Design and calibration of intake model for electric supercharged gasoline engine

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Abstract. The electric supercharger can significantly increase the transient intake air charge of the engine, thus increasing the power output of the engine under heavy load conditions. However, the traditional in-cylinder intake air charge calculation model has heavy calibration workload and low portability, and is not suitable for the intake air charge calculation of the electric supercharged gasoline engine. In order to reduce the difficulty of in-cylinder intake air calibration and improve the accuracy of intake air estimation, based on an electric supercharged gasoline engine, this paper simplifies the actual intake process into valve overlap period and free intake period, and proposes an in-cylinder intake air charge estimation model with intake pressure sensor as the main load sensor based on the mean value engine model. Three unknown coefficients, i.e., residual exhaust gas correction coefficient $\alpha_1$, heat transfer coefficient $\alpha_2$ between manifold and fresh gas, and effective flow area $\alpha_3$ in valve overlap period, are calibrated by using the simulation data of the numerical model of electric supercharged gasoline engine. Compared with the numerical simulation results, the air intake model established in this paper can accurately estimate the actual in-cylinder air intake of electric supercharged gasoline engine under the external characteristic condition, and the calculation error is within 5%, which can provide theoretical reference for the in-cylinder air intake estimation of electric supercharged gasoline engine.

1. Introduction

In order to ensure power performance and reduce emissions, modern gasoline engines often adopt three way catalytic converter (TWC) combined with air-fuel ratio closed-loop control technology. In order to ensure that TWC works in the highest efficiency area under various working conditions, it is necessary to accurately control the air-fuel ratio near the stoichiometric ratio [1-3], and accurate calculation of in-cylinder intake air quantity is the basis for realizing air-fuel ratio control. With the introduction of VVT, EGR and other technologies, the intake system of internal combustion engine has become more complex, and the original in-cylinder charge estimation method based on engine speed and manifold pressure is difficult to apply, so it is necessary to establish a calculation model of in-cylinder intake air quantity for the engine electronic control system.
The existing engine mainly adopts two methods to calculate the fresh charge into the cylinder, one is to measure the gas flow into the cylinder according to the intake mass flow sensor (MAF) upstream of the throttle valve, and calculate the actual intake air quantity in combination with the engine speed. This method has high calculation accuracy in steady-state conditions and is not affected by other factors (environment, engine aging) [4, 5]. However, under transient conditions, due to the dynamic characteristics of manifold filling and emptying, there is a big error between the measured value of in-cylinder air intake based on MAF and the actual value [6]. The other is to calculate the in-cylinder intake air quantity based on the "speed-density method" according to the volumetric efficiency of the intake manifold pressure sensor (MAP) and temperature sensor. This method has faster response and lower cost, and can more accurately estimate the in-cylinder transient intake air quantity. However, this method needs accurate volumetric efficiency data under various working conditions. The volumetric efficiency is usually expressed as a 3-dimension map with engine speed and intake manifold pressure as inputs, which is obtained by bench test calibration. With the increasing complexity of the intake system, the introduction of more variables greatly increases the difficulty of volumetric efficiency calibration, and at the same time, it also takes up the storage space of ECU [7].

Based on this, a large number of scholars have made extensive and in-depth research on the in-cylinder charge prediction model. Grizzle et al. [8] put forward a nonlinear open-loop in-cylinder charge estimator based on MAF, which regards the dynamic characteristics of the sensor as the first-order system delay. The experimental results show that this charge estimation model can significantly improve the prediction accuracy of transient intake air charge. However, in-cylinder charge calculation method based on MAF has high sensor cost, and the calculation load of open-loop estimator is heavy, which is inconvenient for ECU to calculate in real time. T. Leroy et al. [9, 10], taking the steady-state measured values of intake manifold pressure and temperature sensors as inputs, established a prediction model of in-cylinder charge based on physics, which divided the in-cylinder gas into fresh air and residual gas when the intake valve closed, and divided the residual gas into resident residual gas and backflow/scavenging residual gas. The experimental results show that this model can accurately predict the in-cylinder charge, but this method has a large calibration workload and is not portable among different engines. In addition, in order to reduce the calibration workload of volumetric efficiency, many scholars have proposed a volumetric efficiency calculation model based on artificial neural network (ANN) [11-13]. The experimental results of Wu [13] et al. show that the in-cylinder air intake prediction model based on ANN can accurately predict the in-cylinder air charge in the whole input range, but the in-cylinder charge calculation method based on ANN needs a lot of training data, and when the engine operating conditions are outside the training conditions, the accuracy of the model will greatly decrease. In addition, many scholars have studied in-cylinder charge prediction models based on cylinder pressure [14, 15]. Thomasson [14] et al. simplified the actual intake process by using cylinder pressure signal and intake and exhaust valve lift data, and established a calculation model of in-cylinder charge and residual exhaust gas fraction with only one regression coefficient. The steady-state bench test results show that the maximum error of intake air is 8.7%, but the cost of cylinder pressure sensor is too high, which is not convenient for practical application.

Electric turbocharger can significantly increase the transient intake air mass of engine, increase the power output of engine under heavy load, and effectively avoid "turbo lag", so it has received extensive attention in recent years. Honeywell, Caterpillar and Mitsubishi Heavy Industries have done a lot of research on electric turbocharger, and the results show that compared with traditional turbocharged gasoline engine, electric turbocharger can significantly increase the peak torque of engine [16-19], but there is little research on the intake model of electric turbocharged gasoline engine.

In view of the above analysis, based on the average model, this paper simplifies the actual intake process, and establishes an in-cylinder charge estimation model with the intake pressure sensor as the main load sensor for the electric supercharged gasoline engine. The intake process is subdivided into the overlapping period of intake and exhaust valves and the free intake period. Taking the working condition of external electric supercharger as the research condition, the unknown parameters in the intake model are calibrated by using the simulation results of GT-Power numerical model of electric
supercharged gasoline engine. Finally, the calculated value of in-cylinder air charge estimation model is compared with the simulation value. The results show that the calculation error of the model is less than 5%, which can be used for the in-cylinder air charge estimation of electric control system of electric supercharged gasoline engine.

2. Derivation of intake model

2.1. Calculation model of fresh air charge coefficient \( \phi_{\text{air}} \)

At the closing time of the intake valve, the gas in the cylinder is composed of fresh air and residual gas sucked in by the cycle intake stroke. Assuming that the two gases are not mixed in the cylinder, it can be considered that the pressures of fresh intake air and residual gas are \( P_{\text{air}} \) and \( P_r \) respectively, which can be obtained from the ideal gas state equation:

\[
m_{\text{air}} = \frac{P_{\text{air}}V_{\text{ivc}}}{RgT_{\text{air}}}
\]

(1)

In order to facilitate the real-time calculation of ECU, the actual intake mass is usually converted into the ratio of the actual intake mass to the intake mass when the cylinder is full under standard temperature and pressure, namely

\[
\phi_{\text{air}} = \frac{m_{\text{air}}}{m_0} = \frac{P_{\text{air}}V_{\text{ivc}}}{P_0V_0} = \frac{P_{\text{air}}V_{\text{ivc}}}{P_0V_0} \frac{T_0}{T_{\text{air}}}
\]

(2)

In which \( \phi_{\text{air}} \) is the fresh gas charge coefficient; \( m_0 \) is the gas mass when the cylinder is filled under the standard temperature and pressure; \( P_{\text{air}} \) is the pressure of fresh gas in the cylinder; \( P_0 \) and \( T_0 \) are pressure and temperature under standard temperature and pressure respectively, \( P_0 = 101325 \text{Pa} \) and \( T_0 = 273 \text{K} \); \( V_{\text{ivc}} \) is the cylinder volume at time IVC; \( V_0 \) is the cylinder volume when the piston is at bottom dead center; It can be calculated from the relevant formula in [20] according to the crank angle, where \( T_{\text{air}} \) is the temperature of fresh air in the cylinder at the closing time of the intake valve, and its value can be calculated from the gas temperature in the intake manifold, and the specific calculation process will be described in detail later.

Because it is difficult to directly measure the fresh gas pressure \( P_{\text{air}} \) in the cylinder, the total gas pressure in the cylinder at IVC time is approximately considered to be the same as the gas pressure in the manifold, which can be obtained by Dalton's partial pressure law combined with formula (2)

\[
\phi_{\text{air}} = \frac{m_{\text{air}}}{m_0} = \frac{P_{\text{air}}V_{\text{ivc}}}{P_0V_0} \frac{T_0}{T_{\text{air}}} = \frac{(P_{\text{im}} - P_r)}{P_0} \frac{V_{\text{ivc}}}{V_0} \frac{T_0}{T_{\text{air}}}
\]

(3)

Namely

\[
\phi_{\text{air}} = \frac{(P_{\text{im}} - P_r)}{P_0} \frac{V_{\text{ivc}}}{V_0} \frac{T_0}{T_{\text{air}}} \frac{1}{P_0} = \frac{(P_{\text{im}} - P_r)}{P_0} \cdot k_{\text{air}}
\]

(4)
In which $P_{im}$ is the intake manifold pressure and $P_r$ is the residual gas pressure. As mentioned above, $V_m \frac{V_0}{V_m} \frac{T_{exh}}{T_m} \frac{1}{P_0}$ can be calculated by relevant models. When the intake system of the engine estimates the intake air charge with the intake pressure signal as input, only $P_r$ in the equation is unknown, so the calculation process of residual exhaust gas pressure $P_r$ will be described in detail below.

2.2. Calculation model of residual exhaust gas pressure $P_r$

Due to the existence of the clearance volume of the internal combustion engine and the back flow/scavenging phenomenon caused by different pressures on the intake and exhaust sides during valve overlap, at the closing time of the intake valve (IVC), the in-cylinder gas is a mixture of fresh air and residual exhaust gas (or it may be completely fresh air, depending on the scavenging degree), and the mass of residual gas is usually calculated by the following formula [21]:

$$m_r = m_{IVO} + \int_{IVO}^{EVC} \dot{m}_{overlap} \, dt$$  \hspace{1cm} (5)

In which $m_r$ is the total residual gas mass in the cylinder at time $EVC$, $m_{IVO}$ is the residual exhaust gas mass at time $IVO$, and $\int_{IVO}^{EVC} \dot{m}_{overlap} \, dt$ is the mass change during valve overlap.

2.2.1. Calculation of residual gas quality in cylinder at IVO. From the equation of state of ideal gas

$$m_{IVO} = \frac{P_{IVO} \cdot V_{IVO}}{R \cdot T_{IVO}}$$  \hspace{1cm} (6)

In which $P_{IVO}$, $T_{IVO}$ and $V_{IVO}$ respectively represent the pressure, temperature and volume of gas in the cylinder at IVO time. Because it is difficult to obtain the in-cylinder gas state at $IVO$ time, the in-cylinder gas state at IVO time is approximately regarded as the same as the gas state of the exhaust manifold, and is appropriately corrected to obtain

$$m_{IVO} = \frac{P_{IVO} \cdot V_{IVO}}{R \cdot T_{IVO}} = \frac{P_{exh} \cdot V_{IVO}}{R \cdot T_{exh}} \times \alpha_i$$ \hspace{1cm} (7)

In the same way, the gas mass in the cylinder at IVO time is converted into the ratio of $m_{IVO}$ to the gas mass $m_0$ filled in the cylinder under the standard condition, which is obtained

$$\phi_{IVO} = \frac{m_{IVO}}{m_0} = \frac{P_{exh} \cdot V_{IVO} \cdot T_0}{P_0 \cdot V_0 \cdot T_{exh}} \times \alpha_i$$ \hspace{1cm} (8)

In which $\phi_{IVO}$ is the residual gas charge coefficient at IVO time; $m_{IVO}$ is the residual gas mass at IVO time; $P_{exh}$ and $T_{exh}$ are the pressure and temperature of exhaust manifold, respectively; $V_{IVO}$ is the cylinder volume at IVO time; $V_0$ is the cylinder volume when the piston is at the bottom dead center; $\alpha_i$ is the correction coefficient of residual gas at IVO time; It is considered that $\alpha_i$ is related to engine
speed and valve overlap angle, and the specific value of $\alpha_i$ can be calibrated according to the total residual gas coefficient, which will be explained in detail in the calibration section.

2.2.2. Calculation of mass change in valve overlap period. Generally, in order to make full use of the air inertia during the intake stroke, the intake valve is often opened before the top dead center. Similarly, in order to exhaust the exhaust gas as much as possible during the exhaust stroke, the exhaust valve is often closed after the top dead center. Therefore, there is an overlapping period of intake and exhaust valves near exhaust TDC. During the overlapping period, the intake pipe, cylinder and exhaust pipe are directly connected. The existence of valve overlapping angle is conducive to the realization of internal EGR (exhaust gas flowing back from exhaust manifold to cylinder) under small load, which can reduce combustion temperature and inhibit the generation of $NO_x$, thus reducing emissions [10]; Under heavy load, especially for supercharged engines, because the intake pressure is higher than the exhaust pressure, there is a scavenging phenomenon that fresh air flows directly into the exhaust pipe from the intake pipe, which is not only beneficial to cooling high-temperature components, but also beneficial to improving the low-speed performance of turbochargers [22, 23].

![Figure 1. Schematic diagram of gas flow during valve overlap period.](image)

In order to calculate the mass change of valve overlap period, the gas flow in valve overlap period is often regarded as a throttle valve model, and it is obtained

$$m_{overlap} = \int_{IVO}^{EVO} \dot{m}_{overlap} \, dt = \int_{IVO}^{EVO} \frac{1}{6N} \cdot A_{eff} \cdot \frac{2\gamma}{\gamma - 1} \cdot \frac{P_{up}}{R T_{sp}} \cdot \sqrt{\left(\frac{P_{down}}{P_{up}}\right)^{\gamma} - \left(\frac{P_{down}}{P_{up}}\right)^{\gamma+1}} \, d\theta \quad \text{others}$$

$$m_{overlap} = \int_{IVO}^{EVO} \frac{1}{6N} \cdot A_{eff} \cdot \frac{P_{up}}{R T_{sp}} \cdot \sqrt{\gamma \cdot \left(\frac{2}{\gamma + 1}\right)^{\gamma+1}} \, d\theta \leq \left(\frac{2}{\gamma + 1}\right)^{\gamma+1}$$

Where $m_{overlap}$ is the mass change during valve overlap; $IVO$ and $EVO$ represent the crank angle when the intake valve is opened and when the exhaust valve is closed; $N$ is the engine rotational speed; $\gamma$ is the specific heat ratio, taking 1.4; $R$ is the ideal gas constant, which is taken as $287 \, J/kg \cdot K$; $P_{up}$, $T_{sp}$ represents the upstream gas pressure and temperature respectively; $P_{down}$ is the downstream gas pressure, and the selection of upstream and downstream varies with the pressure of intake and exhaust manifold; $A_{eff}$ is the effective gas flow area during valve overlap period, which is related to valve overlap angle and intake and exhaust valve lift.
J. Fox et al. [21] put forward "overlap coefficient" to calculate $A_{\text{eff}}$, but it is necessary to calculate the integral value of intake and exhaust valve lift with rotation angle by this method, which has a large error and increases the calculation load of ECU. In addition, Equation (9) also adopts the form of integral, which is not convenient for ECU to calculate in real time, so Equation (9) is appropriately transformed.

$$m_{\text{overlap}} = m^*_{\text{overlap}} \times \frac{m_{\text{overlap}}}{m^*_{\text{overlap}}} = A_{\text{eff}} \cdot \frac{P_{\text{up}}}{\sqrt{R g T_{\text{up}}}} \cdot \sqrt{\gamma \cdot \left(\frac{2}{\gamma + 1}\right)^{\frac{1}{\gamma - 1}} \times \frac{m_{\text{overlap}}}{m^*_{\text{overlap}}}} \quad (10)$$

In which $m_{\text{overlap}}$ is the mass flow rate during valve overlap period and $m^*_{\text{overlap}}$ is the mass flow rate during critical gas flow. It can be seen from formula (10) that the critical flow mass flow rate is a function of upstream gas state and effective flow area, and has nothing to do with the upstream and downstream pressure ratio of intake and exhaust valves. To simplify the operation, the effective flow area $A_{\text{eff}}$ is expressed as a one-dimensional table with valve overlap angle as input, and it is obtained as follows:

$$m_{\text{overlap}} = \alpha_1 (EVC, IVO) \cdot \frac{P_{\text{up}}}{\sqrt{R g T_{\text{up}}}} \cdot \sqrt{\gamma \cdot \left(\frac{2}{\gamma + 1}\right)^{\frac{1}{\gamma - 1}} \times \frac{m_{\text{overlap}}}{m^*_{\text{overlap}}}} \quad (11)$$

After calibration, one can look up the table to get the effective flow area $A_{\text{eff}}$, and then calculate $m_{\text{overlap}}$. The specific calibration process will be described in detail later.

$$\frac{m_{\text{overlap}}}{m^*_{\text{overlap}}} = \begin{cases} \left(\frac{P_{\text{down}}}{P_{\text{up}}}\right)^{\frac{1}{\gamma}} \cdot \sqrt{\frac{2}{\gamma - 1} \cdot \left(\frac{\gamma + 1}{2}\right)^{\frac{\gamma + 1}{\gamma - 1}} \cdot \left[1 - \left(\frac{P_{\text{down}}}{P_{\text{up}}}\right)^{\frac{\gamma + 1}{\gamma - 1}}\right]} & \text{others} \\ 1 & \frac{P_{\text{down}}}{P_{\text{up}}} \leq \left(\frac{2}{\gamma + 1}\right)^{\frac{1}{\gamma - 1}} \end{cases} \quad (12)$$

It can be seen from equation (12) that $\frac{m_{\text{overlap}}}{m^*_{\text{overlap}}}$ is only a function of the upstream and downstream pressure ratio, which can be expressed as $\frac{m_{\text{overlap}}}{m^*_{\text{overlap}}} = f\left(\frac{P_{\text{down}}}{P_{\text{up}}}\right)$.

Therefore, the calculation model of mass flow during valve overlap can be obtained:

$$m_{\text{overlap}} = \alpha_1 (EVC, IVO) \cdot \frac{P_{\text{up}}}{\sqrt{R g T_{\text{up}}}} \cdot \sqrt{\gamma \cdot \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma + 1}{\gamma - 1}} \times f\left(\frac{P_{\text{down}}}{P_{\text{up}}}\right) \times \text{No}_{\text{cylinder}}} \quad (13)$$

In which EVC and IVO respectively represent the crank angle when the exhaust valve is closed and the intake valve is opened, $P_{\text{up}}$, $P_{\text{down}}$ respectively represents the upstream and downstream gas pressure, $T_{\text{up}}$ is the upstream gas temperature, $\text{No}_{\text{cylinder}}$ is the number of cylinders, and $\gamma$ is the adiabatic index, taking 1.4.
It can be seen from [24] that the relationship between mass flow and charge coefficient can be converted by the following formula:

$$m = \phi \times \frac{N \times V_o}{2578}$$  \hspace{1cm} (14)$$

Therefore, the variation of gas charge coefficient during valve overlap can be expressed as:

$$\phi_{\text{overlap}} = \alpha_3(EVC, IVO) \times \frac{P_{\text{up}}}{\sqrt{R_g T_{\text{up}}}} \times \sqrt{\gamma \cdot \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma + 1}{\gamma - 1}}} \times f \left(\frac{P_{\text{down}}}{P_{\text{sp}}}\right) \times N_0 \times \frac{2578}{N \times V_o} \sqrt{\gamma - 1}$$  \hspace{1cm} (15)$$

Where $\phi_{\text{overlap}}$ is the variation of gas charge coefficient during valve overlap, $N$ is the engine speed, and $V_o$ is the cylinder volume when the piston is at the bottom dead center, then the total residual gas charge coefficient in the cylinder at IVC time can be expressed as

$$\begin{cases}
\phi_{\text{residual}} = \phi_{\text{IVO}} + \phi_{\text{overlap}} & P_{\text{im}} < P_{\text{exh}} \\
\phi_{\text{residual}} = \phi_{\text{IVO}} - \phi_{\text{overlap}} & P_{\text{im}} > P_{\text{exh}}
\end{cases}$$ \hspace{1cm} (16)$$

Equation (16) shows that when the intake manifold pressure is less than the exhaust manifold pressure, the exhaust gas flows back into the cylinder, and when the intake manifold pressure is greater than the exhaust manifold pressure, fresh gas will flow into the exhaust manifold through the cylinder, resulting in scavenging.

By combining equations (8), (15) and (16), the calculation model of the total residual gas charge coefficient at the closing time of the intake valve can be obtained

$$\begin{cases}
\phi_{\text{residual}} = C_1 \times \alpha_1 + C_2 \times \alpha_3(EVC, IVO) & P_{\text{im}} < P_{\text{exh}} \\
\phi_{\text{residual}} = C_1 \times \alpha_1 - C_2 \times \alpha_3(EVC, IVO) & P_{\text{im}} > P_{\text{exh}}
\end{cases}$$ \hspace{1cm} (17)$$

Among $C_1 = \frac{P_{\text{exh}}}{P_0} \times \frac{V_{\text{IVO}}}{V_0} \times \frac{T_{\text{im}}}{T_{\text{exh}}}$, $C_2 = \frac{P_{\text{sp}}}{\sqrt{R_g T_{\text{sp}}}} \times \left(\gamma \frac{2}{\gamma + 1}\right)^{\frac{\gamma + 1}{\gamma - 1}} \times f(\frac{P_{\text{down}}}{P_{\text{sp}}}) \times N_0 \times \frac{2578}{N \times V_o}$, The value can be calculated directly according to the relevant input, $\alpha_1$ and $\alpha_3(EVC, IVO)$ are the coefficients to be calibrated in the model.

2.2.3. **Calculation model of residual gas temperature $T_r$.** When the residual gas coefficient $\phi_{\text{residual}}$ is obtained, in order to obtain the required residual gas pressure, it is necessary to calculate the residual gas temperature $T_r$ in the cylinder at time IVC. Assuming that the residual gas expands adiabatically, $T_r$ can be calculated according to the initial expansion temperature. Assuming that the residual exhaust gas and fresh air are never mixed in the cylinder, the residual gas temperature at the initial expansion time can be roughly estimated according to the upstream and downstream manifold pressures:

When $P_{\text{im}} < P_{\text{exh}}$, the exhaust gas flows back into the cylinder, and the residual gas temperature $T_{\text{ie}} = T_{\text{exh}}$ at the beginning of expansion; When $P_{\text{im}} > P_{\text{exh}}$, fresh gas flows into the exhaust side from the cylinder; Assuming that fresh air enters the cylinder, the residual exhaust gas is pushed out first, and then flows directly from the intake side to the exhaust side; Then when $\phi_{\text{residual}} > 0$, the gas composition in the cylinder is a mixture of residual waste gas and fresh air, which still meets $T_{\text{ie}} = T_{\text{exh}}$, and when...
\( \phi_{\text{residual}} < 0 \), the exhaust gas in the cylinder is completely swept away, and all the gas composition in the cylinder is fresh air, without residual gas expansion process.

In which \( T_{\text{exh}} \) is the exhaust manifold airflow temperature, assuming that the initial expansion pressure is \( P_{\text{exh}} \) and the temperature is \( T_1 \), which can be obtained from the ideal gas state equation combined with the gas adiabatic expansion equation

\[
\frac{T_e}{T_1} = \left( \frac{V_{\text{EVC}}}{V_{\text{IVC}}} \right)^{y-1} \tag{18}
\]

According to the calculation model of fresh air charge coefficient

\[
\phi_{\text{air}} = \frac{P_{\text{air}} \cdot V_{\text{IVC}}}{R_g \cdot T_{\text{air}}} \cdot \frac{1}{m_0} = P_{\text{air}} \cdot k_{\text{air}} \tag{19}
\]

In like manner; in a similar way

\[
\phi_{\text{residual}} = \frac{P_r \cdot V_{\text{IVC}}}{R_g \cdot T_r} \cdot \frac{1}{m_0} = P_r \cdot k_r \tag{20}
\]

Then

\[
\frac{\phi_{\text{residual}}}{\phi_{\text{air}}} = \frac{P_r \cdot T_{\text{air}}}{P_{\text{air}} \cdot T_r} = \frac{P_r \times k_r}{P_{\text{air}} \times k_{\text{air}}} \tag{21}
\]

Therefore

\[
\frac{k_r}{k_{\text{air}}} = \frac{T_{\text{air}}}{T_r} \tag{22}
\]

\[
P_r = \frac{\phi_{\text{residual}}}{k_r} = \frac{\phi_{\text{residual}}}{k_{\text{air}}} \cdot \frac{T_r}{T_{\text{air}}} \tag{23}
\]

2.3. Calculation model of fresh gas temperature \( T_{\text{air}} \) in cylinder

It can be seen from formula (4) and formula (23) that both the calculation model of fresh gas charge coefficient and the calculation model of residual gas pressure \( P_r \) need the fresh gas temperature \( T_{\text{air}} \) in the cylinder, and \( T_{\text{air}} \) is difficult to be measured directly by the sensor during the actual operation of the engine, so it is often calculated according to the measured value \( T_{\text{in}} \) of the air flow temperature of the intake manifold.

In the actual intake process, the fresh gas will exchange heat with manifold, intake valve, cylinder and other components, resulting in temperature change. Because the actual heat exchange process is complex, the actual process is often simplified, and the heating of gas by each component is simplified as manifold heating and intake valve and cylinder heating. The inlet airflow temperature can be calculated by the following formula:

\[
T_{\text{Impl}} = (T_{\text{wall}} - T_{\text{in}}) \times \alpha_s (N, P_{\text{in}}) + T_{\text{in}} \tag{24}
\]
\(T_{\text{import}}\) is inlet flow temperature, \(T_{\text{wall}}\) is inlet manifold wall temperature, replaced by engine water temperature, \(T_{\text{im}}\) is inlet manifold air flow temperature, \(N\) is engine speed, \(P_{\text{im}}\) is inlet manifold pressure, \(\alpha_2(N, P_{\text{im}})\) is heat transfer coefficient between inlet manifold wall and fresh intake air, which is related to engine speed and load. For convenience of calculation, \(\alpha_2\) is set as 3-dimension map with input of engine speed and inlet manifold pressure, which will be used after calibration. The specific calibration process is in the fourth step.

After obtaining the inlet airflow temperature, the calculation model of the fresh air temperature \(T_{\text{air}}\) in the cylinder is as follows:

\[
T_{\text{air}} = (T_{\text{wall}} - T_{\text{import}}) \times [1 - e^{\left(-\frac{1200}{8}\right)}] + T_{\text{wall}}
\]  

(25)

The formula shows that the fresh air will not only be heated by the intake manifold, but also exchange heat with the valve and cylinder wall. At this point, combining the calculation model of fresh air charge coefficient \(\phi_{\text{air}}\), residual gas pressure \(P_r\), and in-cylinder fresh gas temperature \(T_{\text{air}}\), the final calculation model of in-cylinder intake air quantity can be obtained.

In order to obtain the relevant 2-dimension and 3-dimension map in the intake model, and to study whether the intake model can accurately estimate the actual intake air mass of an electrically supercharged gasoline engine under various operating conditions, and reduce the actual calibration workload, this paper based on the one-dimensional full-load numerical model of a turbocharged gasoline engine, and verified the numerical model with multi-parameters by bench test data. On this basis, the numerical model was extended to the form of electrically supercharged gasoline engine, and the relevant map in the intake model was calibrated by using the numerical model of electrically supercharged direct injection gasoline engine. The specific process is as follows.

3. Establishment and calibration of performance simulation model of electric supercharged gasoline engine

The engine used in this paper is a 1.5-liter direct injection supercharged gasoline engine, and the main features of the engine are listed in Table 1.

| Table 1. Performance parameters of direct injection turbocharged gasoline engine. |
|---------------------------------|---------------------------------|
| Name                           | Performance parameter          |
| Engine Type                     | Four cylinder inline           |
| Cylinder diameter /mm           | 76 mm                          |
| Travel /mm                      | 82.6 mm                        |
| Compression ratio               | 10                             |
| Displacement /L                 | 1.5 mm                         |
| Length of connecting rod /mm    | 133.2 mm                       |
| Rated power /KW                 | 110(5200r/min)                 |
| Maximum torque/(n m)            | 220(2000-4400r/min)            |
| Injection type                  | Direct Injection               |
| Gas distribution type           | Variable valve timing          |

The actual engine is a combination of several subsystems, including intake system, engine block, exhaust system, etc. According to the actual geometric structure, working principle and experimental data of the direct injection supercharged gasoline engine, the physical model of related subsystems is established by using GT-Power. The flow coefficient measured by the steady flow experiment of the airway is used to simulate the gas flow in the intake and exhaust pipes and cylinders; The combustion process in cylinder is calculated by Weibe model, and the heat transfer in cylinder is calculated by Woschni model; The turbocharger model is established by using map data of turbine and compressor,
and the friction model is established by using friction loss data collected by reverse towing experiment, and the final thermodynamic simulation model of turbocharged engine is obtained. Using bench test data, the thermodynamic model is calibrated with multiple parameters. Fig. 2 shows the comparison between simulated and experimental values of circulating intake air mass, effective power, torque and specific fuel consumption in full-load cycle, and fig. 3 shows the comparison between simulated and experimental values of circulating intake air mass, effective power, torque and specific fuel consumption in part-load cycle at engine speed of 3600 rpm. The results show that the simulated values of various characteristic parameters are basically consistent with the experimental values under various working conditions, and the maximum deviation is less than 5%. Therefore, the numerical model can characterize the real engine to a certain extent and can be further studied on this basis.

**Figure 2.** Simulation and experimental comparison of air intake, effective power, torque and specific fuel consumption in full-load cycle.

**Figure 3.** Simulation and experimental comparison of air intake, effective power, torque and specific fuel consumption in 3600 rpm cycle.
In order to calibrate the unknown parameters in the intake model and verify whether the established intake model can accurately estimate the actual in-cylinder air mass of the electric supercharged gasoline engine under various operating conditions, and to reduce the calibration time and cost, this paper transforms it into a numerical model of the electric supercharged gasoline engine based on the numerical model of the turbocharged gasoline engine.

GT-Power has no special electric supercharger module, but Speed Boundary Rot module can simulate self-powered general equipment such as motor. Therefore, in this study, the Speed Boundary Rot module and the compressor module are combined to establish the electric supercharger model. To simplify the calculation, it is assumed that the transmission efficiency of the motor and the transmission efficiency between the motor and the compressor are both 1 in the GT-Power model, which means that the power consumption of the electric supercharger is the same as that of the compressor. The control principle of the electric supercharger is to control the motor speed so that the electric supercharger is in the high efficiency zone at each operating point of the engine, and adjust the efficiency of the inter cooler to ensure that the intake temperature of the electric supercharged gasoline engine is the same as that of the original engine under the same operating conditions. The complete numerical model of electric supercharged gasoline engine is shown in Figure 4.

![Figure 4. GT-Power numerical model of electric supercharged gasoline engine.](image)

4. Numerical calibration and verification of intake model

Coefficients to be calibrated include: residual gas correction coefficient $\alpha_1$ at IVO time, heat exchange coefficient $\alpha_2$ between intake manifold wall and fresh gas, and effective gas flow area $\alpha_3 (EVC, IVO)$ during valve overlap. Considering that the actual working conditions of electric supercharger are mostly starting working conditions and heavy load working conditions, this paper only selects full load working conditions as data acquisition points. The engine speed covered by data acquisition working conditions ranges from 1200rpm to 5600 rpm with a step size of 400rpm, the intake advance angle ranges from 10deg before TDC to 40deg before TDC with a step size of 10deg, and the exhaust late closing angle ranges from 10 deg after TDC to 40deg after TDC with a step size of 10 deg. The collected operating points are shown in Table 2.
Table 2. Collect data of working point.

| Data name                          | Unit   | Acquisition range  | Step length |
|------------------------------------|--------|--------------------|-------------|
| Engine speed                       | rpm    | 1200-5600          | 400         |
| Intake advance angle               | deg    | 10-40              | 10          |
| Exhaust late closing angle         | deg    | 10-40              | 10          |
| Rotating speed of electric supercharger | rpm  | 100000-200000      | 10000       |

4.1. Calibration of residual gas correction coefficient $\alpha_i$ at IVO time and effective flow area $\alpha_3$ during valve overlap period

It can be seen from equation (17) that in the calculation model of total residual gas charge coefficient in cylinder, $\alpha_i$ and $\alpha_3(EVC, IVO)$ are unknown coefficients of two terms on the right side of the equation respectively. It can be seen from section 2.2.2 that equation (17) can be regarded as a binary linear equation containing two unknown coefficients, so the residual gas charge coefficient can be calculated from the residual gas quality data collected under various working conditions, and then the values of $\alpha_i$ and $\alpha_3(EVC, IVO)$ can be obtained by using the least square linear regression method.

The specific calibration method is as follows:

Firstly, the intake and exhaust valve timing is fixed, the valve overlap angle is guaranteed to be 20 degrees unchanged, and the engine speed is fixed at 1200 rpm. Then, the exhaust gas mass scatter data at each electric supercharger speed is collected, and the residual gas charge coefficient is calculated. Then using the least square linear regression, we can get the values of $\alpha_i$ and $\alpha_3(EVC, IVO)$ at this engine speed and this valve overlap angle, repeat the above steps until all valve timing and engine speed are traversed, and all $\alpha_i$ and $\alpha_3(EVC, IVO)$ calibration results within the research working condition range can be obtained. The $\alpha_i$ and $\alpha_3(EVC, IVO)$ calibration results are shown in Fig. 5 and Fig. 6 respectively.

![Figure 5](image)

**Figure 5.** Correction coefficient $\alpha_i$ of residual waste gas at IVO time.
4.2. Calibration of heat transfer coefficient $\alpha_2$ between intake manifold wall and fresh gas

In order to make the fresh air temperature in the cylinder as close as possible to the real value when the intake valve is closed, it is usually necessary to calibrate the heat transfer coefficient $\alpha_2(N, P_{in})$ between the intake manifold wall and the fresh air. During calibration, the intake air flow temperature $T_{Intport}$, the intake manifold air flow temperature $T_{im}$, and the intake manifold wall temperature $T_{wall}$ are collected, and $\alpha_2(N, P_{in})$ can be calculated by equation (24). The calibrated heat transfer coefficient between the manifold wall and the fresh air is shown in Figure 7.
4.3. Validation of intake model
The Matlab/Simulink simulation tool is used to model the in-cylinder charge calculation process of the above-mentioned electric supercharged gasoline engine, and the calibrated and complete data of residual gas correction coefficient $\alpha_1$, heat transfer coefficient $\alpha_2$ between intake manifold wall and fresh gas, and effective gas flow area $\alpha_3$ during valve overlap are input into the Simulink model in the form of map and simulated offline. In order to study whether the intake model can accurately estimate the in-cylinder intake air mass under various operating conditions, this paper compares the calculation results of the intake model with the GT-Power numerical simulation results at different valve overlap angles at high, medium and low engine rotation speeds, as shown in Figure 8-10.

**Figure 8.** Comparison between calculated value and simulation value of fresh air charge coefficient model in 2400 rpm cylinder.

**Figure 9.** Comparison between calculated value and simulation value of fresh air charge coefficient model in 4000 rpm cylinder.
Figure 10. Comparison between calculated value and simulation value of fresh air charge coefficient model in 5600 rpm cylinder.

The results show that the calculated value of the in-cylinder fresh gas charge coefficient model is very close to the GT-Power simulation value at various engine rotational speeds and valve overlap angles, and the maximum error is within 5% except for a few points. Therefore, the physical-based in-cylinder intake air calculation model established in this paper can be used to accurately estimate the steady-state intake air of the electric supercharged gasoline engine. Further analysis shows that the intake manifold pressure is approximately linear with the in-cylinder fresh gas charge coefficient at each engine speed. At lower engine rotational speeds (such as 2400rpm and 3200rpm), the intake air mass increases with the increase of valve overlap angle, which is mainly because there are relatively more residual gases in the cylinder at lower engine speeds, and the scavenging effect is increased after the valve overlap angle increases, thus reducing the residual gas coefficient and allowing more fresh air to enter the cylinder. Moreover, the inertia of air flow becomes more obvious after the valve overlap angle increases, thus increasing the intake air mass in the cylinder. However, at high engine rotational speed (e.g., 5600rpm), the increase of valve overlap angle has no obvious effect on the increase of in-cylinder intake air mass, which is due to the fact that the actual duration of valve overlap period becomes shorter with the increase of engine rotational speed, the intake inertia cannot be fully utilized, and the residual gas coefficient itself is small, and the fresh air at the intake side directly flows into the exhaust side. In addition, when the fixed intake manifold pressure and valve overlap angle are unchanged and the engine rotational speed is gradually increased, the actual intake air mass in the cylinder gradually decreases with the increase of engine rotational speed, as shown in fig. 11, which is due to the decrease of intake air density due to the increase of intake air temperature after the increase of engine rotational speed.
5. Summary
In this paper, based on the mean value engine model, an in-cylinder intake air estimation model with intake manifold pressure sensor as the main load sensor is established for electric supercharged gasoline engines, and the unknown parameters in the intake air charge estimation model are calibrated by using the intake air mass simulation results generated by the numerical model of electric supercharged gasoline engines.

(1) The actual intake process is simplified, and the in-cylinder gas is divided into fresh air and residual gas. Based on the ideal gas state equation and one-dimensional compressible fluid orifice flow equation, a calculation model of in-cylinder intake air mass is established with the intake manifold pressure as the main input.

(2) Comparing the calculated value of the in-cylinder air intake model with the simulation value of the numerical model under various working conditions, the results show that the intake model can accurately estimate the actual in-cylinder air intake of the electric supercharger under various working conditions, and the calculation error is within 5%, which can provide theoretical reference for the estimation of the in-cylinder air intake of the electric supercharged gasoline engine.

(3) Simplify the calculation process of residual gas, replace the in-cylinder gas state at IVO time with the gas state of exhaust manifold, put forward the correction coefficient \( \alpha_1 \) of residual gas at IVO time, consider the heat transfer of manifold and cylinder to fresh gas, put forward the heat transfer coefficient \( \alpha_2 \) of manifold to gas, simplify the calculation process of gas effective flow area \( A_{ef} \) during valve overlap period, put forward the effective flow area coefficient \( \alpha_3(EVC,IVO) \), and calibrate the three parameters by numerical model, which simplifies the calibration workload and saves the cost.

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