Model Investigation of Natural Gas Engine Performance to Achieve Variable Heat/Electricity Ratios for a CCHP System by Varying Spark Ignition Timings

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Abstract: For electric reliability and to save energy, the distributed power generation combining cooling and heating supply called a CCHP system for architectures has many potential advantages and is widely adopted to provide electric power and to satisfy local heating and cooling loads by waste heat recovery with low carbon intensity. However, the current CCHP system usually has a fixed ratio of the power and heat due to the features of its power unit, which leads to difficulties in the load management. In this paper, based on the operation of an internal combustion engine fueled with natural gas, a novel method is proposed and studied to achieve a controllable rate of heat/power to meet different load requirements of the electricity and heat (cooling or heating loads). By varying the ignition timing of the spark ignition engine, the combustion process within the cylinder can be adjusted to occur at different crank angles so that the engine crank shaft output power (related to the generated electricity) and the heat from the exhaust gas are changed accordingly. To study the effects of ignition timing on engine power and exhaust heat energy, a two-zone model was established with a predictive combustion model. The changes in the combustion process, output power, exhaust gas temperature, and heat energy were mostly our concern. The results show that the heat/electricity ratio can be adjusted from normally 1.0 to 1.6, and they can be controlled independently under partial load operating conditions. To solve the potential thermal failure of the turbine, the extraordinarily high exhaust temperature will be adjusted by compressed air.

Keywords: CCHP; heat/electricity ratio; natural gas engine; ignition timing

1. Introduction

With the development of the economy, people pursue a higher quality of life. The living environment, such as shopping centers, hospitals, office buildings, and residential communities, needs an air condition system for refrigeration in the summer and heating in the winter [1,2]. In terms of energy efficiency and environmental protection, relying solely on commercial electric power is not so economical and environment friendly because of the low electric generation efficiency and high environmental cost. Combining the cooling, heating, and power (CCHP) system has the potential of decreasing energy consumption and pollutant emissions by generating electricity locally and recycling the waste heat for cooling and heating [3–5]. A CCHP system can achieve efficiency higher than 90%; however, because of the regional, seasonal, and diurnal variations, the system’s configurations and especially the operating strategies need to be studied further to meet the requirements and achieve a better operating economy [6,7]. Qin et al. [8] and Rey et al. [9] studied an efficient algorithm for the operation and control of the CCHP system. Lai and Hui [10] and Zheng et al. [11] proposed a novel thermal storage strategy for a CCHP system, which can receive 3.79% and 20.22% annually in total costs. Wang et al. [12] analyzed the influence of...
building type and climate zone on the performance of the CCHP system, which proved that a heat storage tank or gas-fired boiler is required.

The above studies prove that the current CCHP system is limited by the so-called heat-electricity ratio. Once the system is constructed, the power generation unit (PGU), either a turbine or an internal combustion engine, cannot change the rate of the generation and the exhaust heat easily without adding extra-devices. Therefore, most CCHP systems need to add devices to store the excess heat or add a gas-fired auxiliary boiler to supplement the heat shortage. Thus, those CCHP systems should be optimally designed and controlled to minimize the system’s construction cost and operating energy consumption.

Currently, the research on the cogeneration of gas-fired internal combustion engine mainly focuses on the selection of system configuration and the optimization of operating conditions. These two factors directly determine the economic benefits and energy saving and environmental protection performance of the cogeneration system.

Abbasi et al. [13] evaluated the combined cooling, heating, and power generation system with a gas internal combustion engine, gas turbine, Stirling machine, and molten carbonate fuel cell as the prime movers from the three aspects of system economy, energy saving, and environmental-protection ability. The decision method shows that when the internal combustion engine is used as the prime mover, the overall performance of the system is better in terms of energy saving effect, carbon emission reduction, and annual cost saving. Taking a hospital in Italy as an example, Luca et al. [14] established a dynamic simulation network of the energy system of the cogeneration system and studied the optimal system capacity under the premise of long-term uncertainties in energy demand. The cogeneration system uses natural gas internal combustion engine as the driving energy. The author’s model calculates the energy-saving performance of the internal combustion engine in the power range from 600 kw to 1600 kw. Wang Zhihe et al. [15] took a six-story hotel in Shanghai as the research object; established a linear optimization model of the combined cooling, heating, and power system; and analyzed the economic performance and environmental performance of the four different configurations of the combined supply system. They determined the type and capacity of the optimal drive machine for the system. Shi Rutao of Shandong University [16] carried out Simulink modeling for the combined cooling, heating, and power-generation system based on natural gas internal combustion engines. In order to solve the problem of low system efficiency caused by more power generation than the user’s electricity demand, three configuration combinations of the combined power supply system were proposed. In addition, the influence of natural gas internal combustion engine operating conditions on the refrigeration of the subsequent absorption chiller was studied. The results showed that the power change of the internal combustion engine and the working condition changes of the cooling water flow would have a greater impact on the operating status of the subsequent lithium–bromide absorption chiller.

For the operation strategy of the combined cooling, heating, and power system, scholars at home and abroad have carried out many optimization studies. Most scholars use the method of establishing an optimization function for the combined supply system and use the optimization algorithm to solve the optimal operation control method of the system.

Ehsan et al. [17] studied a co-generation system with new energy as the driving energy. The new energy driving system includes wind power generators, fuel cells, photovoltaic power generation, and tidal steam turbines. Zhu Han [18] developed an optimization software for the economic and environmental benefits of the combined cooling, heating, and power system and improved the software according to the “heating to determine the electricity” mode commonly used in practical applications in China. The cogeneration system is analyzed, and the results show that, compared with the traditional energy system, the operation strategy of the cooling, heating, and power cogeneration system still needs to be further studied after the system is installed, and its strategy optimization has a very important impact on the economical and energy-saving operation of the system.

The above research shows that the predecessors carried out related research on system cost saving and energy saving from the aspects of system configuration selection, operation
strategy optimization, etc., but less research on the working conditions of the system’s prime mover.

Currently, natural gas is widely used as fuel for PGU and CCHP systems: It is cheap, easy to gain, and clean. Based on the internal combustion engine technology, for a CCHP system with a natural gas-fueled internal combustion engine as its PUG, a novel method is proposed to achieve a variable heat/electricity ratio.

The operating principle of the natural gas engine [19,20] is that the gas mixes with the air before entering the cylinders. The mixture is inducted into the cylinder and is compressed by the up-moving piston. The compressed mixture is ignited before the top dead center. The ignited mixture burns in the form of flame propagation throughout the cylinder. The burned natural gas increases the temperature and pressure of the gases within the cylinder, which pushes the piston downward during the expansion stroke and output torque through a connecting rod and crankshaft mechanism. Near the piston’s bottom dead center, the engine exhaust valves open, and the burned gas is discharged at a high temperature. Generally, the engine is set to run under the best thermal efficiency condition: that is, to output its maximum power to drive the generator to give maximum electricity. The heat balance of the engine indicates that the brake power takes about 30% and the exhaust gas heat energy takes about 30% of the total fuel energy. In this case, obtaining more electricity requires higher engine power output, which results in more exhaust heat energy discharged and vice versa. That is the reason why the PGU CCHP system has a limited heat/electricity rate.

It is known that the combustion phase is controlled by the spark timing for the spark ignition (SI) engines. Thus, if the spark timing is postponed, the combustion within the cylinder is postponed, which will result in a reduction in the engine’s power output and the increase in the exhaust gas heat energy. That is to say that the heat/electricity ratio of a CCHP system can be controlled by spark ignition timings without using extra heat devices and control systems. Then, the constrained heat/electricity rate is broken.

In this study, GT-Power software is used to build a model to simulate engine performance under different spark timings. The engine combustion process occurring in the cylinder and the exhaust temperature is particularly concerned. The relationship between the spark timing, the engine power, and the exhaust heat energy is investigated. Moreover, the limitation is focused on the exhaust temperature, 650 °C, for the safe operating of the turbine. The results can be used to establish the engine control strategy of spark ignition timing.

2. Experiment Setup and Simulation Model
2.1. Engine Specifications and Measurement Apparatus

In this paper, a 12V190 natural-gas SI engine was chosen, which was modified from the same type of diesel engine. Its technical specifications are listed in Table 1. It is a V-type arranged engine, and each side has 6 cylinders matched with an independent turbocharger, SJ150-9C. The rated power is up to 600 kW at speed of 1000 rpm.

| Parameters                  | Specifications                                |
|-----------------------------|----------------------------------------------|
| Engine type                 | 12 cylinders in 60° V type arrangement       |
| Stroke number               | 4                                            |
| Bore × Stroke (mm)          | 190 × 210                                    |
| Connecting rod length (mm)  | 410                                          |
| Total displacement (Lit.)    | 71.45                                        |
| Compression ratio           | 11                                           |
| TDC clearance height(mm)    | 1.8                                          |
| Firing order                | 1-8-5-10-3-7-6-11-2-9-4-12                  |
| Rated power (kW)            | 600                                          |
| Rated speed (rpm)           | 1000                                         |
| Turbocharger                | Dual, SJ150-9C                               |
| FMEP@1000 rpm(bar)          | 0.7                                          |
The engine test setup was illustrated in Figure 1. This engine test was carried out on the 12V190 engine of Shengdong Group. The parameters to be measured mainly include the combustion pressure in the cylinder, the pressure before and after the turbine, the pressure before and after the compressor, the temperature before the intercooler, the intake pipe pressure, the airflow, and the gas flow.

**Figure 1.** Schematic of the experimental setup.

2.2. Simulation Models

Based on the GT-Power software and the engine features, the developed engine model consisting of the engine crank object, engine cylinder object, turbine-compressor object, intake, and exhaust systems is shown in Figure 1. To better understand the developed models by spark retarding to change the rate of the engine output power and exhaust heat energy, except for the basic geometry parameters of 12V190 natural gas engine, some important systems and related models are introduced below.

2.2.1. Intake and Exhaust System

As shown in Figure 2, the intake system consists of a filter, fire barriers, compressors, intercoolers, throttle, intake runners, intake ports, intake valves, and those pipes connecting the objects in the system. Likewise, the exhaust system includes exhaust valves, exhaust ports, turbines, exhaust runners, mufflers, and the pipes necessary to connect these objects. The specific geometry of the pipes, runners, ports, fire barriers, and intercoolers in the intake and exhaust systems was set according to the actual engine specification. The discharge coefficients of the intake and exhaust valve versus the lift profile were provided by the engine manufacturer. The flow resistance, discharge coefficients, and heat transfer coefficients of other parts were calibrated carefully according to the experimental data, including the pressure and temperature in the manifold, and the mass flow rate of air and natural gas.

The turbine converts part of the thermal energy of the exhaust gas into air compression energy, which results in the rising of air pressure and temperature. There is a special sub-model to simulate the operation of the turbocharger and compressor. The compressor objective modeled by a compressor MAP consisting of the pressure ratio, mass flow rate,
and efficiency calculated the air mass flow rate, speed, outlet temperature, and pressure and consumed power at a given compressor speed and input power. Likewise, the turbine objective modeled by a turbine MAP consisting of the pressure ratio, mass flow rate, and efficiency calculated the output power according to the mass flow rate of exhaust gas, speed, temperature, and pressure of the inlet and outlet.

**Figure 2.** Full engine model in GT−Power.

### 2.2.2. Engine Combustion and Flow Model

There are three kinds of combustion models available for the combustion process simulation, which are non-predictive, semi-predictive, and predictive combustion models. The non-predictive and semi-predictive combustion models do not take into account the physical models in the cylinder, which is not suitable to study the variable affecting the burn rate significantly. In this investigation, the spark timing directly influencing the combustion phase varies to achieve the variable heat/electricity ratio. Therefore, the predictive combustion model and the spark-ignition turbulent flame model must be selected to study the effects of variable spark timing on the flame-propagation burning process within the cylinder.

The predictive combustion model takes into account the cylinder’s geometry, spark location and timing, air motion, and fuel properties for predicting the burn rate for the homogeneous charge. A two-zone combustion methodology was applied during the combustion process. As shown in Figure 3, the total cylinder volume is divided into two
zones, the unburned zone and the burned zone, and is separated by the flame front. The burn rate is governed by the following four equations:

\[
\frac{dM_e}{dt} = \rho_u A_e (S_T + S_L) \tag{1}
\]

\[
\frac{dM_b}{dt} = \frac{M_e - M_b}{\tau} \tag{2}
\]

\[
\tau = \frac{\lambda}{S_L} \tag{3}
\]

\[
\lambda = \frac{C_{\lambda} L_t}{Re_t^{0.5}} \tag{4}
\]

where \(M_e\) is the entrained mass from the unburned zone into the burned zone through the flame front. \(T\) is the time of combustion. \(\rho_u\) is the density of the unburned mixture. \(A_e\) is the entrainment surface area at the edge of the flame front. \(S_T\) and \(S_L\) are the turbulent and laminar flame speeds, respectively. \(M_b\) is the mass of the burned mixture. \(T\) is the time constant. \(\lambda\) is the Taylor microscale length, which represents the dimension of ignition sites within the flame front. Hence, \(\tau\) represents the characteristic time scale for the burnup of a single ignition site. The Taylor microscale is computed from the turbulent integral length scale \((L_t)\) according to Equation (4). \(Re_t\) is the turbulent Reynolds number. \(C_{\lambda}\) is a constant called the Taylor microscale multiplier.

![Figure 3](image-url)

Figure 3. Two zone combustion methodology schematic diagram.

As mentioned above, the predicted burn rate is a function of turbulent and laminar flame speed. The laminar flame speed of natural gas is calculated by the following equation:

\[
S_L = [B_m + B_\phi (\phi - \phi_m)] (\frac{T_u}{T_0})^\alpha (\frac{p}{p_{ref}})(1 - 2.06EGR)^{0.77C_{EGR}} \tag{5}
\]

where \(\phi\) is the equivalence ratio, \(B_m\) is the maximum flame speed at equivalence ratio with value \(\phi_m, B_\phi\) is the laminar speed roll-off value, \(T_0\) and \(p_{ref}\) are 298 K and 101,325 Pa, \(T_u\) and \(p\) are the temperature and the pressure in the combustion chamber, \(\alpha\) and \(\beta\), respectively, are temperature and pressure exponent, and \(C_{EGR}\) is dilution effect multiplier. EGR is the mass fraction of the residual gas, including the internal and external residual gas. The coefficients of natural for calculating the laminar flame speed are shown in Table 2.

Table 2. The coefficients of natural for calculating the laminar flame speed.

| Fuel | Natural Gas |
|------|-------------|
| \(B_m\) | 0.397 |
| \(B_\phi\) | -1.649 |
| \(\phi_m\) | 1.061 |
| \(\alpha\) | 5.75\(\phi^2\) - 12.15\(\phi\) + 7.98 |
| \(\beta\) | -0.925\(\phi^2\) + 2\(\phi\) - 1.473 |
The turbulent flame speed is calculated by the equation below:

\[ S_T = C_{TFS} u' \left[ 1 - \frac{1}{1 + C_{FKG} \cdot \left( \frac{R_f}{L_t} \right)^2} \right] \]  

(6)

where \( C_{TFS} \) is a constant called the turbulent flame speed multiplier, \( u' \) is the turbulence intensity, \( C_{FKG} \) is another constant called flame kernel growth multiplier, \( R_f \) is the flame radius, and \( L_t \) is the turbulent integral length scale.

As mentioned above, the in-cylinder turbulence intensity and turbulent length scale are necessary to calculate the turbulent flame’s speed. The turbulence intensity and integral length scale were calculated based on the \( K-\epsilon \) approach [21]. The tumble coefficient associated with the valves was measured on a steady-state flow bench and provided by the engine manufacturer.

In addition, the head and piston geometries, necessary for the turbulence and combustion model, were also provided by the engine manufacturer. In this paper, the combustion chamber was simplified to be a flat head with a diameter of 190.0 mm, a cylinder, a piston with a bowl for which its diameter is 136.0 mm, and a depth of 18.5 mm. The spark location and the combustion chamber parameters are shown in Table 3.

| Table 3. Some flame object parameters. |
|----------------------------------------|
| Spark location                        | X (mm) | Y (mm) | Z (mm) |
|                                       | 0.0    | 0.0    | -3.0   |
| Head dome                             | X (mm) | Y (mm) | Diameter (mm) | Height/Depth (mm) |
|                                       | 0.0    | 0.0    | 190.0   | 0.0          |
| Piston cup                            | X (mm) | Y (mm) | Diameter (mm) | Height/Depth (mm) |
|                                       | 0.0    | 0.0    | 136.0   | 18.5         |

2.2.3. Model of Heat Transfer

The heat transfer model is used to calculate the convective heat transfer from the gas to the combustion chamber wall surface within the cylinder. A built-in Woschni heat transfer model is used to solve the convective heat transfer coefficient. This model calculates the in-cylinder heat transfer by Equation (7):

\[ h_c = C_{htm} \cdot B^{-0.2} \cdot p^{0.8} \cdot T^{-0.5} \cdot \omega^{0.8} \]  

(7)

where \( B \) is the cylinder bore(m), \( p \) and \( T \) are the gas pressure (kPa) and temperature (K) in-cylinder, and \( \omega \) is the local average gas swirling speed (m/s) in the cylinder. \( C_{htm} \) is the convective heat transfer coefficient multiplier used to calibrate the heat-transfer model. The temperatures of the head, cylinder, and piston were set to be 450 K, 400 K, and 500 K, respectively.

2.2.4. Engine Friction

In this study, the engine speed was fixed at 1000 rpm and the mean effective pressure of engine friction (FMEP) varied a little with the variation of the load. Hence, FMEP was set to be 0.7 bar, which is measured through the motoring test at wide-open throttle (WOT) by the engine manufacturer.

3. Model Calibration

Corrections for the gas environment include intake flow, intake temperature, intake pressure, and exhaust temperature.

Figure 4 shows the error curve of the comparison between the simulation results of the intake and exhaust parameters and the test value of 600 kW. It can be seen from the figure that the errors are controlled within 5%.
Figure 4. Comparison between simulation and test of intake and exhaust parameters. (a) Intake flow (b) Intake Pressure (c) Intake temperature (d) Exhaust temperature.

Figure 5 shows the simulated and measured cylinder pressure at the condition of 600 kW and 400 kW, 1000 r/min. The excess air/fuel ratio is fixed at 1.36. The spark ignition timing is set at –22 °CA.

Figure 5. Comparison of the simulated and measured cylinder pressure.
In the simulation, the initial spark size accounts for the energy to initiate the flame kernel and was adjusted to a value of 1 mm in diameter to fit the measured cylinder pressure trace. Although the simulated $p_{\text{max}}$ is a little lower than the test data under 600 kW case, there is no doubt that the error (<5%) between simulation and actual test data is acceptable.

At the same time, the discharging flow was calibrated too. For 600 kW operations, the simulation shows that the air flow rate is 1640 kg/h with a pressure ratio of 1.69, and the exhaust temperature is 860 K. Their errors are 3.96%, 0.05%, and 0.77%.

4. Results and Discussion

To investigate the influence of spark timing on engine power output and exhaust gas heat energy with the validated model developed for the 12V190 natural gas engine, the spark timing was set from 34° to 8° CA BTDC by a calculating step of 4° CA under different power at constant speed 1000 r/min. The crankshaft power, cylinder pressure, heat release rate, exhaust temperature, and energy balance were simulated and analyzed to determine the adjustable range of the ignition advance angle.

4.1. Engine Combustion Characteristics

Figure 6a illustrates the in-cylinder pressure at different spark timings. In the simulation, the engine throttle is kept at the opening of 600 kW. When the spark timing is delayed from 34° CA BTDC, $p_{\text{max}}$ decreases with postpone heat release. As a result, the brake power decreases while the exhaust temperature increases. When the spark timing is delayed to 14° CA BTDC and later, in Figure 6b, the curves of the calculated accumulated heat-release rate indicate that the combustion lasts until the exhaust valve opens. Therefore, if the spark timing is delayed after 14° CA BTDC, part of the mixture will burn after EVO during the engine exhausting process. In this case, more heat energy will be directly expelled to the exhaust gas in the form of higher temperature and pressure. Therefore, the ratio of the engine power and the exhaust gas energy is changed.

![Figure 6.](image_url)

**Figure 6.** (a) Effect of Ignition Delay on Engine Cylinder Pressure. (b) Effect of Ignition Delay on Engine Heat Release Rate.

4.2. Engine Brake Power

Under the same opening of the throttle, the engine power changes with the ignition timing. Figure 7 shows their relationship. There exists MBT. If the ignition time is earlier than MBT, the mixture combuts earlier, which results in more compression work and heat transfer. On the other hand, when it is later than MBT, the combustion’s released heat is less expanded, which results in less work delivered to the crankshaft and more heat expelled with exhaust gas.
In Figure 7, the MBT power is 608 kW, if it runs at the ignition timing of 8 °CA BTDC, the engine power is simulated to be 492 kW. The power reduction is 116 kW. The reduced work becomes the heat energy of the exhaust gas and cooling water.

### 4.3. Energy Balance in Different Working Conditions

Figure 8 shows each energy ratio after adjusting the ignition advance angle under the power of 600 kW–300 kW. As can be seen from the histogram of energy proportion of each part, the energy balance has little relation with power, but a great relation with ignition time. Taking the 600 kW condition as an example, the crankshaft output power increases first and then decreases with the delay of ignition advance angle, and the variation value is small. The cooling water’s energy decreases gradually with the delay of ignition advance angle. For example, when the ignition advance angle is delayed from 30 °CA BTDC to 26 °CA BTDC, the cooling water energy decreases by 2.0%. The exhaust energy increases gradually with the delay of ignition advance angles. For example, when the ignition advance angle is delayed from 26 °CA BTDC to 22 °CA BTDC, the exhaust energy increases by 2.2%. The miscellaneous loss decreases slightly, and it is always below 5% in all working conditions.
4.4. Exhaust Gas Temperature and Energy

In CCHP system, the exhaust gas is used to provide heat for the absorption refrigerator; thus, the exhaust temperature is a key parameter. When the ignition timing is delayed, the combustion process is thus delayed, and as a result, the temperature of the cylinder gas changes accordingly. Figure 9 shows the changed cylinder gas temperature.

Figure 9. Cylinder temperatures.

The exhaust valve opens at 55 °CA BBDC. The delayed combustion process also results in higher exhaust temperatures in the pipe. For example, if the ignition timing is delayed to 10 °CA BTDC, compared with MBT case, the exhaust temperature increases by 130 K. This indicates that the exhaust temperature can be adjusted flexibly by changing the spark timing.

As it is known, the heat energy of natural gas burned in the cylinder can be simply divided into three parts: the brake power, the exhaust heat energy, and the heat transferred to the cooling system. Figure 10 shows the ratio of the simulated exhaust heat energy. In the figure, the marked powers are the engine power at MBT. The delayed ignition timing increases the ratio of the exhaust heat energy/power. The tendencies are quite similar. The ratio of the exhaust heat energy at 34 °CA BBDC is about 32%, while the ignition is delayed to 8 °CA BBDC, the ratio increases to 50%. Figure 11 shows the variation trend of exhaust energy with load at different ignition advance angles. It can be seen from the figure that with the increase in the load, the exhaust energy increases significantly. For example, when the ignition advance angle is 22 °CA BTDC, when the power increases from 400 kW to 500 kW, the exhaust energy increases by 88.7 kW.

Figure 10. Exhaust gas energy.
The increased exhaust heat energy can be used by the CCHP system for heating or refrigerating. That is to say that when there is a higher cooling demand in the summer or heating demand in the winter of a CCHP system, it can be simply treated by delaying the engine spark timing without extra heating equipment.

4.5. Exhaust Gas Temperature Control

From the above analysis of the combustion characteristics in the engine cylinder, it is known that retarded ignition can make the engine produce more exhaust energy, thereby providing more heat load. However, the increased exhaust heat energy means a higher exhaust temperature. The solid line in Figure 12 is the calculated exhaust temperature, and it can be higher than 1000 K when ignition timing is delayed to 8°CA BBDC. The high exhaust temperature exceeds the limit of the turbine, 650°C. To solve the problem, the compressed air is guided to the exhaust before it enters the turbine by a controllable valve. The improved exhaust system is shown in Figure 13. Add the thermostatic valve at the red circle position in the picture. The temperature control valve takes the temperature of the turbine intake pipe as a signal, it is controlled by the set opening and closing temperature. When the temperature is higher than the opening set temperature, the valve opens, and when the temperature is lower than the closing set temperature, the valve closes. Due to the dilution of the low-temperature air, the exhaust temperature can be controlled below the limit as shown in Figure 12. The reasonable flow rate directly to the exhaust is shown in Figure 14.
4.6. Crankshaft Power Control

Delayed ignition means delayed combustion, resulting in a drop in crankshaft power. Figures 15 and 16 are the comparisons of the flow and crankshaft power before and after improvements. As observed from the figure, since the improved exhaust system allows more compressed air to enter the exhaust port for cooling, increasing the draft pipe increases the intake air flow, thereby increasing the crankshaft’s power. Therefore, in order to ensure the stable output of crankshaft power, in this paper, a throttle controller is added to the model after improving the intake system to adjust the throttle opening to ensure the required crankshaft’s output power. The controller is connected with two sensors and a driver. The sensor’s sensing signals are the crankshaft speed signal and the average effective pressure signal. The driver drives the throttle valve and controls the throttle’s
valve opening to make the average effective pressure meet the set requirements in order to achieve the purpose of power control.

![Figure 15](image1.png)

Figure 15. Comparison of intake pipe flow before and after improvement.

![Figure 16](image2.png)

Figure 16. Comparison of crankshaft power before and after improvement.

With the delay of ignition, in order to ensure that the average effective pressure meets the requirements, the throttle valve controller must increase the throttle valve opening so that the intake air volume increases and more fuel is provided for the cylinder to make up for the loss of power. Figure 17 shows the power performance of the engine under constant power operation after adding the throttle controller. It can be seen that when the ignition advance angle is greater than 12° CA BTDC, the crankshaft power in the power range of 300–600 kW can be well maintained at the control power level. When the power is 600 kW and when the ignition is later than 12° CA BTDC, the crankshaft power begins to decrease significantly; when the power is 500 kW and when the ignition is later than 10° CA BTDC, the crankshaft power decreases to a certain extent; crankshaft power has been good at 400 kW and 300 kW. It can be seen that when the ignition advance angle is changed and the throttle’s valve opening correspondingly increased or decreased, the crankshaft output power of the engine can be controlled at a certain power level.
For safety, the postponed ignition is constrained that the combustion occurred within the cylinder ends at the EVO, and the exhaust temperature is not higher than 650 °C. The simulated engine brake power and exhaust heat energy are shown in Figure 18, where ‘electricity’ is the engine brake power to follow the word in CCHP. Under full load conditions, as shown in Figure 7, ignition delay results in a decrease in brake power, and the maximum heat/electricity is about 1.7, while under partial load conditions, for example, when the engine runs at 400 kW, the engine power can be maintained to supply more exhaust heat. The engine can be controlled by postponing the ignition timing and enlarging the throttle’s opening accordingly. This means that the CCHP system can change heat/electricity according to the required heat or electricity without extra equipment.

This paper established a one-dimensional 12V190 engine model by using GT-Power to simulate the effects of ignition timing on engine power and exhaust heat energy, an important parameter of heat/electricity ratio in CCHP system. The predictive combustion model was adopted in the 12V190 engine model, and it is validated to obtain accurate results of engine brake power and exhaust heat energy under a wide range of spark timings. By the model and engine’s experimental tests, the following conclusions can be drawn:
1. The established GT-power model containing a predictive combustion model meets the requirement of the study well.

2. For a natural gas engine powered CCHP system, the idea of changing the heat/electricity ratio of can be reached by delaying the ignition timing of the SI engine. The delayed ignition timing causes a postponed combustion process and a reduced ratio of brake power and exhaust energy to meet the variable heating or refrigerating demand to the exhaust heat energy without extra equipment.

3. The range of ignition delay is simulated according to the combustion in the cylinder. In most cases, 10°CA BBDC is the limit for the combustion completion before the exhaust valve opens.

4. The extra higher exhaust temperature is harmful to the turbine’s operation safety and this can be coped by introducing the compressed air to the exhaust before entering the entrance of the turbine.

5. The model with the added throttle controller is operated at constant power, and the results show that when the ignition’s advance angle changes, the throttle valve opening changes accordingly, and the crankshaft’s output power level can be guaranteed within a certain range.

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