Application of Space-time Conservation Element and Solution Element Method in Intake and Exhaust Flows of High Power Density Diesel Engine

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Abstract

A one-dimensional pipe flow model of single-cylinder diesel engine is established to investigate the intake and exhaust flow characteristics of diesel engine in the condition of high power density (HPD). A space-time conservation element and solution element (CE/SE) method is used to derive the discrete equations of the partial differential equation for the intake and exhaust systems. The performance parameters of diesel engine with speed of 2100 r/min are simulated. The simulated results are in accordance with the experimental data. The effect of increased power density on charging coefficient is analyzed using a validated model. The results show that the charging coefficient is slowly improved with the increase in intake pressure, and is obviously reduced with the increase in engine speed.

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1. Introduction

As the global demand for energy conservation and emission reduction is getting more and more attention, the development trend of diesel engine is to reduce its displacement and improve its power density\cite{1-5}. The basis characteristics of HPD diesel engine are high speed and high boost pressure, which makes the research on exhaust process of HPD diesel engine different from the traditional research. The gas flow states of intake and exhaust systems have great influence on the power, fuel economy and emission performances of engine. The research on engine intake and exhaust flow has been always a project of particular interest to the researchers\cite{6}.

Currently, the finite volume method is a main numerical method for simulating the intake and exhaust systems of diesel engine \cite{7-10}.

The CE/SE method is a new numerical method, which was originally proposed by S. C. Chang from NASA Lewis Research Center in the 1990s. It treats the space and time fluxes uniformly, establishes the conservation and solution elements based on the principle of conservation of space flux and time flux, and ensures the conservation of the format in local and global. It needs no other numerical approximation technique except simple Taylor expansion, and does not need to apply any monotonicity restriction or characteristic computing technology. Therefore, the method has the advantages of simple structure, convenient calculation and high accuracy of shock wave-capturing calculation, etc \cite{11-16}.

In the paper, the CE/SE method is used to simulate the intake and exhaust processes of a HPD single-cylinder diesel engine, test and verify the applicability of the method in the one-dimensional unsteady flow simulation, and provide the reliable bases for the design and experiment of engine intake and exhaust systems in HPD diesel engine.
2. Governing equations and algorithm

2.1. Governing equations

Under the conditions of variable section, wall friction, heat transfer or combustion and gradually changing entropy, the variation of parameters when fluid flows through a pipe with length of $dx$ is shown in Fig. 1.

In Fig. 1, $x$ is the inflow section; $u$, $\rho$, $p$ and $F$ are the flow velocity, density, pressure and section area, respectively. The fundamental equation of unsteady flow in the intake and exhaust pipes of internal combustion engine is derived according to the law of conservation of mass, the momentum theorem, and the law of conservation of energy.

By introducing state vector $w$, flow vector $f$ and source vector $s$, the above fundamental equation can be rewritten to a uniform conservation equation

$$\nabla \cdot w(x,t) + \frac{\partial f(w)}{\partial x} + s(w) = 0$$

where $w(x,t) = \begin{pmatrix} \rho \\ \rho u \\ \rho e_0 \end{pmatrix}$, $f(w) = \begin{pmatrix} \rho u \\ \rho u^2 + p \\ \rho u h_0 \end{pmatrix}$,

$$s(w) = c_1(w) + c_2(w)$$

$$= \begin{pmatrix} \frac{\mu u}{} \\ \frac{\mu u^2}{} \\ \frac{\rho u h_0}{} \end{pmatrix} \frac{1}{\hat{F}} \frac{dF}{dx} + \begin{pmatrix} 0 \\ \rho G \\ -pq \end{pmatrix}$$

where $c_1(w)$ is a term caused by variation of delivery pipe section, and $c_2(w)$ is a term caused by friction and heat transfer.

2.2. Derivation of discrete equation based on CE/SE method

Let $x_1 = x$ and $x_2 = t$ represent two coordinates in two-dimensional Euclidian space $E_2$. By introducing vector $h_m = (f_m, w_m)$, $m = 1, 2, 3$, and divergence operator $\nabla = (\partial / \partial x, \partial / \partial t)$, the governing equation can be rewritten to

$$\nabla \cdot h_m + s_m = 0, \quad m = 1, 2, 3$$

According to Gauss divergence theorem, the integral form of Eq. (3) is

$$\mathcal{N} \int_{S(V)} h_m \, ds + \int \mathcal{V} \, s_m \, dV = 0, \quad m = 1, 2, 3$$

where $S(V)$ is the closed surface of any area $V$ in space $E_2$. In Eq. (3), obviously the space and time are unified to treat.

In the staggered grid shown in Fig. 2(a), there is a corresponding CE/SE for every grid point (e.g. A, C and F). A solution is approximately given in CE, satisfying the conservation law in SE. As shown in Fig. 2(b), SE($j$, $n + 1/2$) corresponding to the grid point A($x_j$, $t_{n+1/2}$) is a diamond area BDFG surrounded by dashed lines, while CE($j$, $n + 1/2$) is a rectangle area BCEF surrounded by solid lines.

For any point ($x$, $t$) $\in$ SE($j$, $t_{n+1/2}$), $w_m$ and $f_m$ can be approximately calculated by first order Taylor expansion, and the source item $s_m$ can be approximately calculated by zero order Taylor expansion. In this way, if $w_{int}$, $f_{int}$, $f_{max}$, $f_{int}$ and $s_m$ are treated as the functions of $w_m$ and $w_{max}$ and every explicit scheme of CE($j$, $n + 1/2$) is derived, Eq. (4) can be simplified to
(w_m)^{n+1/2} = \frac{1}{2} [(w_m)^n_{j-1/2} + (w_m)^n_{j+1/2}] + \frac{1}{\Delta x} [(g_m)^n_{j-1/2} - (g_m)^n_{j+1/2}] - \frac{\Delta t}{4} [(\tilde{g}_m)^n_{j-1/2} + (\tilde{g}_m)^n_{j+1/2}]

where

\begin{align*}
I_m &= \frac{1}{2} [(w_m)^n_{j-1/2} + (w_m)^n_{j+1/2}] + \frac{1}{\Delta x} [(g_m)^n_{j-1/2} - (g_m)^n_{j+1/2}] - \frac{\Delta t}{4} [(\tilde{g}_m)^n_{j-1/2} + (\tilde{g}_m)^n_{j+1/2}] \\
\end{align*}

For the discontinuous flow field of dramatic changes, \((w_{mx})^{n+1/2}\) is extended to be the weighted average of \((w_{mx-})^{n+1/2}\) and \((w_{mx+})^{n+1/2}\), i.e.

\begin{align*}
(w_{mx})^{n+1/2} &= \frac{[w_{mx+}]^{n+1/2}_j \cdot [w_{mx-}]^{n+1/2}_j + [w_{mx-}]^{n+1/2}_j \cdot [w_{mx+}]^{n+1/2}_j}{[w_{mx+}]^{n+1/2}_j + [w_{mx-}]^{n+1/2}_j} \\
\end{align*}

when \(\left|\frac{[w_{mx+}]^{n+1/2}_j}{[w_{mx-}]^{n+1/2}_j}\right| = 0,

\begin{align*}
(w_{mx})^{n+1/2} &= \frac{[w_{mx-}]^{n+1/2}_j + [w_{mx+}]^{n+1/2}_j}{2} = 0.
\end{align*}

where \(c\) is a real constant taken as 1 or 2.

2.3. Calculation of boundary conditions and initial conditions

Before the intake and exhaust valves are opened, the flow rate in pipe is almost zero, and the pressure fluctuation is also low. This state is defined as the initial conditions, the flow rate in pipe is zero at this time, and the static pressure in pipe is equal to the prevailing atmospheric pressure.

Only the state vector \(w_m\) at the boundary and the value of its spatial derivative \(w_m\) at each integer time layer are needed in the space-time conservation element and solution element method. However in each half-time layer \((n = 1/2, 1 + 1/2, 2 + 1/2, \ldots)\), the values of \(w_m\) and \(w_{mx}\) at each grid point are calculated from the previous integer time layer, thus the values are no boundary values.

The quasi-steady flow assumption which is commonly used in engine numerical simulation is taken as the boundary condition of pipe end to obtain the values of \((w_m)^n\) on the boundary.

3. Validation of simulation model

3.1. Simulation model

Most of intake and exhaust systems of engines are the pipe flow systems. Since the flow area of the diesel exhaust valve changes rapidly, an unstable flow is caused by heat exchange when the working fluid flows through the valve. The unstable flow is assumed to be the adiabatic and isentropic quasi-steady flow for simplifying the calculation, and the flow control equations are solved by using CE/SE method.

Based on the actual working conditions of single-cylinder diesel engine, an one-dimensional thermal power model has been established for performance simulation, as shown in Fig. 3.

3.2. Experimental conditions and methods

In the experiment, a single-cylinder four-stroke direct injection diesel engine is used. Its bore is 132mm, the stroke is 145mm, the length of connecting rod is 262mm, and the compression ratio is 17. A high pressure common rail is used for fuel supply, the fuel injection advance angle is 15°, the opening and closing angles of intake valve are 27°CA BTDC and 40°CA ATDC, respectively, and the opening and closing angles of exhaust valves are 58°CA BBDC & 25°CA ATDC, respectively.

The measured Hg pressure is used as test intake pressure \(p_{in}\), and the average value from the pressure sensor of the combustion analyzer is used as test exhaust pressure \(p_{ex}\). The simulation boundary conditions are that the intake pressure is 0.3 MPa, the intake temperature is 333 K; the exhaust pressure is 0.15 MPa, the exhaust temperature is 523 K; the combustion model is double Vibe model, and the heat transfer model is Woschni1978 model.

3.3. Analysis of experimental results

According to the principle of that the simulation parameters corresponds with the experimental parameters, the effective power \(P_e\), peak firing pressure \(P_{max}\), fuel consumption \(b_c\) and other performance parameters of diesel engine are compared at the engine speed of 2100 r/min. The deviation between the simulated values and the experimental results is minimized, as
shown in Table 1. The simulated main performance parameters are in good agreement with the experimental results, and the maximum error of effective fuel consumption is up to 3.18%.

Fig. 4 shows the pressure fluctuation curves of single-cylinder diesel engine. It can be seen from Fig. 4 that the experimental results are in good agreement with the simulated values, and the correlation coefficient of the two curves is 0.99. The phases of their peak pressures are approximately the same, but the simulated value is 0.4 MPa larger than the experiment value.

The pressure curves of exhaust duct outlet at the measuring point MP2 are shown in Fig. 5. Fig. 5 shows the variations of phases obtained from the experimental results and simulated values are the same, the peak values are relatively close to each other, and the correlation coefficients of the two curves are 0.89. Thus it can be seen that CE/SE method is very applicable to the unsteady flows of intake and exhaust pipes of diesel engine. The model can be used to analyze the charging coefficient in next step.

The outlet pressure of exhaust duct at the measuring point MP2 is simulated by using finite volume method. The results are shown in Fig. 6. The exhaust pressure fluctuation obtained by the experimental and simulated results has a significant phase difference, a larger non-physical oscillation partially appears in the calculated result, the deviation of the peak value is high, and the correlation coefficient of the two curves is 0.58. It indicates that the finite volume method has obvious disadvantages in simulating the exhaust system of HPD diesel engine in which the shock flow exists.

4. The influence of improved power density on charging coefficient

The influence of improved rotational speed and exhaust pressure on the charging coefficient of single-cylinder diesel engine is analyzed by using a calibration model and CE/SE method. When the rotational speed is improved, the excess air coefficient remains unchanged, and the fuel delivery per cycle is adjusted according to inflow air mass. Fig. 7 shows the variation of charging coefficient with intake pressure. It can be seen from Fig. 7 that the experimental result is relatively close to the simulated result. The charging coefficient slowly increases with the increase in intake pressure.

Fig. 8 shows the variation curve of charging coefficient with engine speed. It can be seen from Fig. 8 that the

<table>
<thead>
<tr>
<th>$P/\text{kW}$</th>
<th>$P_{\text{max}}/\text{MPa}$</th>
<th>$b_0/\text{g kW}^{-1}\text{h}^{-1}$</th>
<th>$m_{\text{in}}/\text{g}$</th>
<th>$m_{\text{ex}}/\text{g}$</th>
<th>Deviation/%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Experimental</td>
<td>80.20</td>
<td>17.05</td>
<td>237.53</td>
<td>7.35</td>
<td>7.64</td>
</tr>
<tr>
<td>Calculated</td>
<td>81.82</td>
<td>17.46</td>
<td>229.97</td>
<td>7.14</td>
<td>7.42</td>
</tr>
<tr>
<td>Deviation/%</td>
<td>2.02</td>
<td>2.40</td>
<td>3.18</td>
<td>2.86</td>
<td>2.88</td>
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</tbody>
</table>
experimental result is relatively close to the simulated result. With the increase in engine speed, the charging coefficient obviously reduces.

5. Conclusions

(1) CE/SE method is used for the discrete governing equations. The program for the simulation of intake and exhaust flows of HPD diesel engine is simpler and more applicable.

(2) The simulated results of diesel engine exhaust process show that the simulated main performance parameters are in good agreement with the experimental results at the engine speed of 2100 r/min, the maximum error of effective fuel consumption is up to 3.18%; the simulation curve of pressure fluctuation at the exhaust duct outlet is in good agreement with the experimental curve, and the correlation coefficient is 0.94.

(3) CE/SE method can be used to reflect the intake and exhaust flow characteristics of HPD diesel engine. A solution is provided to multi-component mixed flow issue or other issues of other systems of engine. It can also provide more accurate initial conditions for in-cylinder multidimensional simulation.

References