In order to study the lean combustion process of a natural gas engine by separating the combustor, a spark ignition natural gas engine with separated combustors was retrofitted from a S195 single-cylinder diesel engine. **Methods:** The electronic control system controlled the gas supply and the spark plug ignition. A low pressure injection valve was set in the inlet pipe to form a lean mixture while a high pressure injection valve was placed in the subsidiary chamber to create a rich mixture, which was then ignited and injected into the main combustor, where the lean mixture was subsequently ignited again to achieve stratified combustion. **Results:** The test results showed that steady ignition is feasible in the system and verified the impact of the shape of the main combustor on HC, the impact of channel diameter on NO_X production, and the impact of the ratios of high-pressure gas and low-pressure gas on HC and NO_X. The combustion conditions of high-pressure gas and low-pressure gas in the engine combustor vary greatly. Our results signify that the shape of the main combustor has a great impact on the performance of the engine, that is, a shorter propagation distance can reduce the generation of HC. **Conclusion**: The best ignition advance angle under different conditions was determined using a spark ignition natural gas engine. The ratios of high-pressure gas and low-pressure gas greatly impact the performance and emission of the engine. The reduced diameter of the channels between the main and subsidiary combustors can enhance the stratification and facilitate the secondary ignition.

Keywords: stratified combustion; natural gas engine; compound gas supply

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1. Introduction

Natural gas (NG) qualifies as an alternative fuel under the European Union Directive (2014/94/UE) because it is a substitute for energy sources derived from crude oil [1,2].

It is used in a variety of forms to power internal combustion engines, including natural gas (NG), compressed natural gas (CNG), and liquefied natural gas (LNG) [3]. In addition to methane (CH4) (typically 90%), NG may contain trace quantities of ethane, propane, butane, and other organic and mineral compounds [4]. Due to the comparatively high quantity of CH4 in natural gas (e.g., 98% in Poland), designs with methane supply should also be evaluated [4].

The current number of NG vehicles is 24,452,517. China has the greatest number of NG vehicles, 5 million, followed by Iran, with 4 million. China has the greatest number of NG vehicles and refueling stations globally. Typically, compressed natural gas (CNG) under 20 MPa or liquefied natural gas (LNG) at an ultra-low temperature (−162 °C) and atmospheric pressure can be stored in vehicles [5]. NG has a limited flammability range and a high auto-ignition temperature of 813 K (540 °C). In addition, CNG is lighter than air under the same conditions and will quickly rise and disperse, reducing the likelihood of a conflagration. Pure NG engines or vehicles, including CNG and LNG; gasoline/NG bi-fuel mode retrofitted from spark ignition (SI) gasoline engines; and diesel/NG dual fuel (DF) mode based on compression ignition (CI) diesel engines, are the three basic application modes for NG engines or vehicles [5,6]. NG has a high ignition temperature and is difficult to ignite via compression. Initially, CNG gas cylinders are added to gasoline vehicles to convert them to gasoline/CNG bi-fuel mode. The bi-fuel vehicle has two fuel supply systems, but only one can be used simultaneously. Natural gas has a higher octane rating than gasoline, resulting in superior antiknock performance. Bi-fuel vehicles cannot enhance thermal efficiency by increasing the compression ratio (CR) when fueled with natural gas to satisfy the antiknock requirements for gasoline. In other terms, NG's benefits cannot be demonstrated in bi-fuel mode. Consequently, pure NG vehicles are designed and manufactured with higher CRs (1113) than petroleum vehicles (9.511), resulting in enhanced thermal efficiency [7]. Theoretically, the CRs of NG vehicles can increase when fueled with pure methane. However, in practice, they are typically not higher due to the complex composition and regional variation of NG. The dual fuel mode permits the use of NG in diesel-powered vehicles. Dual-fuel vehicles have two fuel supply systems that must operate simultaneously in order to consume natural gas with diesel that has been compressed and ignited. Consequently, the CRs of dual fuel motors are typically greater [8,9].

Alternative fuels are any non-conventional materials or substances that can be used as fuels. Compared to conventional fuels, these alternative fuels emit fewer air pollutants and are extremely cost-effective [1]. Natural gas is considered one of the most promising alternative fuels, and its primary component is methane (CH_4) [10]. Natural gas is compressed to less than 1% of the volume it occupies at standard atmospheric pressure to produce compressed natural gas (CNG). CNG has a high octane rating (RON = 110–130), and is, therefore, compatible with spark-ignition (SI) internal combustion engines (ICEs). Due to the high RON of CNG, engines can be operated with a greater compression ratio to improve thermal efficiency [11]. In addition, because CNG has a low carbon-to-hydrogen (C/H) ratio, it releases less CO_2 per unit of energy. CNG appears, therefore, to be an outstanding fuel for SI engines [12].

As natural gas is cleaner and abundant in supply, developing natural gas engines contributes to solving the traditional energy crisis and lowering pollution. Lean combustion can lower the emission temperature, make the fuel more cost-effective, and reduce harmful gas emissions. However, due to the high air/fuel ratio and low mixture concentration, it is vital to improve ignition reliability and flame propagation speed for smooth lean combustion. Separated (vortex chamber) combustors ensure two-stage ignition. After part of the gas in the subsidiary combustor is ignited, high-temperature and high-pressure gas is injected into the main combustor at high speed through the jet hole in the form of a jet flame with a high temperature, which quickly ignites the lean mixture in the main combustor and forms a strong turbulent flow in it, significantly improving the combustion speed. This study aimed to investigate the lean combustion process of natural gas engines by separating the combustor. A spark-ignition natural gas engine with separated combustors was retrofitted from a S195 single cylinder diesel engine. To the best of our knowledge, this is the first study that reports a retrofitted spark-ignition natural gas engine with separated combustors using a S195 single-cylinder diesel engine.

2. Structural Design of a Natural Gas Engine with Compound Gas Supply

A new combustion system was developed for the spark ignition of natural gas directly injected into a cylinder. The combustion system had separated combustors and a compound gas supply system. The subsidiary chamber featured a vortex chamber type and was placed on one side of the cylinder. The top pit of the piston of the main combustor was near the vortex chamber. A cylindrical connection channel was adopted between the main and subsidiary combustors. The prototype of the test engine was a 195 single-cylinder diesel engine with a diameter of 95 mm; a travel distance of 105 mm; and a compression ratio of 20, which was later reduced to 10. The cylinder diameter and travel distance remained the same as those of the prototype. Figure 1 shows the combustion system of the spark-ignition natural gas engine.

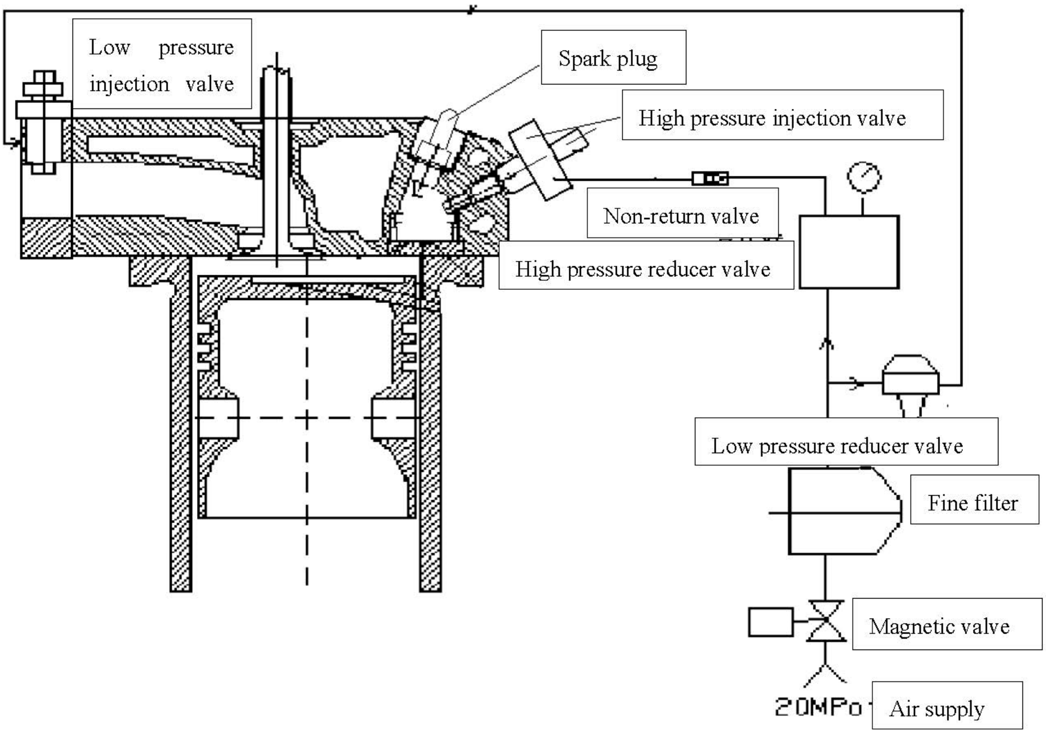

Figure 1. New combustion system of a spark-ignition natural gas engine.

2.1. Piston Design

According to the required compression ratio, the volume of the main combustor on the engine piston was calculated as 46 mL. In order to study the impact of the main and subsidiary combustors on the stratification, ignition, and combustion of the mixture, three kinds of pistons were designed, respectively. As shown in Figure 2, the depth of the combustors of the Type 1, Type 2, and Type 3 pistons decreased successively, while their diameters increased successively. Therefore, the propagation distance of the flame changed successively after being injected from the subsidiary chamber.

Figure 2. Shapes of the three main combustors.

2.2. Inserts Design

The inserts between the main and subsidiary combustors are decisive in the combustion process. The diameter of the insert channel can affect the intensity of the eddy and turbulent flow in the subsidiary combustor and the air inlet speed. An increased channel diameter leads to decreased flow resistance, increasing the air entering the subsidiary combustor. In addition, the enhanced intensity of the eddy current leads to rapid diffusion of fuel, weakening the characteristics of two-stage ignition. The tendency of single-stage combustion increased the maximum combustion pressure. When the diameter of the channel was decreased, more obvious concentration stratification resulted in a higher concentration of the mixture in the subsidiary combustor and a lower concentration of the mixture in the main combustor. The mixtures in the main and subsidiary combustors remained not conducive to the generation of NOX. Accordingly, inserts with 8 mm and 14 mm channel diameters were analyzed, respectively, to determine the best channel diameter. The shape of the insert determines the shape of the subsidiary combustor. The inserts were divided into plane and ball types, as shown in Figure 3. The subsidiary combustor was shaped like a bell when the plane type insert was used. When the ball-type insert was used, it was shaped like a ball. Such two kinds of subsidiary combustors impact the overall combustion process differently. The ball type has a strong eddy current and a fast combustion speed, while the bell type has a relatively larger volume and can hold more of the mixture.

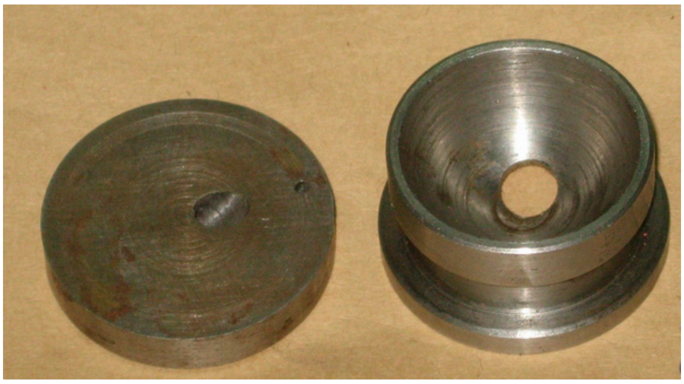

Figure 3. Two inserts with different shapes.

2.3. Installation of Injection Valve and Spark Plug

As shown in Figure 4, the high-pressure injection valve was installed at the diesel nozzle of the original machine, and the spark plug was installed at its electric plug. Due to the serious latent heat effect of natural gas, when injected directly into the combustor, the temperature in the combustor will dramatically decrease, which will become even more obvious as the engine load and the required fuel increase. Therefore, a connector for the high-pressure injection valve was designed. The natural gas was first injected into the connector after being ejected from the high-pressure injection valve. The exterior of the connector was linked with the machine and the combustor to keep it at a relatively high temperature. Natural gas can preheat through the connector, weakening the latent heat effect, which was strongly proven by the test.

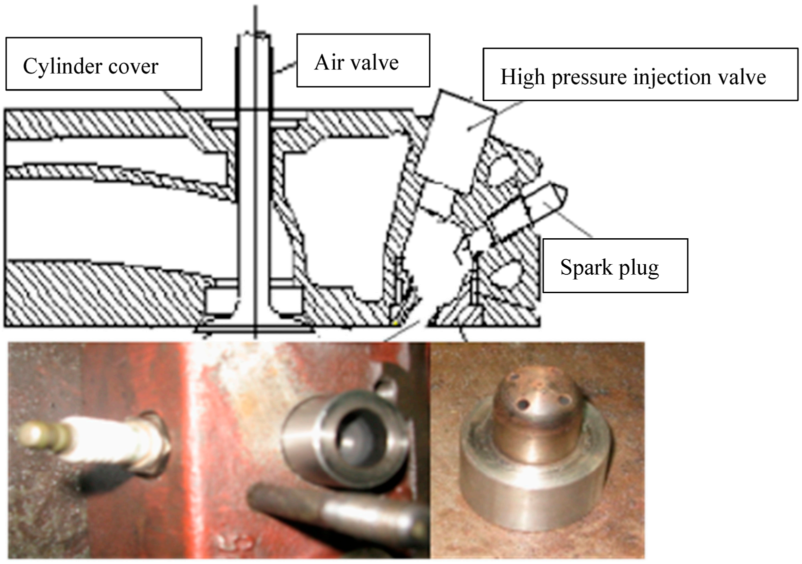

Figure 4. Installation plan.

2.4. Natural Gas Supply System Composition

In order to realize the design idea of concentration stratification spark ignition in a natural gas cylinder, considering the gas supply mode of natural gas, concentration stratification in the combustion chamber should be realized depending on the gas supply position and timing. Thus, a natural gas compound gas supply system was designed. The system's composition is shown in Figure 5.

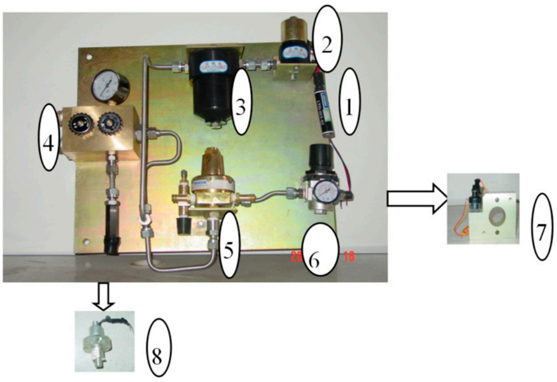

Figure 5. Natural gas supply system composition. (**1.** Overcurrent protector. **2.** Stop valve. **3.** Filter **4**. High pressure decompressor. **5.** Primary low-pressure decompressor. **6.** Secondary low-pressure decompressor. **7.** Low-pressure injection valve. **8.** High-pressure injection valve).

In the above figure, it is shown that the decompression process was mainly controlled by machinery, while the jet process was controlled by a computer. The engine parameters were collected by the acquisition card, and the control parameters were calculated by the CPU and output to the actuator to control the operation of the engine. The clock frequency ensures the real-time accuracy of the control.

2.5. Injection Valve Drive Module Design

The function of the driver is to input the drive signal to the injection valve, and then to control the injection valve's action. In the design, based on the principle of integration, some integrated chips are selected as far as possible, and a current-type drive circuit is designed according to the characteristics of the injection valve. Figure 6 shows the schematic diagram of the drive circuit, and its characteristics are as follows:

(1) Voltage and current range
Voltage: 9–30 volts
Current: 1 amp

(2) Input signal
Optocoupler drive voltage: 5 volts
Control signal: TTL voltage, pulse width modulation signal

(3) Table 1 describes the drive ports.

Table 1. Injection valve drive circuit port description.

Port Number	Port Name	Color	Description
1	POWER	red	Positive electrode of power supply
2	GND	black	Negative electrode of power supply
3	VCC	orange	5 V
4	IN	blue	TTL voltage, PWM signal
5	OUT	yellow	Injection valve drive signal
6	DRVGND	white	Injection valve drive power supply circuit

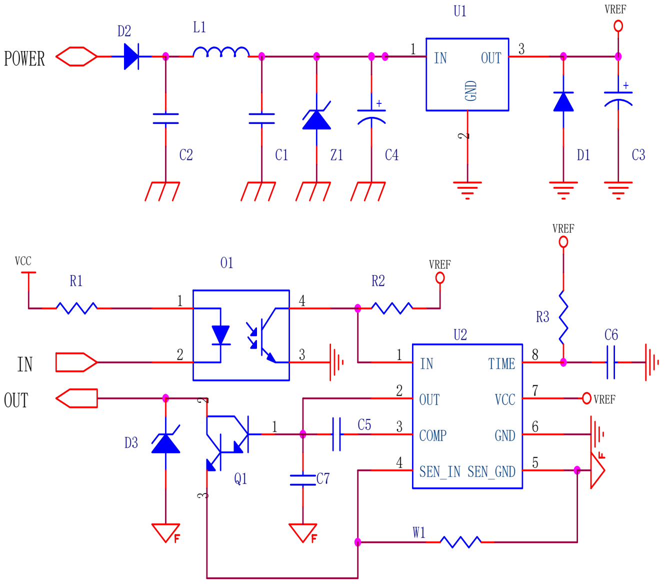

Figure 6. Driving circuit schematic diagram.

3. Design of Electric Control System of Natural Gas Engine with Compound Gas Supply

3.1. Design of Electronic Control System for Gas Supply

The type selection and modification design and processing of the pressure regulator for the high-pressure gas supply system; the design and processing of the high-pressure injection valve and mounting seat; and the design and processing of the low-pressure injection valve and seat were developed. The assembly and air tightness tests were carried out for the whole system. The characteristics of each component of the whole system were as follows. Stop valve: quickly cut off the supply of natural gas; high-pressure pressure regulator: the gas from 20 MPa pressure relief to 10–15 MPa, its range was adjustable based on the original pressure regulator redesign; and primary pressure reducer: the high-pressure natural gas was reduced from 20 MPa to a certain pressure level. Due to the endothermic cooling effect of natural gas in the process of decompression, the pressure reducer heat exchanger device was configured to ensure the normal operation of the pressure reducer. Two-stage pressure reducer: reduced the natural gas pressure to about 0.4 MPa, and the pressure range was adjustable; high-pressure injection valve: injection of high-pressure natural gas into the engine; low-pressure injection valve: low-pressure gas is injected into the engine intake pipe.

The function of the gas supply system is mainly to reduce the high-pressure gas (20 MPa) supplied by the gas cylinder to a relatively stable pressure, so that it can be used by the natural gas engine and injected into the cylinder or the intake pipe through the injection valve. Spray penetration was discovered to be substantially affected by injection pressure in the CNG engine system. Spray penetration increased as the injection pressure increased. Immediately after the injection began, the discharge angle was influenced by the injection pressure. Under a higher injection pressure, the initial discharge angle was smaller, but increased more rapidly. The current findings achieved the design idea of concentration stratification spark ignition in a natural gas cylinder, considering the gas supply mode of natural gas. Concentration stratification in the combustion chamber should be realized depending on the gas supply position and timing. These adjustments can overcome the previous concerns of the CNG engine design [13].

3.1.1. Electronic Control Injection System Hardware

The electronic control system adopts standard bus PC to make use of its rich hardware and software resources, and also facilitates the continuous upgrade and development of the system's hardware and software. In order to realize the continuous acquisition of multi-parameter and multi-cycle input, a high-speed A/D acquisition card with a high-cost performance was selected. The I/O control card was used to achieve gas injection timing control and injection quantity control.

The acquisition card model AC6111 was selected, and its features are as follows:

- 16 single-end input, 12 bits A/D;
- Speed: 400 KHZ, input voltage: 5/10/±5/±10 V;
- Working mode: timer trigger, external clock trigger, external hardware trigger;
- 4 K word FIFO, support continuous acquisition, automatic channel scanning;
- Two-channel 12-bit D/A, output 10/±10 volts;
- Switching capacity: 8 in/8 out (output pull-down current: 100 MA).

The I/O card model PCI8554 was selected and has the following features:

- 32-bit PCI bus, plug and play;
- Board with four 82C54 timer/counter chips;
- 10 independent 16-bit decrement counters;
- Cascade counter based on 8 MHz system clock.

Each counter can be programmed to select four clock sources.

- Provides anti-bounce filtering for external clock input signals and external interrupt signals;
- Anti-bounce clock frequency can be programmed;
- Universal 8-way TTL digital input and 8-way TTL digital output;
- Dual interrupt system;
- Supports 5 V and 12 V power supply protection;
- Compact half-length card structure;
- 100-pin SCSI-II connector.

From the point of view of their characteristics, they fully meet the control requirements: there are sufficient channels, acquisition speed, and counters.

3.1.2. Parameter Acquisition

Sampling Frequency

The speed of the engine is set as n, the number of strokes as τ, and the number of acquisition channels as i. In order to ensure the accuracy of the MAP, cylinder pressure is collected twice for one crankshaft angle, so the fundamental wave frequency of the cylinder pressure is

$$f_0 = \frac{720n}{60} = 12n \text{ (Hz/s)} \tag{1}$$

According to the Nyquist sampling theorem, the sampling frequency should be at least twice the fundamental frequency of the cylinder pressure to ensure that the sampling is not distorted. Thus, the sampling frequency is

$$f_1 = 2f_0 = 24n \text{ (Hz/s)} \tag{2}$$

In the sampling of engine parameters, the sampling frequency of cylinder pressure should be the highest among the parameters, so the total frequency of the acquisition card should be

$$f = if_1 = 24 \times i \times n \text{ (Hz/s)} \tag{3}$$

FIFO

FIFO is the acronym of first-in-first-out. Acquisition cards generally have FIFO registers, used to cache high-speed acquisition data. To determine the FIFO parameter, that is, the size of the FIFO, it is necessary to know the relevant parameters, such as the sampling frequency, the delay of the system, and the time for the data to be moved to the buffer. The main idea is that the speed of moving in must be less than the speed of moving out. The time for the FIFO is set to be filled to t, the size of the FIFO to M, and the frequency of the acquisition card to f. Then,

$$t = \frac{M}{f}\ (\mathrm{s}) \tag{4}$$

The acquisition of engine parameters must occur in real time. In order to obtain the effect of real-time acquisition, the program adopts the double-buffer sampling algorithm. The board provides the FIFO half-full interrupt function, which can be used to realize the double-buffer continuous data collection function.

3.2. Electronic Ignition System Design

The SCM control system, with the SCM as its core, is equipped with a peripheral circuit and software and has several functions. It consists of hardware and software. The former is the basis of the system, while the latter is used for proper adjustment and adoption of the hardware to complete the tasks of the control system. Generally speaking, as the tasks of the control system change, hardware and software configurations vary. Therefore, the design of the SCM control system should include the hardware and the software. In order to ensure the normal operation of the system, the anti-interference capacity should be considered in the design of hardware and software. The engine's electronic system consists of a signal-collecting system and modules concerning the system control, display, and ignition control.

3.2.1. Signal Acquisition System

The rotary encoder is connected to one end of the camshaft, and the output TDC signal and counting signal are connected to the photoelectric isolator. After photoelectric isolation, the external interrupt and counting interrupt are input.

3.2.2. System Control Unit

An At89s52 microcontroller, produced by atmel, is adopted. AT89S52 (Atmel Corporation, San Jose, CA, USA) is a low-power, high-performance CMOS 8-bit microcontroller with an 8 k bytes ISP (in-system programmable) Flash read-only program memory that can be repeatedly erased 1000 times. Integrated with a universal 8-bit central processing unit and an ISP Flash storage unit, the powerful microcomputer AT89S52 provides a cost-effective solution for many embedded control applications.

3.2.3. Display Unit

The basic principle of the dynamic interface of LED displays is to use the "visual retention" effect of the human eye. The interface circuit connects the 8 pen segments, a to h, of the display in parallel to form the "field port". The common COM of each display is independently controlled by the I/O line, called the "bit scan port". When the CPU sends a font to the field output, all monitors can receive it. Which display is lit depends on which common pole of the LED display is connected to the output end of the scanning port at the time. The so-called dynamic is the use of the cyclic scanning method, time-sharing in turn through the common pole of each display so that each display turns on. When the scanning speed reaches a certain level, the human eye cannot distinguish it, and it is thought that the various displays light up at the same time.

3.2.4. Ignition Control Unit

(1) Ignition coil

Due to the fact that the ignition processes of natural gas and air mixtures and that of gasoline and air mixtures are significantly different, the ignition temperature of natural gas under normal pressure reaches 537 °C, which is much higher than gasoline. In order to obtain reliable ignition, the ignition energy required is large, so the selection and matching of the coil is a very important part of the high-energy ignition electronic control system. By calculating the power-on time, it can be ensured that the ignition energy of the engine is kept at a certain high level. An IG-9006 ignition coil (Tianjin Sparry electronic Technology Co., LTD, Tianjin, China) was selected for this project. Given the distinct discharge principles of the inductor and capacitor, it should be noted that only the IG-9006 ignition coil can generate the breakdown voltage if the spark channel is blasted out by external flows. Even if the capacitor has sufficient energy, the spark channel cannot be reformed once the energy in the ignition coil runs out [14].

(2) Ignition module

The ignition module mainly includes a thyristor switch control circuit, a voltage regulation control circuit, and a capacitor switching control circuit. The thyristor switch control circuit is the control core of the whole system, and its on or off state controls the on–off state of the discharge circuit: When the thyristor is off, the system is in a state of charging the capacitor; when the thyristor triggers the closure of the moment, the capacitor discharges ignition to the ignition coil. The ignition module model DZ491Q (Brilliance Jinbei Automobile Co., LTD, Shenyang, China) was selected. This ignition module model, DZ491Q, has an advantage over the previously modules used in engines [15], as this ignition controller has the functions of primary current rise rate control, closing angle control, shutdown and power-off protection, and overvoltage protection.

(3) Spark plug

The spark plug introduces the pulsed high-voltage electricity formed by the ignition coil into the combustion chamber, generating an electric spark between the electrodes and igniting the mixture. An important aspect of the spark plug is the thermal characteristic, which is a sign of maintaining the self-cleaning temperature of the spark plug and the ability to withstand the heat load. Spark plugs can be divided into the hot type, the cold type, the medium type. Considering the structure of the designed natural gas engine and the combustion characteristics of natural gas fuel, medium and cold-type spark plugs were selected.

The spark plug electrode gap is large, which is conducive to increased mixture contact between electrodes and rapid ignition, thereby improving the performance of the engine and its reliability and stability in an idle state. But it is limited by the ignition voltage and ignition energy. Natural gas engines are difficult to start at low temperatures, even more so than gasoline engines. An important reason is the physical and chemical properties of gaseous fuels. Because the molecular structure of natural gas contains more hydrogen (compared with gasoline), more water vapor is produced when burning, which produces a spark plug electrode shunt in the engine, destroying the spark formation rhythm. Therefore, when burning natural gas, the optimal gap of the spark plug electrode is slightly wider. An FK20HR11 spark plug (DENSO Corporation, Aichi, Japan) with unique technology from DENSO was selected.

The microcontroller program was written in the MCS52 microcontroller (Intel Corporation, Santa Clara, CA, USA) C language and solidified into 89S52 pieces after compilation. The software design principle diagram is shown in Figure 7.

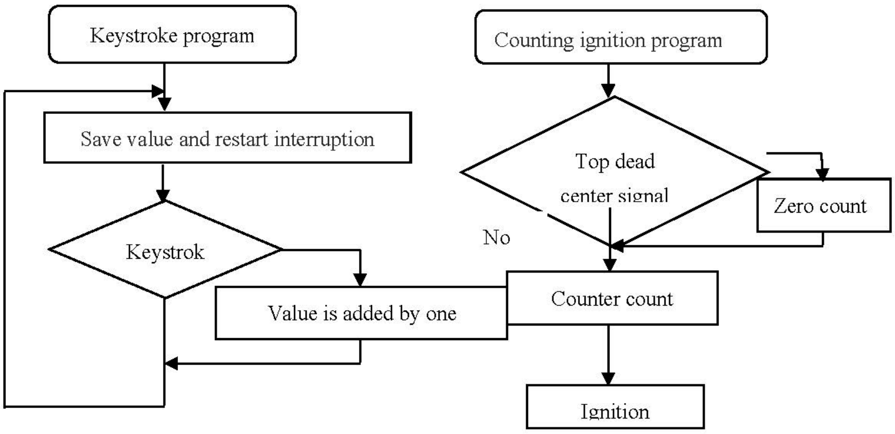

Figure 7. Schematic diagram of software design.

4. Test Methods and Analysis

4.1. Impact of Ratios of High- and Low-Pressure Compound Gas Supply on Emission

When high-pressure gas was ejected through the high-pressure valve to the subsidiary combustor, a higher concentration of the mixture was generated within it, because the injection timing was close to the top dead center and the mixing time was short. In addition, it was easier to generate NOX than HC, as there were strong eddy and turbulent flows in the eddy chamber, which could increase the combustion speed and result in high temperatures. The low-pressure gas was ejected through the low-pressure valve in the inlet channel to generate a homogeneous mixture in the main and subsidiary combustors, with a relatively low concentration of the mixture and slow flame propagation, especially when it spread to the edge of the combustor, where the temperature was low. As the combustion deteriorated, a considerable amount of HC was generated. Accordingly, the authors applied different ratios of high- and low-pressure gas supply to study its impact on combustion emission. The impact of increasing ratios of high-pressure gas on the emissions is shown in Figure 8, with an ignition advance angle of 22 degrees and a rotation speed of 1000 r/min.

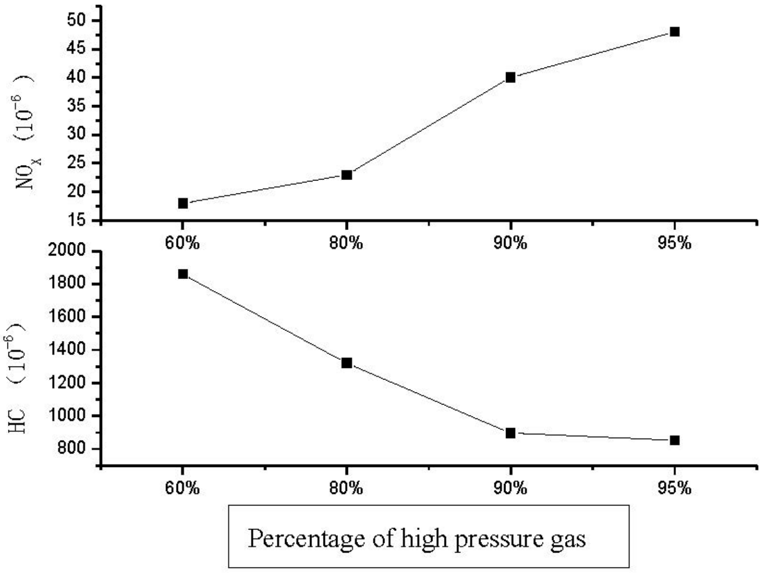

Figure 8. Impact of the high-pressure gas ratio on emission at 1000 r/min.

4.2. Impact of the Ignition Advance Angle on the Combustion Process

On the basis of compound gas supply, the low-pressure injection valve began to inject when the inlet valve was opened, and the high-pressure injection valve began to inject at 140 degrees before the top dead center. The injection duration was 70 °CA for the gas supply, and the engine speed was 1000 r/min. An indicator diagram of different ignition advance angles under this condition is shown in Figure 9.

As shown in Figure 9, when the ignition advance angles were 10 °CA and 15 °CA, the ignition was unfavorable for combustion, which led to low maximum pressure and affected engine operation. When the ignition advance angles were 25 °CA, 30 °CA, and 35 °CA, the maximum combustion pressure decreased significantly compared with that when the ignition advance angle was 20 °CA, which is due to the fact that when the high pressure natural gas was compressed and injected into the cylinder before the top dead center, the gas was strongly expanded and endothermal. A previous study demonstrated that, when the crankshaft angle was at 250 °CA (BTDC), the velocity was approximately 2 m/s in both the experiment and simulation, and a vortex was located in the center of the combustion chamber [16]. When the ignition advance angle was too large and close to the end point of high-pressure injection, the temperature in the cylinder was less subjected to the diffusion and heat absorption of natural gas. As a result, the mixture at a lower temperature in the cylinder was not fully ignited with the same ignition energy, which led to decreased combustion pressure in the cylinder and caused flame-out when a larger ignition advance angle was tested. When the ignition advance angle was 20 °CA, the natural gas could be fully ignited and the maximum combustion pressure of the engine was moderate. The maximum pressure appeared 15 °CA after the top dead center, so the advance angle could be determined as the best for the compound gas supply under this condition. Similarity, a previous study reported that in the final phase of compression, the tumble was compressed to the lower cylinder head and disappeared at 20 °CA (BTDC). By maximizing the tumble when the trailing spark electrode was located at the posterior of the tumble zone, the flame propagation speed could be increased. In addition, the fuel in the chamber's center and rear could be consumed without delay [16].

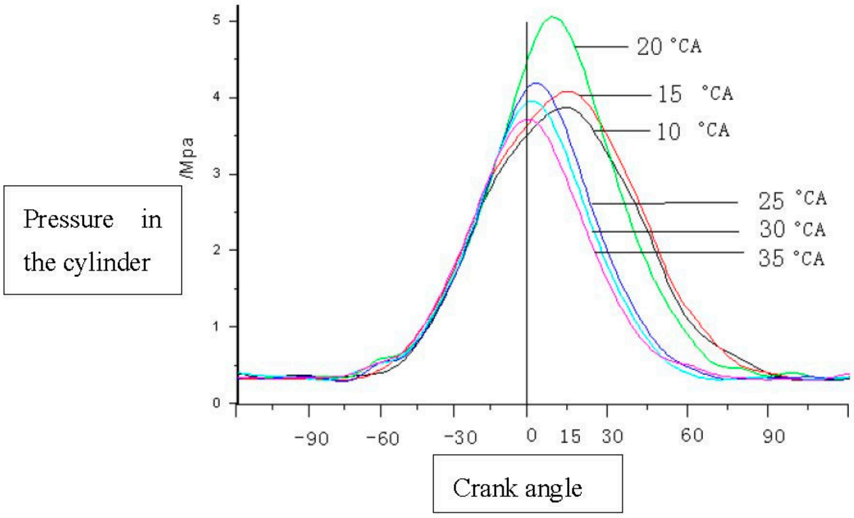

Figure 9. Indicator diagram of different ignition advance angles with compound gas supply.

4.3. Impact of High-Pressure Gas Injection Time on Combustion Emission

In the final phase of compression, the tumble was compressed to the lower cylinder head and disappeared at 20 °CA (BTDC). By maximizing the tumble when the trailing spark electrode was located at the posterior of the tumble zone, the flame propagation speed could be increased. In addition, the fuel in the chamber's center and rear could be consumed without delay [17]. With a compound gas supply, the low-pressure injection valve started to inject gas when the inlet valve was opened, with a duration of 200 °CA. The high-pressure injection valve was tested with different advance angles, with an injection duration of 110 °CA. The high-pressure natural gas injected by the high-pressure injection valve directly entered the subsidiary combustor. From the ending of the high-pressure valve injection to the beginning of the spark plug ignition, the rich mixture continued to spread from the subsidiary combustor to the main combustor, leading to a gradually decreased concentration of the mixture and gradually increased mixing degrees of the remaining mixture in the subsidiary combustor, as well as a gradually increased concentration of the mixture in the main combustor. As shown in Figure 10, with the advancement of the injection time of the high-pressure valve, the end time of injection gradually progressed away from the ignition time, enabling the high concentration of the mixture in the subsidiary combustor to have more time to diffuse to the main combustor. Accordingly, the oxygen concentration in the subsidiary combustor also increased, and the mixing degree of the remaining mixture in the subsidiary combustor was also enhanced. Therefore, when the spark plug ignition was performed, the mixture was combusted more fully, with a higher engine speed, less HC, and more NO_X.

Previously, Douville [18] investigated the combustion and emission characteristics of a single-cylinder, naturally aspirated, two-stroke engine and a six-cylinder, turbocharged, two-stroke engine, which operated at medium to high speeds and moderate loads. According to the experiment results of Douville [18], peak cylinder pressure and NOx emissions increased with the advancing injection timing due to the associated compression effects and higher in-cylinder temperature; in addition CO emissions initially decreased and then leveled off with advancing injection timing due to the improved oxidation with earlier fuel introduction. However, the trends of HC emissions, soot emissions, and thermal efficiency were dependent on the engine type. Dumitrescu [19] and Trusca [20] also conducted experiments on the single-cylinder, naturally aspirated engine with a retrofitted electric control system for the injector at medium speed and low loads; the conclusions of Dumitrescu [19] and Trusca [20] were consistent with those of Douville [18] in terms of NOx and CO emissions, although there were controversies in terms of thermal efficiency and HC emissions due to the modifications in the injector system. In the current study, the oxygen concentration in the subsidiary combustor increased, as did the degree of mingling of the remaining mixture in the subsidiary combustor. As a result, when the spark electrode was ignited, the mixture was burned more completely, resulting in a higher engine speed, less HC, and more NOX. These properties of the current system demonstrate profound effects on the combustion and emissions of pilot-ignited direct injection natural gas engines.

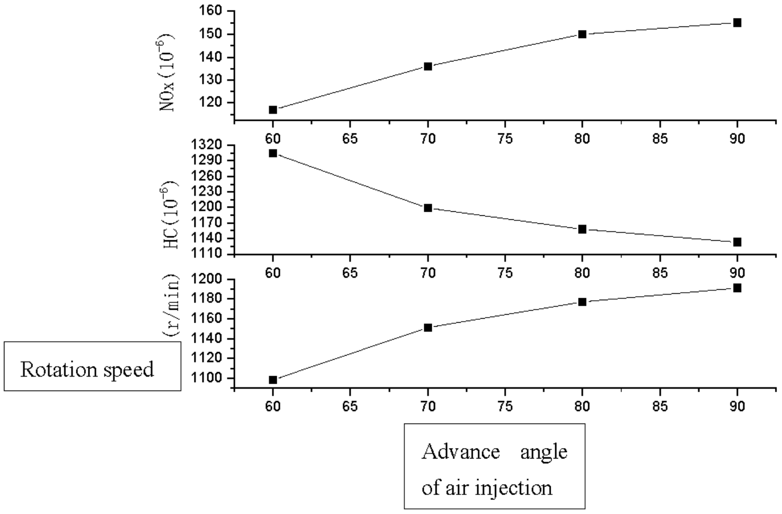

Figure 10. Impact of high-pressure gas injection time on emission.

4.4. Load Characteristics of the Engine

The load characteristics and emission characteristics of the engine at 1200 r/min are shown in Figures 11 and 12, respectively.

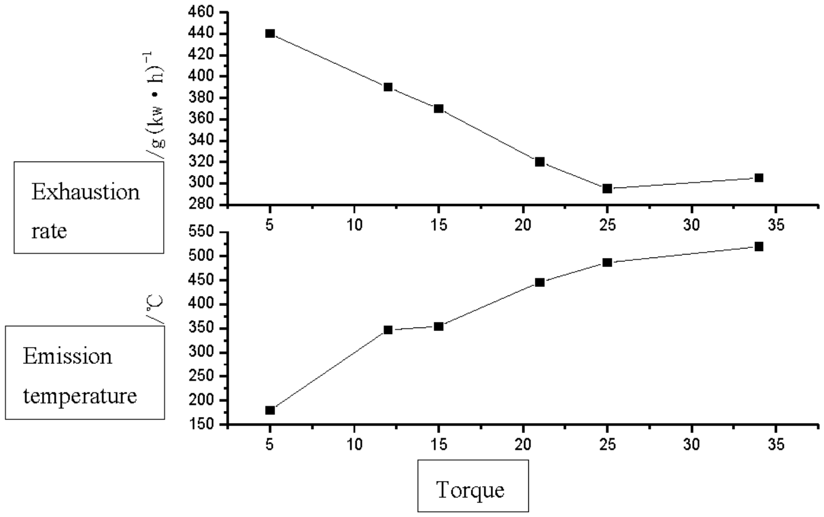

Figure 11. Load characteristics of the engine at 1200 r/min.

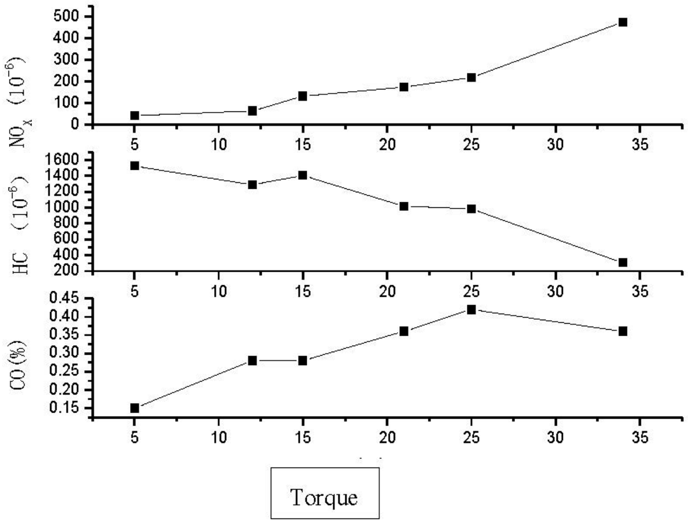

Figure 12. Emission characteristics of the engine at 1200 r/min.

As shown in Figure 12, as the load increased, the emission temperature of the engine as well as the amounts of NO_X and CO also increased, while that of HC decreased.

5. Conclusions

The following conclusions can be drawn according to the bench test of a spark-ignition natural gas engine with compound gas supply:

(1) During the study of the operating process of the spark-ignition natural gas engine, the impact of the ignition advance angle on the engine's performance was analyzed and the best ignition advance angles under different conditions were determined.
(2) The combustion conditions of high-pressure gas and low-pressure gas in the engine combustor varied greatly. The ratios of high-pressure gas and low-pressure gas both had great impacts on the performance and emission of the engine.
(3) The shape of the main combustor has a great impact on the performance of the engine, that is, a shorter propagation distance can reduce the generation of HC. Reducing the diameter of the channels between the main and subsidiary combustors can enhance the stratification and facilitate the secondary ignition, thus achieving lean combustion in the main combustor and inhibiting the generation of NO_X.

The best ignition advance angle under different conditions was determined using a spark-ignition natural gas engine. The ratios of high-pressure gas and low-pressure gas greatly impacted the performance and emission of the engine. The reduced diameter of the channels between the main and subsidiary combustors was able to enhance the stratification and facilitate the secondary ignition. CNG is a promising alternative fuel because it is readily available, it produces pure emissions, and its performance is comparable to that of fossil fuel; in other words, its combustion can accomplish the same efficiencies as liquid-based petroleum fuel. CNG provides a lower BTE than liquid-based fuels like diesel oil or gasoline. However, CNG has a lower BSFC than diesel and petroleum fuels. CNG combustion always produced higher temperatures for exhaust gases than gasoline combustion did. CNG was also discovered to have a lower peak cylinder gas pressure than diesel operation. CNG combustion produces slightly less power than diesel, but this can be circumvented by combining CNG with hydrogen, which increases power output. However, further studies to increase the combustion power of CNG and outcompete the diesel engine are critical.

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