Multi-objective Optimization of Intake Pipe and Valve Timing Based on Improved NSGA-II

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Abstract. The current research is based on a high-speed gasoline engine. The influence of intake pipe structural parameters and valve timing on engine’s performance was analysed by establishing thermodynamic model through software GT-Power. Full factorial experimental design was adopted and moving least squares method to establish response surface model at the full-load speed of 5500 r/min. Multi-objective optimization is carried out with genetic algorithm to solve Pareto frontier curve of economic and dynamic performance. The optimization effect by simulation was verified afterwards. The optimization can serve as guidance to determine valve timing of the fully variable valve and intake pipe. Optimization method proposed to this paper provides control parameters and theoretical basis for fully variable valve and intake pipe, reduces experimental workload and improves development efficiency.

1. Introduction
These guidelines, written in the style of a submission to J. Phys.: Conf. Ser., show the best layout for your paper using Microsoft Word. If you don’t wish to use the Word template provided, please use the following page setup measurements. Gas change process is the most important factor affecting the performance of the engine. Therefore, by properly designing the intake system, the dynamics of the gas can effectively improve the engine volumetric efficiency and the performance of the engine. The engine intake system mainly includes an intake pipe, an intake valve, an air cleaner, a throttle valve, and the like. Among them, the length, diameter of the intake pipe and cam phase have the most significant influence on the dynamic effect of the intake flow.

Professor Winterbourne and Dr. Parson of UMIST published the books “Theory of Engine Manifold Design: Wave Action Methods for IC Engines” [1] and “Design Techniques for Engine Manifolds Wave Action Methods for IC Engines” [2]. They summarized the design theory of intake manifold comprehensively. Sammut et al. [3] proposed an optimization of volumetric efficiency by variable length of intake manifold. The wave theory shows that the matching of the relevant parameters of the intake system is mainly related to the piston operating speed and the intake resonance characteristics. Taylor et al. [4] designed a two-stage variable length intake pipe. The results showed that the torque can be effectively increased by 45N·m and with fuel consumption reduced by 1%. The results of Murata et al. [5] show that the late intake valve closing along with the use of intake turbocharging, exhaust recirculation (EGR) and high-pressure fuel injection can reduce the emission of NOX and soot while ensuring the thermal efficiency of the diesel engine. AWANO et al. [6]...
installed late intake valve closing cam mechanism based on a single cylinder diesel engine. The research shows that it can greatly change the ignition delay period, realize the control of combustion phase and reduce the emission of NOx. HOMANN et al. [7] proposed an electromechanical valve (EMV) actuator, which uses the tracking controller composed of linear feedback and non-square iterative learning controller to reduce the noise and vibration during the valve opening and closing. However, few studies have focused on the integrated optimization of intake manifold and valve timing, and the application of response surface methodology and genetic algorithm is rare.

The research object is a high-speed gasoline engine. GT-Power is used to establish engine’s thermodynamic model and the influence of intake pipe structural parameters and valve timing on performance is analyzed. Taking the case of 5500r/min as an example, using the full factorial experiments design, the approximate response surface model is fitted by Moving Least Squares (MLS) method and the multi-objective optimization is performed by NSGA-2 algorithm to solve the Pareto frontier curve of BSFC and torque. After Analyzing the optimization results and choose a more reasonable solution. The simulation experiment verifies the optimization effect. Optimization combined response surface and genetic NSGA-2 has a guiding role in the determination of the fully variable valve and the structural parameters of the intake pipe.

2. Building GT-Power Model Analysis of Intake pipe and Valve Timing
Using GT-Power to establish the one-dimensional thermodynamic model based onto the basic data of a four-stroke high-speed gasoline engine. The model includes intake and exhaust pipes, air filters, throttles, fuel injectors, camshafts, cylinder and other modules and corresponding connecting holes. Since the optimization involves the adjustment of the structure of the intake pipe and the valve timing, so the combustion model adopted a predictable model SI-Turb. The temperature of the inlet, inner wall of the cylinder, the fuel injection pulse width and the air-fuel ratio were obtained from experiments. Using the WoschniGT and Chen-Flynn models to calculate heat transfer coefficient in the cylinder and the friction pressure loss through the pipes. In order to ensure the accuracy of the model, the predictability of the model needs to be verified. The bench testing was carried out on the original machine under the condition of full load Multi-objective Optimization of Intake Pipe and Valve Timing Based on Improved NSGA-II with engine speed ranging from 2000r/min to 9000r/min. The model is shown in Fig 1.

![GT-Power model map of the engine](image-url)
Fig 2 shows the comparison between the experimental value and the simulated value. The result indicates that engine’s full load speed characteristics curve of the simulation is highly consistent with the experiments. With maximum errors of BSFC (Brake specific Fuel Consumption), torque and power do not exceed 3%. The errors of other data such as backpressure, intake pipe pressure, NOx and PM emissions are all much less than 5%, which means the established model can be used as the basis of optimization research.

![Figure 2. Comparison of the engine’s full load speed characteristics curve](image)

**3. Analysis of Intake pipe and Valve Timing**

3.1. Intake pipe length and diameter

Gas flow in the intake process of internal combustion engine is a typical one-dimensional unsteady flow. In the downward process of piston, the pressure in cylinder drops rapidly. The expansion wave formed near the intake valve changes its phase when it transmits to the mouth of the intake pipe, reflecting the compression wave to push the air to intake valve. The length and diameter of intake pipe should be designed reasonably, so that the peak of compression wave reaches the intake valve when it closes. During specific engine speed range, the intake volume is enhanced, thus improving the volumetric efficiency and economic performance of the engine. The above dynamic effects can be described by Helmholtz resonance effect formula (1).

\[
N = \frac{K}{13.5} \sqrt{\frac{\pi D^2}{L V}} \sqrt{\frac{R - 1}{R + 1}}
\]

Fig 3 shows the pressure wave before the intake valve with different intake pipe length under 2500, 5500, 8500r/min. In the low and middle engine speed zone, both increasing the length of pipe and reducing the diameter of pipe properly can increase the amplitude, frequency of pressure wave in front of intake valve. Therefore enhancing the dynamic effect and improving the volumetric efficiency. While in the high-speed zone, longer intake pipes will increase the resistance of intake flow and suppress the dynamic effect. When the intake valve closes, the reflected compression wave is away from the wave peak position, so the short and coarse intake pipe is more suitable for the working condition at this time. The intake pipe mainly uses the amplitude and frequency of pressure wave to increase the mass flow rate for the early stage of intake stroke, but the drawback is that the phenomenon of intake backflow will be aggravated due to the pressure difference between incylinder and intake and exhaust ports in the later stage of intake stroke. It is noteworthy that the length of intake pipe needed to achieve Helmholtz theoretical resonance is longer at lower engine speed because of the low resonance frequency of single cylinder high-speed gasoline engine. Considering the reserved compartment for the engine and manufacturing processes, the restriction conditions of intake pipe length \( L \) and diameter \( D \) are as follows: \( 250 \leq L \leq 30 \); \( 50 \leq D \leq 25 \).
3.2. Intake valve open angle
Fig4 is a comparison of mass flow rate in front of intake valve under different intake advance angles. Fig5 is the relationship between BSFC, torque and intake advance angle. As shown in Fig4-5, when the intake valve opens before TDC (Top Dead Centre) and the exhaust valve closes after TDC, the valve overlap angle is formed, which will affect the scavenging process and the pump gas loss. Due to the valve overlap angle and compression stroke, the phenomenon of inlet backflow will occur, resulting in the increase of the residual gas coefficient in the cylinder and drop of volumetric efficiency and torque. The intake advance angle has little effect on the inlet backflow caused by compression stroke. Larger intake advance angle and valve overlap angle results inlet backflow occupying more proportion of the intake stroke and increasing inlet backflow volume. Opening of intake valve lags TDC can eliminate the phenomenon of backflow, but at the same time, it will shorten the intake stroke and enhance the inlet backflow caused by compression stroke. This not only is detrimental to the charging efficiency but also affects the dynamic performance of the original engine as well.

3.3. Intake valve close angle
Fig6 is a comparison chart of the mass flow rate of the intake under different LIVC (Late Intake Valve Closing) angle. Fig7 is the relationship between BSFC, torque and LIVC angle. It can be seen from figure6 that the LIVC angle of the intake valve has a more significant effect on the charging efficiency and dynamic performance. LIVC angle increases the intake volume mainly by utilizing the inertia effect and the crest of compression wave reflected from the dynamic effect. With the increase of the LIVC angle, the time-area value increases when the intake valve opens. The intake resistance caused by the closing of the intake valve is small, while the intake volume increases in the early stage of the intake stroke. However, the intake backflow caused by the compression stroke increases correspondingly, and the volumetric efficiency decreases. Reducing the LIVC angle can alleviate or even eliminate the intake backflow, but compared with the EIVC (Early Intake Valve Closing), the
intake volume decreases in the early stage of intake stroke. Besides, the intake resistance increases when the intake valve closes, which has a negative impact on the performance. Therefore, there is an optimal intake valve-closing angle, which can effectively utilize the dynamic effect of intake intake flow. Closing the intake valve when the in-cylinder pressure and intake backpressure are similar will restrain the intake backflow, and improve the volumetric efficiency. It can be seen from figure7 that BSFC and torque have opposite trend with the change of LIVC angle. There is an optimal value of LIVC angle, which makes the highest torque, while BSFC also maintains a low level.

3.4. Exhaust valve open angle
Fig8 is a comparison of the exhaust mass flow rate under different exhaust advance angles. Because the exhaust pressure in the cylinder is much higher than the exhaust port pressure at the early stage of the exhaust stroke, there is hardly exhaust backflow phenomenon. With exhaust valve opening after BDC (Bottom Dead Center), meaning extended expansion stroke, power has a tendency to increase. However, at the same time, due to large flow resistance of the exhaust caused by initial small throttle area of the exhaust valve, shortened exhaust stroke, and the high remaining pressure in the cylinder, loss of pumping gas continues to climb and the same is true with thermal loads of the piston, finally reverse the trend of power increasing. Normally, exhaust valve opens before BDC because exhaust can be expelled rapidly by using in-cylinder expansion and combustion pressure. This is conducive to reduce piston load, residual gas coefficient and gas pumping loss in cylinder during exhaust stroke. As a result, there exists an optimal exhaust valve open time, which is supposed to seek an optimal balance between high-pressure circulation and gas exchange. Fig9 shows the relationship between BSFC, torque and exhaust advance angle. BSFC and torque have the same trend with exhaust advance angle. The optimization of exhaust advance angle needs to be weighed and compromised.
3.5. Exhaust valve close angle

Fig 10 is a mass flow rate contrast chart of exhaust valve under different EVC (Exhaust Valve Closing) angle. It can be seen from figure 11 that the larger LEVC (Late Exhaust Valve closing) angle, the higher the pressure in exhaust port is than that of cylinder, which make exhaust backflow more obvious. Apart from it, the scavenging process will directly inhale some fresh air or combustible mixture into the exhaust pipe, causing negative impact on volumetric efficiency. EEVC (Early Exhaust Valve closing) can effectively avoid the phenomenon of exhaust backflow. However, in contrast with EEVC, the exhaust resistance and the work required for the piston to exhaust is reduced by LEVC angle. Besides, The inertia of the exhaust flow and the scavenging can continue to expel the exhaust in the cylinder and reduce the coefficient of residual gas in the cylinder, which is beneficial to improving the dynamic performance. The reasonable range of valve overlap angle can reduce the heat load of piston, improve the scavenging effect and reduce the exhaust temperature. Fig 11 shows the relationship between BSFC, torque and EVC angle. The effect of EVC angle on BSFC and torque is opposite to that of intake advance angle. It indicates that there is an optimal intake overlap angle, which can effectively improve the dynamic and economic performance.

4. Establishing response surface model

From the analysis above, it can be seen that intake pipe mainly affects the amplitude and frequency of fluctuation, and improves the performance by using the inertia effect of intake flow. Valve timing mainly improves the volumetric efficiency by restraining the intake and exhaust backflow, which reduces the resistance of flow and the residual gas coefficient in cylinder, therefore achieve the optimal balance between backflow and intake flow. Performance optimization needs to consider the combination of multiple factors. There is a complex coupling interaction between valve timing and intake pipe structural parameters, and the optimization process involves solving multiple performance objectives. It is a typical multi-objective optimization problem. The ultimate goal of optimization is to provide Pareto disassembly satisfying relevant constraints, so as to solve the problems of low matching degree of cam phase and large loss of pump gas. It can also improve the torque under different working conditions and reduce the BSFC, thermal load of piston.

In order to systematically research and utilize the relationship between intake and valve timing phase variables, DOE experiments were designed. Compared with the orthogonal experimental design, screening representative experimental cases, the full factor experimental design can comprehensively evaluate the main effects of parameters and all the coupling interactions among parameters, and can make full use of the range of values of each variable for calculation. Six factors were selected as DOE experimental factors: intake valve opening angle (IVO), intake valve closing angle (IVC), exhaust valve opening angle (EVO), exhaust valve closing angle (EVC), intake pipe length and diameter. Each experimental factor took several levels. Tab 1 shows detailed designs of the full factorial experiments. $\alpha, \beta, \gamma, \delta, L, D$ represents IVO, IVC, EVO, EVC, intake pipe length and diameter.

| parameters | unit | Low level | High level |
|------------|------|-----------|------------|

![Figure 10. Comparison of mass flow rate under different EVC angles](image-a)

![Figure 11. Relationship between BSFC, torque and EVC angle](image-b)
Response surface Methodology (RSM) is a method to construct a continuous approximate model of the mapping relationship between response factors and response values. The optimal response values can be predicted by using the method of interpolation search method. Because there are many independent variables of valve timing and intake pipe structural parameters, and the relationship between them is strongly non-linear, it is necessary to establish a model with curvature. Usually, the second-order model accuracy of engine intake system is enough. Compared with other quadratic methods, Moving Least Squares can change the fitting accuracy and the smoothness of response surface by choosing different weights and basis functions. Therefore, the second-order response surface models of BSFC and Tr are fitted by moving least square method.

Response Surface Formula:

\[
Y = \beta_0 + \sum_{i=1}^{n} \beta_i x_i + \sum_{i=1}^{n} \sum_{j=i+1}^{n} \beta_{ij} x_i x_j + \varepsilon \tag{2}
\]

From equation (2), \(\varepsilon\) represents error; \(x_i\) \& \(x_j\) are different response variables and \(\beta_0, \beta_i, \beta_{ij}\) are constant term, linear effect coefficient, second-order effect coefficient and interaction coefficient, respectively.

Fitting by Moving Least Squares:

\[
\begin{align*}
Y &= \sum_{i=1}^{n} p_i(x) \beta_i(x) = p^T(x) \beta(x) \\
p^T(x) &= [p_1(x), p_2(x), \ldots, p_n(x)]^T \\
\beta(x) &= [\beta_1(x), \beta_2(x), \ldots, \beta_n(x)]^T
\end{align*} \tag{3}
\]

From equation (3), \(\beta(x)\) is the vector of regression coefficients to be determined, \(p^T(x)\) is a set of bases in the polynomial space of the model, and \(m\) is the number of all base functions.

Determining the vector of regression coefficient based on the weighted square sum of errors \(J\):

\[
J = \sum_{i=1}^{n} w_i(x) [Y_i - \gamma_i]^2 = \sum_{i=1}^{n} w_i(x) [p^T(x_i) \beta(x) - \gamma_i]^2 \tag{4}
\]

From equation (4), \(n\) is the number of nodes in the region, \(f(x)\) is the fitting function, \(\gamma_i\) is the node value at \(x = x_i\), \(\gamma_i = y_i(x_i)\), \(w_i(x)\) is the compactly supported function, and \(\beta(x)\) is the vector of regression coefficients to be determined.

BSFC and Torque response surface models under 5500r/min are as follow:

\[
\begin{align*}
\text{BSFC} &= 199.87 - 0.1819x - 0.4174\beta - 0.3264\gamma + 0.1075L + 0.0054D + 0.166D + 0.003062a + 0.000729\beta^2 \\
&\quad + 0.006259\gamma^2 + 0.011976\delta^2 - 0.00534D^2 + 0.001968a\beta + 0.002373a\gamma + 0.001650a\delta - 0.000326aL + 0.003164aD \\
&\quad + 0.00117\beta\gamma + 0.001054\beta\delta + 0.00045\beta L + 0.005619\beta D - 0.002141a\gamma - 0.000372D - 0.001487\delta D - 0.000406LD \\
T_r &= 9.579 + 0.00746\alpha + 0.02257\beta - 0.00618\gamma - 0.03186\delta + 0.006767L - 0.0553D - 0.000311a^2 - 0.000659\beta^2 \\
&\quad + 0.000274\gamma^2 - 0.001650a^2 - 0.000009L^2 + 0.001045D^2 + 0.000079a\beta - 0.000327a\gamma - 0.00051a\delta + 0.000024a \\
&\quad - 0.000374aD - 0.000133a\gamma + 0.000083\beta\delta + 0.000035\beta L - 0.000056\beta D + 0.0000365a\delta + 0.000031aL \\
&\quad - 0.000133\gamma D + 0.000029D\delta + 0.000092LD
\end{align*} \tag{5}
\]

| parameteres | unit | Low level | High level |
|-------------|------|-----------|------------|
| \(a\)       | °    | 0         | 50         |
| \(\beta\)   | °    | 10        | 60         |
| \(\gamma\)  | °    | 10        | 60         |
| \(\delta\)  | °    | 0         | 50         |
| \(L\)       | mm   | 50        | 250        |
| \(D\)       | mm   | 25        | 45         |
$R^2$, $R^2_{adj}$ and $R^2_{pred}$ for response surface model BSFC are 0.9524, 0.9508 and 0.9489 while 0.9605, 0.9591 and 0.9579 for torque. The minimum of them is still higher than 0.94, and the difference between $R^2_{adj}$ and $R^2$ is far less than 0.2-0.3, which shows that the response surface model has a high degree of fitting the observed values and a strong predictive ability.

5. Tables Multi-objective Optimization With NSGA-2

The length and diameter of intake pipe, Cam Timing Angle and Angle Multiplier of cam profile in GT-Power simulation model are set as vector variables, and genetic optimization control is carried out by SIMULINK. According to the analysis of cam structure and flow simulation, the constraints on valve timing are determined. The optimum range of IVO is 0-50°, IVC is 10-60°, EVO is 10-60°, and EVC is 0-50°. At the same time, in order to ensure the drive efficiency and dynamic response speed of the optimized cams, the average and maximum velocity of the intake and exhaust valves are less than the corresponding original level, functioning as another constraint condition.

Mathematical model of multi-objective optimization model on power and economic performance of valve timing and intake pipe:

$$\begin{align*}
&\text{MAX} (\text{TORQUE}); \text{MIN} (\text{BSFC}); \\
\text{s.t.} & \quad 0 \leq \alpha \leq 50 \\
& \quad 10 \leq \beta \leq 60 \\
& \quad 10 \leq \gamma \leq 60 \\
& \quad 0 \leq \delta \leq 50 \\
& \quad 50 \leq L \leq 250 \\
& \quad 25 \leq D \leq 45 \\
& \quad V_{\text{MAX}} \leq V_{\text{MAX}}^0 \\
& \quad \bar{V} \leq \bar{V}_0
\end{align*}$$

From equation (7), $V_{\text{MAX}}$ is the maximum velocity of the optimized intake and exhaust cam; $\bar{V}$ is the average velocity of the optimized intake and exhaust cam; $V_{\text{MAX}}^0$ is the maximum velocity of the original intake and exhaust cam; $\bar{V}_0$ is the average velocity of the original intake and exhaust cam.

The improved non-dominated genetic algorithm (NSGA-2) is adopted to solve the multi-objective optimization problem. Compared with the general genetic algorithm and the NSGA algorithm, the concept of fast non-dominated sorting, cross-grade retention strategy and distributed congestion degree is innovatively employed to reduce the complexity of optimization time and make the optimization results uniformly distributed Pareto solution. The frontier of the set avoids the loss of best individuals in the optimization process. In this paper, Hammersley sampling is introduced based on NSGA-2 algorithm. Compared with Latin Hypercube, Hammersley has better sampling stability and uniformity on multi-dimensional problems. Therefore, Hammersley sampling is used to initialize and ensure the uniformity of individuals. At the same time, Hammersley sampling is invoked regularly based on the original cross-grade retention strategy. Hammersley sampling updates individuals with low fitness and balances the global search ability and convergence speed.

Set the initial population number to 100; genetic generations to 500; the crossover probability to 0.6; and the mutation probability to 0.1; update the population by Hammersley sampling every 25 generations; and the weight of BSFC and torque is 1:1. The Pareto frontier curve under the constraint condition of the solution results is shown in Fig12, with torque improved by 0-20.0% and BSFC reduced by 0-1.4% compared with the original engine. Torque decreases with the decrease of BSFC, and BSFC is restricted with the optimization of torque. The objective function and torque are contradictory, so the optimal value cannot be obtained at the same time. There is a trade-off relationship between the objectives. It is necessary to choose the optimal solution by compromising on the objectives.
To sum up, three groups of cases are selected in the Pareto front curve, as shown in Tab 2. Because the optimization process is based on response surface model, it is necessary to verify the actual optimization effect. BSFC and torque in the table show the rate of improvement compared with the original machine. When the engine speed is 5500 r/min, the optimal solution to the length and diameter of the intake pipe in the three cases is almost the same, and close to the limit value. This is because the optimal combination of length and diameter of the intake pipe in theory is not applicable to the actual situation because of the constraints of the compartment. The optimal solution of the EVC is also kept at 0°. The main reason is that the coupling relationship between the EVC angle and other valve timing angles is relatively weak. Compared with case 2, the torque improvement rate of case 3 increases slightly, but BSFC has not been optimized, but has increased on the basis of the original engine. Therefore, case 3 can be abandoned compared with case 1 and 2.

### Table 2. Design Schemes of 3 Cases Selected

| parameters | case 1 | case 2 | case 3 |
|------------|------|------|------|
| α/°       | 3.5  | 0    | 0    |
| β/°       | 19.4 | 15.6 | 14.5 |
| γ/°       | 34.7 | 46.2 | 53.5 |
| δ/°       | 0    | 0    | 0    |
| L/°       | 249.9| 249.9| 249.8|
| D/°       | 25   | 25   | 25   |
| BSFC/%    | -0.97| -0.42| 0.74 |
| Torque/%  | 7.79 | 12.67| 13.01|

### 6. Conclusion

Intake pipe mainly affects the amplitude and frequency of wave, and improves the performance by using the inertia effect of intake flow while valve timing mainly improves the charging efficiency by restraining the backflow, reducing the resistance of intake and exhaust flow, decreasing the residual exhaust coefficient in cylinder, and achieving the optimal balance of intake flow and backflow.

The second-order response surface model of valve timing and intake pipe structural parameters with respect to BSFC and torque is established by moving quadratic method. The minimum of $R^2$, $R^2_{adj}$ and $R^2_{pred}$ is larger than 0.95 which means a well regression fitting effect and the optimal combination of parameters can be predicted.

An improved NSGA-2 optimization algorithm is adopted to solve the response surface model, and finally a multi-objective Pareto front surface is obtained. Several schemes are selected to analyze the
optimization results, and solutions with practical engineering significance are chosen, which can guide the determination of structural parameters of fully variable valve and intake pipe.

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