Knock combustion characteristics of an opposed-piston two-stroke gasoline engine

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Abstract
The spark plug of an opposed-piston two-stroke (OP2S) gasoline engine is arranged on the side wall of the cylinder liner, far from the center of the combustion chamber; the ignition core of the mixture is offset, the flame propagation distance is increased, the combustion duration is prolonged, and the knock tendency is severe. In this paper, a quasi-dimensional two-zone combustion model is used in GT-Power software to establish a thermodynamic process simulation model and a knock prediction model is included to analyze the effect and matching of the compression ratio, ignition timing, and other thermodynamic process parameters on the knock intensity and engine performance. The extended coherent flame model combustion model is coupled with the Huh–Gosman spray model in AVL-Fire software, and the AnB knock model is used to establish an in-cylinder combustion model and analyze the flame propagation and the knock response rate of the flat and pit piston during the combustion process. With an increase of the compression ratio, the temperature and pressure of the mixture in the combustion chamber increase at the time of ignition, which leads to the knocking combustion in the cylinder. With an increase of the ignition advance angle, the in-cylinder pressure and temperature increase rapidly, which increases the likelihood of knocking combustion. In comparison with the pit piston combustion chamber, the flame propagation speed of the flat piston combustion chamber is relatively slow, which increases the knock tendency. The results show that lowering the compression ratio and delaying the ignition can reduce the in-cylinder knock tendency by setting a compression ratio of 10.5 and an ignition advance angle of 20°CA. When the pit piston is used to organize the squish and inverse squish before and after the inner dead center, the flame propagation process can be promoted. The knock response rate of pit piston is 24.7% lower than that of flat top piston.

Keywords
compression ratio, ignition timing, knock, opposed piston, two-stroke gasoline engine
INTRODUCTION

With the application of modern design technology, the opposed-piston two-stroke (OP2S) engine has been reemphasized, represented by the OPOC diesel engines from Eco Motors and FEV and the opposed-piston engines such as the A47 and A48 from Achates Power. The OP2S engine has the characteristics of low vibration, low noise and high power density, and high thermal efficiency and has a wide range of applications in military auxiliary power plants, auxiliary power units, unmanned aircraft, and small submarine power. The combustion chamber of the OP2S engine consists of the opposed piston top surface and the cylinder liner wall. The OP2S engine is a kind of reciprocating engine with structures that are different from those of traditional engines; it uses an in-cylinder direct injection spark ignition combustion system with injectors and spark plugs arranged on the side walls of the cylinder liner to achieve separation of the scavenging process from the injection process and fast combustion organization. The spark plugs of the OP2S gasoline engine are arranged on the side walls of the cylinder liner, which are far from the center of the combustion chamber. This leads to a shift in the ignition core of the mixture, an increase in the flame propagation distance, a prolongation of the combustion duration, and a high risk of knocking combustion. The distribution of the mixture in the cylinder at the moment of ignition and the structural shape of the combustion chamber are therefore subject to new requirements. The knocking combustion in gasoline engines originates from the end-gas auto-ignition in the combustion chamber. It is influenced by a number of factors, which depend mainly on the parameters of the operating process such as the compression ratio and ignition timing and the shape of the combustion chamber.

The gasoline engine knocking phenomenon is closely related to the in-cylinder gas flow movement. Increasing the intensity of in-cylinder turbulence can increase the flame propagation speed and thus shorten the time required for the flame front surface to propagate to the end-gas mixture. The results of studies by Chen et al. and Yu et al. suggested that a rational design of the in-cylinder flow field structure was important for the development of combustion systems to increase the flame propagation speed while suppressing the detonation. Variations of the main flame propagation, knock intensity, and localized autoignition under different turbulence intensities were discussed. The numerical simulation of bursts began in 1955 when Livengood and Wu proposed the An B knock model, which is based on the assumption that the end mixture is uniform in both temperature and pressure. Halstead et al. proposed a model for the spontaneous ignition of hydrocarbon fuel shells in 1977. The model relies on a degenerate branching chain reaction to propose eight simple chemical reactions to simulate the spontaneous ignition of a combustible mixture. In 1988, Heywood proposed a two-zone deflagration model; the engine combustion chamber is divided into the combustion zone and the unburned zone. When the ignited zone is on fire, the flame reaches the unburned zone. In the flame transfer process, the unburned zone mixture is squeezed by the high temperature and pressure gas in the combustion zone and spontaneous combustion occurs. La fossas developed and validated a model of spontaneous ignition based on the AnB model of “precursor” detonation. Spontaneous ignition occurs when the “precursor” reaches a limiting value. Teraji et al., coupled the Universal Coherent Flame Model (UCFM) with the Livengood–Wu integral model and used the Livengood–Wu integral model as the ignition model. The coupled model is able to predict the time and location of the spontaneous ignition of the end mixture. Zhen et al. explored the knock combustion characteristics of methanol fuel by large eddy simulation (LES) coupled with a simplified reaction mechanism (21 components and 91 primitive reactions) for methanol. To identify the characteristics of the detonation peninsula, numerical simulations were carried out for other fuels. Three alternative C0 fuels with detailed chemistry and transport were used in a one-dimensional reaction wave propagation induced by temperature gradients. Linse et al. argued that the essential characteristics of the popping were stochastic in nature. The Gaussian probability density function of the burst is developed by applying statistics. At the same time, the probability density function coupled with the detailed chemical reactivity mechanism predicts the gasoline engine detonation. Robert demonstrated for the first time with the extended coherent flame model (ECFM) that large eddy simulations are useful in the qualitative analysis of combustion chamber knock combustion in real spark plug ignition engines. With the continuous improvement of computer performance, the numerical simulation of deflagration evolve from simple to complex, from zero-dimensional to three-dimensional, from simplified geometric models to complex geometric models, from simplified mechanisms to detailed mechanisms, from Reynolds averaging methods to large eddy simulation methods. In summary, the flame propagation velocity, equivalent ratio distribution, end mixture thermodynamic state, and ignition advance angle all have a significant effect on the detonation.

In-cylinder combustion in gasoline engines is a complex physicochemical and energy conversion process. The combustion boundary conditions mainly include the intake temperature, the intake pressure, the excess air
coefficient, ignition timing, and the exhaust gas recirculation (EGR) rate. Szwaja et al.\(^{23}\) carried out an experimental study on a single cylinder variable compression ratio gasoline engine. They found that as the ignition advance angle increased significantly from 4 to 18°CA BTDC, the peak in-cylinder pressure increased, with detonation occurring at 18 and 14°CA BTDC. The knocking experiments of spark ignition (SI) and compression ignition (CI) were performed by simultaneous pressure acquisition and high-speed direct photography. The knocking phenomena of the varying initial thermodynamic conditions were reproduced by a high-strength optical rapid compression machine with a flat piston design.\(^{24}\) Pan et al. used one-dimensional numerical simulation, to study the interactions of flame propagation, auto-ignition and pressure waves during various knocking combustion scenarios. Under forced turbulence conditions, the turbulent mixing with the colder fluid from the wall boundary layers leads to some decreases in the temperature of core mixtures.\(^{25,26}\) Wei et al.\(^{27}\) show that different levels of detonation can be achieved by adjusting the ignition timing for gasoline engines. Increasing the intake pressure at the same compression ratio increases the probability of knock combustion.\(^{28}\) EGR, injection strategies, and strong tumble flow-high EGR-Atkinson/Miller cycles can all suppress conventional and super knock in spark ignition engines.\(^{29}\) Ghanaati et al. investigated the in-cylinder knock phenomenon using a four-cylinder engine in combination with experiments and comparative one-dimensional simulation.\(^{30}\) The resonant frequencies of knock combustion ranged from 6 to 20 kHZ. Claudio Forte et al used computational fluid dynamics (CFD) combined with experimental data to propose a method to evaluate the effect of a dual spark ignition system on cyclical variation and knock combustion.\(^{31}\) In summary, the study of detonation based on systematic experiments and visualization techniques allows a macroscopic analysis of the complex physicochemical phenomena in the cylinder during knocking combustion. The combination of numerical simulations with the experiments is more conducive to the in-depth study of knock combustion.

On the basis of the above considerations, the factors of end-gas auto-ignition include the fuel octane number, the end mixture pressure and temperature, and the propagation time from the front of the flame to the end mixture. Both the compression ratio and the ignition advance angle affect the pressure and temperature of the mixture. The shape of the combustion chamber affects the flow in the cylinder and the propagation time from the front of the flame to the end mixture. In this paper, the compression ratio, ignition timing, and the shape of the combustion chamber are chosen for study as the main factors to determine their influence on the end-gas auto-ignition to analyze the knocking combustion characteristics of an OP2S gasoline engine. A matching analysis of the compression ratio and ignition timing and the calculation of knock intensity are carried out for the OP2S gasoline engine by establishing a thermodynamic process simulation model and a knock combustion prediction model. On this basis, the flame propagation and the knock response rate of the combustion process are analyzed by constructing an in-cylinder combustion model for both flat piston and pit piston combustion chamber structures of the OP2S gasoline engine to provide a theoretical foundation for optimizing the design of the OP2S gasoline engine.

2 | SIMULATION MODEL OF AN OP2S GASOLINE ENGINE

2.1 | Concept of an OP2S gasoline engine

As shown in Figure 1, the principle prototype of an OP2S gasoline engine, without the cylinder head and valve mechanism, realizes the exchange process in the cylinder
liner from the intake port to the exhaust port, with the advantages of high efficiency, high power density, and good balance. The OP2S gasoline engine achieves separation of fuel injection and the scavenging process and rapid combustion with a uniflow scavenging flow, direct injection, and ignition with dual spark plugs. A spark plug is installed on the cylinder liner to ignite the mixture. The toothed belt is chosen to synchronize the two output crankshafts of the intake side and the exhaust side of the OP2S gasoline engine. The parameters are shown in Table 1.

### 2.2 Modeling and validation

#### 2.2.1 One-dimensional model of the thermodynamic process

GT-Power software is used to model the thermodynamic process based on the structural style and principle of the OP2S gasoline engine. Because there is no special model for opposed pistons in GT-Power software, the cylinder of the OP2S gasoline engine should be equivalent to a single-piston cylinder, as shown in Figure 2. In the simulation model, the opposed-piston relative motion is equivalent to the single-piston motion.

Using the principle of equivalent volume change rate, the working process of the OP2S gasoline engine is equivalent as follows, as shown in Figure 3.

1. The motion of the opposed piston is equivalent to the motion of the single piston, and the working displacement corresponding to the crankshaft angle is the sum of the displacement values of the intake and exhaust piston.
2. The closest point to the opposed piston is defined as the top dead center (TDC), and the relative displacement of the piston at this point is defined as 0, which is equivalent to the relative displacement of the piston within 360°CA.
3. The change law of cylinder volume is determined by the movement law of the opposed piston. In the modeling process, the relative displacement of the opposed piston is input as the equivalent displacement, and the relationship between the change rate of cylinder volume and the relative displacement of the opposed piston can be considered.

As shown in Figure 4, the one-dimensional scavenging process is simulated by GT-Power. In the GT-Power model, the “Eng Cyl Comb SI Turb” quasi-dimensional model and the Woschni model are used to describe combustion and heat transfer, respectively. This software solves the one-dimensional time-dependent equation to predict the mass flow in intake and exhaust systems; a zero-dimensional model is used to calculate the combustion development, in-cylinder pressure, the thermal exchanges, and the indicated work. For the pressure losses, the three-dimensional effects in particular zones of the engine, such as intake ports and exhaust ports, are considered through appropriate discharge coefficients, defined as the ratio between effective and geometric flow areas.

The Eng Cyl Comb SI Turb quasi-dimensional two-zone combustion model is selected in GT-Power.

### Table 1: OP2S gasoline engine specifications

| Parameter                        | Value          |
|---------------------------------|----------------|
| Bore                            | 56 mm          |
| Stroke                          | 49.5 (x2) mm   |
| Connecting rod                  | 82.5 mm        |
| Effective compression ratio     | 10.5           |
| Number of intake ports          | 10             |
| Number of exhaust ports         | 10             |
| Intake port height stroke ratio | 0.121          |
| Exhaust port height stroke ratio| 0.141          |
| Intake port circumference ratio | 0.75           |
| Exhaust port circumference ratio| 0.6            |
| Opposed-piston phase difference | 15°CA          |
| Intake port radial angle        | 15°            |
| Exhaust port radial angle       | 0°             |
| Rated power                     | 15 kW @ 6000 rpm |
| Maximum torque                  | 28 N•m @ 3000 rpm |
| Fuel consumption rate           | 276 g/kW•h     |
software, which predicts the flame propagation rate, combustion rate, and flame development of the spark ignition engine. Before combustion begins, the entire mass in the cylinder is assumed to be in the unburned zone, which consists of the new combustible mixture for the cycle and the residual exhaust gas from the previous cycle. In the calculation, the energy conservation equations for the unburned and burned zones are solved separately in each time step. The energy equations of the unburned zone and the burned zone can be calculated using Equations (1) and (2).

\[
\frac{d(m_u e_u)}{d\varphi} = -p \frac{dV_u}{d\varphi} - Q_u - \left( \frac{dm_f}{d\varphi} h_f + \frac{dm_a}{d\varphi} h_a \right) + \frac{d}{d\varphi} m_{f,i} h_{f,i},
\]

(1)

\[
\frac{d(m_b e_b)}{d\varphi} = -p \frac{dV_b}{d\varphi} - Q_b + \left( \frac{dm_f}{d\varphi} h_f + \frac{dm_a}{d\varphi} h_a \right),
\]

(2)

where \( m_u \) is the working medium mass of the unburned zone, \( V_u \) is the volume of the unburned zone, \( m_f \) is the fuel mass, \( Q_u \) is the heat transfer capacity of the unburned zone, \( m_a \) is the air mass, \( h_f \) is the fuel mass enthalpy, \( m_{f,i} \) is the mass of the fuel injected, \( h_a \) is the air mass enthalpy, \( e_u \) is the working medium internal energy of the unburned zone, and \( h_{f,i} \) is the mass enthalpy of the fuel injected.

The OP2S gasoline engine knock simulation model is shown in Figure 5. This model is based on the gasoline engine knock theory and is combined with the engineering practice of FEV engine technology. A phenomenological model is used to characterize the detonation of the gasoline engine using “Eng Cyl Wall Detail” and “Eng Cyl T Wall Soln” to simulate the temperature changes on the cylinder wall surface. As shown in Figure 5A, the knock prediction model consists of a signal collector, a signal projector, a knock detection module, and a knock index output. The knock detection module is shown in Figure 5B.

As shown in the below equation, the auto-ignition induction time calculation can be used for deflagration analysis. The auto-ignition time is calculated by giving the relevant parameters of the mixture into such as pressure and temperature.

\[
T = \int_\text{IPC}^{\text{thkn}} \frac{1}{\tau} d\tau,
\]

(3)

where \( \tau \) is the induction time, IPC is the time when the intake ports is closed, thkn is the time of spontaneous combustion.

The induction time can be calculated by below equation.

\[
\tau = 5.72 e^{0.402 \left( \frac{\text{ON}}{100} \right)^{3.402} P^{-1.7} \exp \left( \frac{3800}{A T_u} \right)}.
\]

(4)
where $P$ and $A$ are the parameters related to the physical properties of the fuel, $ON$ is the cetane number of the fuel, $p$ is the in-cylinder pressure, and $T_u$ is the temperature of the unburned zone.

By combining Equations (3) and (4), the time of spontaneous combustion can be calculated and whether the end mixture has self-ignition detonation before the normal propagation of combustion flame can be determined. As shown in Equation (5), the knock intensity is calculated using the knock index (KI) model developed by GAMMA Company.34

$$KI = 10,000 \times A \times km \times \left( \frac{V_{IDC}}{V_I} \right) e^{\frac{T_a}{T_u}} \times \max\{0, (1 - (1 - \varphi)^2)\} \times \tau,$$

where $km$ is the mass fraction of the unburned mixture at the time of detonation, $V_I$ is the in-cylinder volume, $V_{IDC}$ is the cylinder volume at TDC, $T_a$ is the reaction activation temperature, $T_u$ is the transient temperature of the unburned mixture, $\tau$ is the induction time, and $\varphi$ is the crank angle.

### 2.2.2 Three-dimensional model of the thermodynamic process

AVL-Fire software is used to calculate the OP2S gasoline engine thermodynamic process, and the finite volume method is used to solve the hydrodynamic mesh and analyze the gas flow, fuel injection, and mixture formation. The FAME Engine Plus of AVL-Fire software is used to establish the moving mesh of the flow field for the OP2S gasoline engine. The global mesh scale is set to 2 mm to establish a basic hexahedral mesh. In the near-wall area, an unstructured mesh is used as shown in Figure 6. Since the flow near the intake and exhaust ports of the OP2S gasoline engine is complicated, localized mesh refinement is needed in this area. After several trial calculations, the mesh near the port is set to less than 1 mm, and the number of refinement layers is greater than 8. The reduction of the basic mesh scale and the degree of refinement have less influence on the calculation mass flow result. The number of mesh at the outer dead center is 284,000 and the number of mesh at the inner dead center is 62,000.

As shown in Figure 7, a three-dimensional CFD model of OP2S gasoline engine without injection and a combustion process is used to verify the mesh independence of mesh number by comparing the in-cylinder pressure of different mesh scales. The calculation results are less affected by the mesh number when the basic mesh size is less than 2 mm.
GT-Power software is used to simulate the one-dimensional thermodynamic process and verify the three-dimensional model. The Eng Cyl Comb SI Turb quasi-dimensional two-zone model is set as the combustion model, the Woschni GT model is set as the heat transfer model. The pressure and temperature in the cylinder and intake and exhaust ports can be simulated by the one-dimensional thermodynamic process. In the three-dimensional CFD model of the in-cylinder thermodynamic process, the initial calculation is performed when the exhaust ports open, and the temperature and the pressure in the cylinder are high. The initial conditions are shown in Table 2. The top surface of the opposed piston and the intake and exhaust ports are defined as moving walls; others are fixed walls. The temperature of the walls is determined empirically. The boundary conditions of intake port and exhaust port are the scavenging pressure and exhaust back pressure respectively.

In the three-dimensional CFD model of the OP2S gasoline engine, the ECFM model is used for the combustion process and the AnB Knock model is used for knock combustion. The AnB Knock model is used in conjunction with the ECFM combustion model to describe the reaction mechanism of the fuel through a two-step chemical reaction and calculate the generation process of a hypothetical substance that can predict knocking combustion phenomena. At the same time, the ECFM model is coupled with the Huh–Gosman spray model to complete the simulation of the spray combustion process.

For the AnB knock model, a hypothetical precursor is first calculated, and as the mass fraction of the precursor increases to equal that of the unburned substance, chemical oxidation is triggered. Therefore, when the precursor concentration exceeds the critical “concentration,” the fuel consumption due to spontaneous combustion is as follows:

$$\frac{dY_{fu}}{dt} = Y_{fu}A,$$

where \( A = 10^4 e^{-3500/T_{gb}} \), \( Y_{fu} \) is the fuel concentration, and \( T_{gb} \) is the localized temperature of the burned mixture.

The lad phase of knock combustion can be calculated by Equation (7).

$$\theta = A \left( \frac{RON}{100} \right)^P e^{\frac{\theta}{\rho_{fu}}},$$

where RON is the fuel octane number, \( P \) is the unburned mixture pressure, \( A, n, \) and \( B \) are the adjustment coefficients of the formula, and \( T_{fu} \) is the unburned mixture localized temperature.

Lafossas et al. shows that the precursor should be calculated by the exponential form. \( Y_P \) is defined as the precursor concentration, and its calculation before spontaneous combustion is as follows:

$$\frac{dY_{p}}{dt} = Y_{fu} F(\theta), \quad (8)$$

$$F(\theta) = \frac{\sqrt{\alpha^2 \theta^2 + 4(1 - \alpha \theta) \frac{Y_p}{Y_{fu}}}}{\theta}, \quad (9)$$

where \( \alpha \) is a constant and \( Y_{fu} \) is the fuel mass fraction.

When spontaneous combustion occurs, the mathematical expression of the precursor is as follows:

$$\frac{dY_{p}}{dt} = \beta Y_{fu} \left( \frac{3500}{T_{fu}} \right) \left( \frac{\rho}{\rho_{gl}} \times \frac{Y_{fu}}{Y_{fu}} \right), \quad (10)$$

TABLE 2 Boundary and initial conditions

| Computational start angle | 90°CA |
|---------------------------|-------|
| Computational end angle   | 450°CA|
| Scavenging pressure       | 0.12 MPa|
| Exhaust back pressure     | 0.1 MPa |
| In-cylinder initial pressure | 0.78 MPa |
| Intake chamber initial temperature | 322 K |
| Exhaust chamber initial temperature | 520 K |
| In-cylinder initial temperature | 646 K |
where $\beta$ is a constant, $T_{gb}$ is the localized temperature of the burned mixture, $\rho$ is the mixture density, and $\rho_{gf}$ is the fresh charge density.

2.2.3 | Model validation

After the fuel is injected into the cylinder, combined physical processes occur, which consist of fragmentation, turbulent disturbance, evaporation, and wall collision. The Huh–Gosman fragmentation model, the k-\(\varepsilon\)-f turbulence model, the Frolov evaporation model, and the Mundo Sommerfeld wall colliding model are chosen.\(^{37,38}\) Spray experiment conditions from Ma.\(^{37}\) are used in a nitrogen atmosphere at a temperature of 293 K and pressure of 0.1 MPa, and 93\# gasoline is injected at 13 MPa. The spray model is verified in a constant volume test chamber. As shown in Figure 8, comparison indicates that the spray patterns and penetration distances of both simulation and experiment are in good agreement.

The comparison between the simulation results of the thermodynamic process of the OP2S gasoline engine and the experimental results as well as the comparison of the in-cylinder cold flow pressure in the scavenging, compression, and expansion process of the OP2S gasoline engine three-dimensional CFD model and the in-cylinder pressure in the reverse process of the whole engine are shown in Figure 9. By comparing the simulation results of the thermodynamic process of the OP2S gasoline engine with the test results of the whole machine, verification of the one-dimensional thermodynamic process simulation model is completed. At the same time, the simulation results of the thermodynamic cycle process are used as the initial conditions and boundary conditions of the three-dimensional CFD calculation model, and the in-cylinder cold flow pressures of the scavenging process, compression, and expansion process of the OP2S gasoline engine three-dimensional CFD model are calculated and compared with those of the whole engine. The one-dimensional calculation results of the in-cylinder pressure are in good agreement with the experimental results of the operating condition, and the three-dimensional cold flow calculation results are in good agreement with the experimental results of the motored condition, which confirms the accuracy of the thermodynamic process simulation model of the OP2S gasoline engine and the three-dimensional CFD model in the scavenging process and the accuracy of flow calculations.

The parameters of the AnB knock model are shown in Table 3. The AnB knock model is combined with the verified combustion model to simulate the knocking combustion process of the OP2S gasoline engine. The comparison between the simulation result and experimental result is shown in Figure 10. From the comparison of the cylinder pressure curve, verification and validation of the knock model are clear.
3.1 Influence of compression ratio and ignition timing on knock combustion

The one-dimensional thermodynamic process simulation model of the OP2S gasoline engine is used in conjunction with the knock prediction model to analyze the influence of the compression ratio and ignition timing on knock combustion. The effect of different compression ratios on the engine performance is shown in Figure 11. Figure 11A shows that the knock intensity index increases with an increase of the compression ratio, and the knock tendency is pronounced. As the compression ratio increases, the temperature and pressure of the mixture in the combustion chamber increase at the time of ignition. When the mixture ignites, the propagation speed of the front surface of the flame is less than the auto-ignition speed of the mixture at the end of the combustion chamber in high temperature and pressure conditions, and knocking combustion occurs in the cylinder. At low engine speed, the turbulent combustion speed of the mixture in the cylinder is low, the propagation of the flame front surface is slow, so the end mixture has enough time to prepare for spontaneous combustion. With an increase in the speed, the propagation speed of the front surface of the flame increases, the auto-ignition preparation period of the terminal mixture is shortened, the risk of spontaneous combustion decreases, and the knock intensity index tends to decrease. As shown in Figure 11B,C, on reducing the knock intensity index by reducing the compression ratio, the engine power decreases, the fuel consumption rate increases, and the power performance and fuel economy become worse. Figure 11D shows that reducing the compression ratio reduces the expansion work and the thermal efficiency during the thermodynamic cycle, increasing the fuel consumption rate and increasing the exhaust gas temperature. When the compression ratio is set to 10.5, the power and fuel consumption meet the design targets, and the exhaust temperature can be controlled below 880°C.

The above research shows that on adjusting the compression ratio to reduce the knocking tendency, the power performance and fuel economy of the whole machine decrease, so it is necessary to adjust the ignition advance angle simultaneously to control the knocking. When the compression ratio is 10.5, the effect of different ignition advance angles on the engine performance is shown in Figure 12. Figure 12A shows that the increase in the ignition advance angle leads to earlier ignition of the mixture in the cylinder during the compression process, resulting in a rapid increase in the in-cylinder pressure. At the same time, the temperature and pressure of the terminal mixture increase, the knocking combustion tendency is pronounced, and the knock intensity index is increased. Figure 12B,C shows that with the increase of the ignition advance angle, the engine power decreases at low speed and increases at high speed. Meanwhile, the fuel consumption rate increases at low speed and decreases at high speed. At high speed, the crank angle range corresponding to the combustion process increases, and the ignition advance angle needs to be increased. On the contrary, the ignition advance angle needs to be reduced at a low speed. As shown in Figure 12D, when the ignition advance angle is set to 20°CA, the exhaust gas temperature can be controlled below 880°C. The power and fuel consumption meet the design targets.

Compression ratio and ignition timing are the main parameters affecting the performance of the OP2S gasoline engine. Increasing the compression ratio and increasing the ignition advance angle to improve the power performance and fuel economy will increase the tendency of knocking combustion. For the engine with a rated power condition, the power and fuel consumption rates corresponding to different compression
ratios and ignition advance angles are shown in Figure 13. On increasing the compression ratio to improve power and reduce fuel consumption, the different schemes correspond to the optimal values of power and fuel consumption when the ignition advance angle is 20°CA. When the compression ratio is 10.5 and the ignition advance angle is 20°CA, while controlling the knock intensity index and exhaust temperature, the power and fuel consumption meet the design targets.

3.2 Research on flame propagation and the knock response rate

The OP2S gasoline engine three-dimensional CFD model is used to analyze the flame propagation and the knock response rate of the combustion process for the two combustion chamber structures of flat and pit pistons. The combustion chamber of the OP2S gasoline engine is composed of the top surface of the opposed piston and the wall surface of the cylinder liner. The two types of combustion chambers of flat and pit pistons are shown in Figure 14.

Figure 15 shows the in-cylinder squish motion of the flat piston and the pit piston near the compression inner dead center (360°CA). For the flat piston scheme, the radial velocity of the airflow in the cylinder is defined as the squish flow velocity. For the pit piston scheme, the direction of the in-cylinder airflow into the piston pit is defined as the squish flow direction. For engines with rated power conditions, the radial squeezing effect of the flat piston on the airflow in the cylinder near the inner dead center is minimal, and the squish velocity tends to 0. The pit piston squeezes the airflow in the cylinder near the inner dead center. However, the effect is significant, and the squish flow velocity appears symmetric relative to the inner dead point.

In the calculation, the in-cylinder airflow movement of the longitudinal section of the injector centerline at the ignition time (340°CA) is determined, as shown in Figure 16. The results show that squish flow is generated by the design of piston dimples, and tumble flow is generated in the opposite piston dimples. The airflow...
movement of “tumble + squish” promotes turbulent flow in the cylinder at ignition time.

For the engine with rated power condition, comparison of the density of the front surface of the flame of the two combustion chamber structures of the flat and pit pistons is shown in Figure 17. After the mixture is ignited, the flame propagation in the cylinder is marked by the advance of the flame front surface towards the unburned area. After the flame is formed, it is not the concentration of the mixture that dominates the propagation direction of the surface of the flame front but the localized airflow. Due to the large reverse squeezing effect in the combustion chamber of the pit piston, the ignition center is formed near the spark plug. After the inner dead center, the opposite piston starts to move to the outer dead center, and the surface of the flame front quickly spreads outward due to the reverse squeezing action in the combustion chamber of the pit piston. At 365°C, the flame front surface propagation process of the pit piston combustion chamber is significantly earlier than that of the flat piston combustion chamber. At 372°C, the flame front surface of the pit piston combustion chamber has spread to the whole combustion chambers, the flame front surface is thick, and there is still an apparent unburned area on the edge of the flat piston combustion chamber.
For the engine with rated power conditions, a comparison of the knock response rate of the flat and pit pistons is shown in Figure 18. Combined with the comparison of the flame front surface density of the flat and pit piston, it is clear that the flame propagation speed of the flat piston combustion chamber is relatively slow. After the inner dead center, the end mixture is compressed and heated by the burned gas, which has released heat before the arrival of the normal flame, so increases the pressure and temperature of the combustion chamber wall and causes a knock reaction area; the terminal mixture appears to be spontaneously ignited.
At 360°CA, the flame front surface propagates as usual, and the end mixture does not spontaneously ignite. On extracting the results of the variation of the knock reaction rate, due to the slow propagation speed of the flame in the combustion chamber of the flat piston at 365°CA, the mixture on the wall of the combustion chamber has sufficient reaction time to cause spontaneous combustion, and the maximum knock reaction rate is 44.12. In the pit piston combustion chamber, no spontaneous combustion occurred. At 372°CA, due to the slit effect of the squish surface, the combustion chamber of the pit piston shows a slight knock, and the maximum knock response rate is 10.91. Therefore, the OP2S gasoline engine adopts a pit piston combustion chamber to promote flame propagation and avoid knocking effectively.

4 | CONCLUSION

(1) In the OP2S gasoline engine, the principle of equivalent volume change rate is used, and its thermodynamic process simulation model can be established by GT-Power software. At the same time, a quasi-dimensional dual-zone combustion model is used, and seven parameters of the thermodynamic process are collected to establish a knock prediction model, which can calculate the knock index.

(2) With the increase of the compression ratio and the ignition advance angle, the temperature and pressure of the mixture in the combustion chamber increase at the time of ignition, which leads to knocking combustion in the cylinder. By reducing the compression ratio and delaying the ignition, the knocking tendency of the OP2S gasoline engine can be reduced effectively. The compression ratio is 10.5, and the ignition advance angle is 20°CA.

(3) The OP2S gasoline engine combustion model adopts the ECFM model, coupled with the Huh–Gosman spray model, and is used in conjunction with the AnB Knock knocking model. The flame development after spontaneous combustion and the localized pressure increase in the combustion chamber can simulate knocking combustion.

(4) The OP2S gasoline engine uses the pit piston to organize squish flow and reverse squish flow near the inner dead center to promote flame propagation, and the knock response rate is 24.7% of that of the flat piston scheme. When the flat piston is used, the turbulent combustion rate is lower and the flame propagation distance is longer in the flame development period, which is not conducive to rapid combustion. In comparison with the pit piston combustion chamber, the flame propagation speed of the flat piston combustion chamber is relatively slow, which increases the knock tendency.

(5) The above conclusions are obtained by simulation calculations for the knocking combustion characteristics of an OP2S gasoline engine. With the continuous improvement of the principle prototype of the OP2S gasoline engine, a new ignition system and combustion system can be developed. In-cylinder end-gas auto-ignition experiments should be carried out to evaluate the knock combustion characteristics of the OP2S-GDI engine.

AUTHOR CONTRIBUTIONS
Fukang Ma and Jianwei Zhang designed the simulation model; Fang Wang and Fukang Ma performed the simulation calculations; Jianwei Zhang analyzed the data; and Fukang Ma and Fang Wang wrote the paper. All authors have read and agreed to the published version of the manuscript.

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CONFLICT OF INTEREST
The authors declare no conflict of interest.

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REFERENCES
1. Redon F, Kalebjian C, Kessler J. Meeting stringent 2025 emissions and fuel efficiency regulations with an opposed-piston, light-duty diesel engine. SAE 2014 World Congress & Exhibition. Society of Automotive Engineers; 2014.
2. Naik S, Johnson D, Koszewnik J, et al. Practical applications of opposed piston engine technology to reduce fuel consumption and emissions. SAE Paper 2013-01-2754.
3. Regner G, Johnson D, Koszewnik J, et al. Modernizing the opposed piston two stroke engine for clean, efficient transportation. SAE Paper 2013-26-0114.
4. Herold RE, Wahl MH, Regner G, et al. Thermodynamic benefits of opposed piston two stroke engines. SAE 2011 World Congress & Exhibition. Society of Automotive Engineers, 2011.
5. Zhen W, Liu JG. Intensive exploration of Ukraine T-84 main battle tank. Foreign Tank. 2011;44(11):11-20.
6. Hofbauer P. Opposed piston opposed cylinder (OPOC) engine for military ground vehicles. 2005 SAE World Congress. SAE International; 2005.
7. Pirault JP, Flint MLS. Opposed Piston Engines: Evolution, Use, and Future Applications. SAE International; 2009.

8. Cho S, Song C, Kim N, Oh S, Han D, Min K. Influence of the wall temperatures of the combustion chamber and intake ports on the charge temperature and knock characteristics in a spark-ignited engine. Appl Therm Eng. 2021;182(1):1-19.

9. Wei H, Chen C, Zhou H, Zhao W, Ren Z. Effect of turbulent mixing on the end gas autoignition of n-heptane/air mixtures under IC engine-relevant conditions. Combust Flame. 2016;174:25-36.

10. Juan PGM, Andres AA. Effect of the turbulence intensity on knocking tendency in a SI engine with high compression ratio using biogas and blends with natural gas propane and hydrogen. Int J Hydrogen Energy. 2019;44(33):18532-18544.

11. Chen L, Wei H, Chen C, Feng D, Zhou L, Pan J. Numerical investigations on the effects of turbulence intensity on knocking combustion in a downsized gasoline engine. Energy. 2019;166:318-325.

12. Chen L, Pan J, Wei H, Zhou L, Hua J. Numerical analysis of knocking characteristics and heat release under different turbulence intensities in a gasoline engine. Appl Therm Eng. 2019;159:113879.

13. Yu H, Qi C, Chen Z. Effects of flame propagation speed and chamber size on end-gas autoignition. Proc Combust Inst. 2017;36(3):3533-3541.

14. Livengood JC, Wu PC. Correlation of autoignition phenomena in internal combustion engines and rapid compression machines. Symp Combust. 1955;1:347-356.

15. Halstead MP, Kirsh LJ, Quinn CP. The autoignition of hydrocarbon fuels at high temperatures and pressures—fitting of a mathematical model. Combust Flame. 1977;30:45-60.

16. Heywood J. Internal Combustion Engine Fundamentals. Mc Graw Hill; 1988.

17. Lafossas FA, Castagne M, Dumas JP, et al. Development and validation of a knock model in spark-ignition engines using a CFD code. SAE Technical Paper no. 2002-01-2701; 2002.

18. Teraji A, Tsuda T, Noda T, Kubo M, Itoh T. Development of a three-dimensional knock simulation method incorporating a high-accuracy flame propagation model. Int J Engine Res. 2005;6(1):73-83.

19. Zhen X, Wang Y, Xu S, Zhu Y. Study of knock in a high compression ratio spark-ignition methanol engine by multidimensional simulation. Energy. 2013;50(1):150-159.

20. Pan J, Dong S, Wei H, Li T, Shu G, Zhou L. Temperature gradient induced detonation development inside and outside a hotspot for different fuels. Combust Flame. 2019;205:269-277.

21. Linse D, Kleemann A, Hasse C. Probability density function approach coupled with detailed chemical kinetics for the prediction of knock in turbocharged direct injection spark ignition engines. Combust Flame. 2014;161(4):997-1014.

22. Robert A, Richard S, Colin O, Martinez L, De Francqueville L. LES prediction and analysis of knocking combustion in a spark ignition engine. Proc Combust Inst. 2015;35(3):2941-2948.

23. Szważ S, Naber JD. Combustion of n-butanol in a spark-ignition IC engine. Fuel. 2010;89(7):1573-1582.

24. Pan J, Hu Z, Wei H, et al. Understanding strong knocking mechanism through high-strength optical rapid compression machines. Combust Flame. 2019;202:1-15.

25. Pan J, Shu G, Zhao P, Wei H, Chen Z. Interactions of flame propagation, auto-ignition and pressure wave during knocking combustion. Combust Flame. 2016;164(2):319-328.

26. Pan J, Hu Z, Wei H, Wang L, He Y, Wang X. Forced turbulence affected auto-ignition and combustion modes under engine-relevant conditions. Appl Energy Combust Sci. 2020;1-4:1-10.

27. Wei H, Feng D, Pan M, Pan J, Rao X, Gao D. Experimental investigation on the knocking combustion characteristics of n-butanol gasoline blends in a DISI engine. Appl Energy. 2016;175:346-355.

28. Qi Y, Wang Z, Wang J, et al. Effects of thermodynamic conditions on the end gas combustion mode associated with engine knock. Combust Flame. 2015;162(11):4119-4128.

29. Wang Z, Liu H, Reitz RD. Knocking combustion in spark-ignition engines. Progress in Energy & Combustion Science. 2017;61:78-112.

30. Ghanaati A, Muhamad Said MF, Mat Darus IZ. A comparative study on knock occurrence for different fuel octane number. SAE Technical Paper 2018-01-1674, 2018.

31. Forte C, Bianchi GM, Corti E, Fantoni S, Costa M. CFD methodology for the evaluation of knock of a PFI twin spark engine. Energy Proc. 2014;45:859-868.

32. Ma FK, Zhao CL, Zhang SL, et al. Scheme design and performance simulation of opposed-piston two-stroke gasoline direct injection engine. 2015 SAE Word Congress. SAE International; 2015.

33. Liu CZ, Lin GF, Hao YG, et al. Method discussion on performance simulation of opposed piston and opposed cylinder two-stroke engine. Diesel Engine. 2012;34(1):31-34.

34. Gamma Technologies. GT-Power v2020 Engine Performance Application Manual; 2020.

35. Feng HQ, Wei JN, Zhang J. Numerical analysis of knock combustion with methanol-isooctane blends in downsized SI engine. Fuel. 2019;236(9):394-403.

36. Gong CM, Sun JZ, Liu FH. Numerical research on combustion and emissions behaviors of a medium compression ratio direct-injection twin-spark plug synchronous ignition methanol engine under steady-state lean-burn conditions. Energy. 2021;215(10):1-16.

37. Ma FK. Study on the Combustion System of an Opposed Piston Two Stroke Gasoline Direct Injection Engine. Beijing Institute of Technology; 2016.

38. Yue ZY, Michele B, Sibendu S. Spray characterization for engine combustion network spray G injector using high-fidelity simulation with detailed injector geometry. Int J Engine Res. 2020;21(1):148-160.

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