Numerical Simulation of Gas Flows in the Cylinders of Four-Stroke Cycle Engines*  
(1st Report, Analysis of the Process of Swirl Generation during Intake Stroke)  
By Tomoyuki WAKISAKA**, Yuzuru SHIMAMOTO***, Yoshihiro ISSHIKI*** and Toyoshige SHIBATA*  

A method for calculating the three-dimensional gas flows during intake stroke in the cylinders of four-stroke cycle engines has been examined using simplified model engines. The process of induction swirl generation in a model engine with an off-center intake valve has been simulated numerically. The results show that the flow pattern in the cylinder varies in many ways with the distribution of inflow velocities around the intake valve. The effects of the differences in inflow velocity distribution (helical-port and directional-port types) on the generation of swirl are predicted.

Key Words: Internal Combustion Engine, In-Cylinder Flow, Induction Swirl, Numerical Simulation, Navier-Stokes Equation, Finite Volume Method

1. Introduction

It is one of the most important tasks to investigate the gas flows in the cylinders of internal combustion engines for realizing the low-pollution and high-efficiency combustion. Since the in-cylinder flow is in a very complicated three-dimensional turbulent state, there are many barriers to revealing the flow only by experimental study. Therefore, recently the methods of flow analysis by a computer have been developed. So far, however, most of the flow analyses have been performed on the axisymmetric or two-dimensional flows[7-9], and few analyses about the three-dimensional flows in actual engines have been reported[10-12].

The objective of this study is to perform a three-dimensional numerical analysis of the process of swirl generation during intake stroke in the cylinders of four-stroke cycle engines. First, using simple model engines, investigation has been made into the method for calculating a three-dimensional unsteady compressible viscous flow in the cylinder and also into the method for treating the boundary of an off-center intake valve. Second, using a model engine which imitates a practical engine, the process of induction swirl generation has been simulated numerically and the differences in the state of swirl generation due to the inflow velocity distribution around the intake valve have been examined.

2. Nomenclature

The main symbols in this study are as follows:

- $c_v$ = specific heat at constant pressure
- $h$ = specific enthalpy
- $p$ = pressure
- $r$ = radial coordinate
- $R$ = gas constant
- $t$ = time
- $T$ = temperature
- $v$ = flow velocity vector
- $v_r$ = radial flow velocity
- $v_\phi$ = tangential flow velocity
- $v_z$ = axial flow velocity
- $v_t$ = axial flow velocity relative to the expanding/contracting coordinate
- $v_{r*}$ = resultant velocity of $v_r$ and $v_\phi$
- $v_{z*}$ = resultant velocity of $v_r$ and $v_z$
- $z$ = axial coordinate
- $\delta r$, $\delta \phi$, $\delta z$ = calculation grid spacing in $r$, $\phi$ and $z$ directions
- $\Delta t$ = time increment
- $\Theta$ = tangential coordinate
- $\theta$ = crank angle measured from TDC of intake stroke
- $\mu$ = viscosity
- $\xi$ = non-dimensional axial coordinate
- $\rho$ = density
- $\sigma$ = Prandtl number

Subscripts

- eff = effective
- lam = laminar
- pis = piston

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** Associate Professor, Department of Mechanical Engineering, Kyoto University (Yoshidahonmachi, Sakyo-ku, Kyoto, 606 Japan).
*** Professor, Department of Mechanical Engineering, Kyoto University.
**** Instructor, Department of Mechanical Engineering, Kyoto University.
+ Engineer, Mitsubishi Motor Corporation (1 Tatsumi-cho, Uzumasa, Ukyo-ku, Kyoto, 616 Japan).
turb = turbulent
val = intake valve

3. Basic equations

Use is made of a cylindrical coordinate system (r, θ, z) whose z-axis is set on the cylinder axis and its origin is at the center of the cylinder head as shown in Fig. 1. Air, which is assumed to be an ideal gas, is used as the working gas. The conservation equations of mass, momentum and energy of air are expressed generally by the following partial differential equation (namely, general conservation equation):

$$\frac{\partial (\rho \phi)}{\partial t} + \frac{1}{r} \frac{\partial}{\partial r} \left( r \rho \phi \frac{\partial \phi}{\partial r} \right) + \frac{\partial}{\partial z} \left( \rho \phi \frac{\partial \phi}{\partial z} \right) = S_\phi \quad \cdots \cdots \cdots (1)$$

where,
\( \phi \) = dependent variable averaged spatially in a control volume (Table 1)
\( \Gamma_\phi \) = effective diffusivity concerning \( \phi \) (Table 1)
\( S_\phi \) = source term expressing the production or dissipation of \( \phi \).

The positions of boundaries in the cylinder change due to the motion of the piston and intake valve. These moving boundaries are easily treated by applying the following expanding/contracting coordinate transformation:

In the case of the off-center intake hole model without any intake valve (Fig. 1(a)), for the region in the cylinder (0 ≤ z ≤ z_{pm}), the coordinate transformation

\[ \xi = z / z_{pm} \]

is made.

Using the expanding/contracting cylindrical coordinates (r, θ, ξ), Eq. (1) is expressed as

$$\frac{1}{z_{pm}} \frac{\partial}{\partial \xi} \left( \rho z_{pm} \phi \right) = \frac{1}{r} \frac{\partial}{\partial r} \left( r \phi \frac{\partial \phi}{\partial r} \right) + \frac{\partial}{\partial \xi} \left( \frac{\partial \phi}{\partial \xi} \right) + \frac{1}{z_{pm}} \frac{\partial}{\partial \xi} \left( \rho \frac{\partial \phi}{\partial \xi} \right) + S_\phi \quad \cdots \cdots \cdots (2)$$

where,
\( z_{pm} = z / z_{pm} \)
\( S_\phi = - \xi v_{pm} \)

\( (z_{pm} = \text{distance between the cylinder head and the piston head}, \ v_{pm} = \text{piston velocity}) \). The expression for \( S_\phi \) in Eq. (2) is shown in Table 1.

In the case of the models with a central or off-center intake valve (its thickness is assumed to be zero) (Fig. 1(b)), the following coordinate transformation, which differs with regions, is applied:

(i) For region I (0 ≤ z ≤ z_{val}):

\[ \xi = z / z_{val} \]
\[ z_{pm} = z_{val} \]
\[ \xi v_{pm} = \xi v_{val} \]

Fig. 1 Coordinate system

<table>
<thead>
<tr>
<th>Equation</th>
<th>( \phi )</th>
<th>( \Gamma_\phi )</th>
<th>( S_\phi )</th>
<th>Note:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass</td>
<td>1</td>
<td>0</td>
<td>0</td>
<td>div ( v = \frac{1}{r} \frac{\partial}{\partial r} (rv_r) + \frac{1}{z_{pm}} \frac{\partial}{\partial z} (z_{pm}v_z) )</td>
</tr>
<tr>
<td>Radial momentum</td>
<td>( r \mu_{\text{int}} )</td>
<td>( \frac{\partial \phi}{\partial r} + \frac{1}{r} \left( \frac{\partial}{\partial r} \left( r \mu \phi \frac{\partial \phi}{\partial r} \right) \right) + \frac{\partial}{\partial z} \left( \mu \phi \frac{\partial \phi}{\partial z} \right) + \frac{1}{z_{pm}} \frac{\partial}{\partial \xi} (\mu \phi \frac{\partial \phi}{\partial \xi}) )</td>
<td></td>
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<tr>
<td>Tangential momentum</td>
<td>( \mu_{\text{int}} )</td>
<td>( \frac{1}{r} \frac{\partial}{\partial r} (z_{pm}v_r) + \frac{1}{r} \left( \frac{\partial}{\partial r} \left( z_{pm} \mu \phi \frac{\partial \phi}{\partial r} \right) \right) + \frac{\partial}{\partial z} \left( z_{pm} \mu \phi \frac{\partial \phi}{\partial z} \right) )</td>
<td></td>
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</tr>
<tr>
<td>Axial momentum</td>
<td>( \mu_{\text{int}} )</td>
<td>( \frac{\partial}{\partial z} \left( z_{pm} \mu \phi \frac{\partial \phi}{\partial z} \right) )</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Energy</td>
<td>( \left( \frac{E_{\text{int}}}{\text{deg}} \right) )</td>
<td>( \phi \frac{\partial \phi}{\partial t} + \mu \phi \frac{\partial \phi}{\partial r} + \frac{\partial}{\partial r} (r \phi \mu \frac{\partial \phi}{\partial r}) + \frac{\partial}{\partial z} \left( z_{pm} \frac{\partial \phi}{\partial z} \right) + \frac{1}{z_{pm}} \frac{\partial}{\partial \xi} (\mu \frac{\partial \phi}{\partial \xi}) )</td>
<td></td>
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\( \left( \text{deg} = 0.017 \right) \)
\( k = c_s \gamma \)
(ii) For region II $(z_{va} \leq z \leq z_{va}^0)$:

$$\xi = \frac{(z - z_{va})}{(z_{va}^0 - z_{va})}$$

$$z_{va} = z_{va}^0 - z_{va}^0 (\xi - 1)$$

$$z_{min} = \text{distance between the cylinder head and the valve face, } z_{va} = \text{intake valve velocity.}$$

The general conservation equation in either region is expressed as Eq. (2).

The turbulent viscosity $\mu_{turb}$ has to be determined on the basis of some appropriate turbulence model. In this study, $\mu_{turb}$ is to be determined by the Subgrid Scale model:

$$\mu_{turb} = (C_\mu \Delta t)^2 \left[ (e_{t} + e_{s} + e_{r}) + (e_{t} + e_{s} + e_{r}) \right]$$

where,

- $C_\mu = \text{constant (assigned the value 0.1)}$
- $\Delta t = \text{characteristic length of the calculation grid (real volume of each control volume)}$

In using this model, it is not necessary to solve any more conservation equations, and furthermore, it includes only one empirical constant in contrast with the $k-\varepsilon$ model. Taking these advantages into consideration, $\mu_{turb}$ by the Subgrid Scale model is used in this study.

4. Method for numerical calculation

4.1 Discretized equations

In this study, the general conservation equation (2) is analyzed numerically using the modified Tank and Tube method which has been developed originally by Gosman et al. and Spalding et al. The method is described in the following:

The space in the cylinder is divided into many small hexahedrons, namely, control volumes, an example of which is shown in Fig. 2, by the grid lines of cylindrical coordinates $(r, \theta, z)$. It is assumed that pressure $p$, density $\rho$ and temperature $T$ distribute uniformly in each control volume and they are allocated at the center (point $P$ in Fig. 2) of the volume. Points N, S, T, B, E and W denote the neighboring grid points around the point $P$. It is also assumed that the three components of velocity vector, $u_r$, $u_\theta$ and $u_z$ (or $u_r$), distribute uniformly on the faces of the control volume and each of them is allocated at the center (point $n$, s, t, b, e or w) of each face as shown in Fig. 2 (Staggered Grid method). By integrating the equation "Eq. (2) x $z_{va}^0"$ of the control volume on the basis of the above-mentioned assumptions and by deriving the finite difference of the temporal derivative term, the following equation is obtained:

$$\frac{(\rho z_{va}^0 V)_{P}^{new} - (\rho z_{va}^0 V)_{P}^{old}}{\Delta t} = F_{P}^{new}$$

where,

$$F_{P} = \int_{b}^{r_{ext}} \int_{b}^{i_{ext}} \int_{i_{ext}}^{r_{ext}} \text{[right hand side of Eq. (2)]} \, d\theta \, dz$$

$$V = \text{volume of the control volume}$$

$$= \frac{1}{2} (r_1^2 - r_i^2) \Delta z \Delta \theta$$

Subscript P denotes the point P, and superscripts OLD and NEW represent the values at time $t$ and that at $t + \Delta t$, respectively.

The symbol $f$ in Eq. (3) is a weighting factor ($0 \leq f \leq 1$) concerning the time at which convection, diffusion and source terms are evaluated. In the case of the original Tank and Tube method, the value 1 is assigned to $f$ (fully-implicit method). In this study, however, the value 0.5 is assigned to $f$ (Crank-Nicolson's implicit method) for using the scheme having the second-order accuracy with respect to time. Since these implicit schemes have no limiting conditions concerning stability, it is not necessary to reduce the time increment according to the decrease of grid spacing or the increase of flow velocity unlike the explicit scheme ($f = \theta$), and it is possible to calculate with a relatively large time increment. By performing the integration of Eq. (4) and by deriving the finite difference of the convection and diffusion terms by means of the Hybrid method, the following equation is obtained:

$$F = \frac{\Sigma m_c}{\Sigma} \phi_c + \Sigma (m_c \gamma_c)(\phi_c - \phi_P)$$

$$+ (\Sigma z_{va}^0 V)_{P}$$

where, subscript $c$ denotes the point n, s, t, b, e or w; when $c$ is the subscript of $\phi$, it denotes the point N, S, T, B, E or W. The symbol $m_c$ represents the mass flux flowing into the control volume through each face of the volume, and $\gamma_c$ is a coefficient which expresses the contribution of the convection and diffusion terms (their detailed expressions have been described in Reference (10)).

The same operation as the above for the mass conservation equation gives

$$\frac{(\rho z_{va}^0 V)_{P}^{new} - (\rho z_{va}^0 V)_{P}^{old}}{\Delta t} = (\Sigma m_c \gamma_c)_{P}$$

$$+ (1 - f)(\Sigma z_{va}^0 V)_{old}$$

Fig. 2 Control volume
By substituting Eq. (5) into \( \Phi^{\text{NEW}} \) in Eq. (3) and by carrying out the operation "[Eq. (6) \times \Phi^{\text{NEW}}] \times 1/f" in the following final discretized equation is obtained:

\[
A_i \Phi_i^{\text{NEW}} = \sum_c A_{ci} \Phi_i^{\text{NEW}} + A_{ci} \Phi_c^{\text{OLD}} + B_c , \quad (7)
\]

(subscript c denotes the point N, S, T, B, E or W). Here,

\[
A_i = \left( \alpha_i \rho_i c_{pi} \right)^{\text{NEW}}, \quad A_{ci} = \left( \frac{\rho_i c_{pi} V_i}{\alpha_i} \right)^{\text{NEW}} + \left( \frac{1}{f} \right) P_c^{\text{OLD}},
\]

\[
B_c = \left( S_p \alpha_i \rho_i c_{pi} \right)^{\text{NEW}} - \left( S_p \rho_i c_{pi} V_i \right)^{\text{NEW}}.
\]

For stabilizing the calculation, the source term \( S_p \) is linearized as

\[
S_p = S_{p0} + S_{p1} \Delta p.
\]

Equation (7) is a discretized equation for the conservative quantity (energy) with respect to the control volume shown in Fig. 2. As to the velocity components \( v_x, v_y, \) and \( v_z \), three discretized equations similar to Eq. (7) are also obtained by an operation similar to the above with respect to each control volume.

4.2 Calculation algorithm

Calculation is carried out using the SIMPLE method\(^{[14]} \). The calculation procedure at a time step \( t + \Delta t \) is as follows:

First, using the guessed pressure \( p^* \) (the value at the previous time step is usually used), the discretized equations for three velocity components are solved to obtain their approximate values. Second, such pressure correction \( \Delta p \) at each grid point, as is necessary for the velocity components to satisfy the mass conservation equation (6), is calculated from the following \( \rho \) equation:

\[
A_i \rho_i = \sum_c A_{ci} \rho_c + E_i , \quad (8)
\]

where,

\[
A_i = \text{coefficient derived from each discretized equation for the velocity components (subscript c denotes the point N, S, T, B, E or W)}
\]

\[
A_{ci} = \left( \alpha_i \rho_i c_{pi} V_i \right)^{\text{NEW}} + \left( \frac{1}{f} \right) P_i^{\text{OLD}}
\]

\[
E_i = \left( \text{right hand side - left hand side of Eq. (6)} \right) / \Delta t,
\]

Then, the velocity components and pressure at each grid point are corrected by \( \Delta p \). Temperature \( T \) is calculated by solving the discretized equation for energy. Density \( \rho \) at any point is computed from the equation of state \( \rho = \rho(T, \rho) \) whenever \( \rho \) and \( T \) are renewed.

Next, returning to the first step to use the corrected pressure \( p \) as the new guessed pressure \( p^* \), the above-mentioned procedure is repeated several times. When the solution is converged, the procedure is advanced to the next time step. The criterion for convergence is that all the spatially averaged values of the absolute residual sources of the discretized equations for mass, momentum and energy become equal to or lower than a prescribed value (0.05% of the reference value of each conservative quantity). Each discretized equation and the \( \rho \) equation are solved by the TDMA method\(^{[14]} \).

4.3 Boundary conditions

No-slip and constant-temperature conditions are imposed on the velocity and temperature of air on the wall, respectively. The pressure gradient normal to the wall is set at zero. For calculating the velocity and temperature at the grid points adjacent to the wall, the following law of the wall\(^{[17]} \) is applied:

\[
\frac{u}{u_w} = 0.4 \ln \left( \frac{y}{y_0} \right) \quad \text{when} \quad y_0 \leq 130.3,
\]

\[
\frac{u}{u_w} = \sqrt{2} \ln \left( \frac{y}{y_0} \right) \quad \text{when} \quad y_0 < 130.3,
\]

\[
y = \text{distance from the wall}
\]

\[
u_w = \text{friction velocity}
\]

\[
u = \text{velocity component parallel to the wall}
\]

\[
w = \text{wall heat flux}
\]

\[
T_w = \text{wall temperature}
\]

4.4 Initial conditions

The air in the cylinder at TDC of intake stroke (\( \theta = 0^\circ \)) is assumed to be quiescent and at the standard pressure and room temperature. Calculation is carried out from TDC at every 5° crank angle.

5. Calculation models and results

5.1 Model engines

The specifications of every model engine are as follows:

Cylinder bore = 130 mm, stroke = 150 mm, connecting rod length = 250 mm, compression ratio = 8.5, diameter of intake valve or hole = 52 mm, engine speed = 1200 rpm. Cylinder head and piston head are flat.

Three calculation models, namely, off-center intake hole model, central intake valve model and off-center intake valve model are considered. The first and second models are used for examining the inflow boundary conditions and so on as a preparatory step for treating the last model which imitates a practical engine. In the cases of the models with an off-center circular opening (hole or valve), the center of the opening is placed at the position of 30.5 mm away from the cylinder axis. The valve lift \( l \) is prescribed by the equation

\[
l = l_{\text{max}} \sin[(\pi (\theta - \theta_v))/\Delta \theta]
\]

\[
(l_{\text{max}} = 14.6 \text{ mm, } \theta_v = -35^\circ, \Delta \theta = 250^\circ).
\]
5.2 Off-center intake hole model

This model has a circular opening (intake hole) at the off-center position on the cylinder head as shown in Fig. 3(a). Using this model, the method for treating the inflow condition through the off-center hole is investigated, and it is examined whether the three-dimensional in-cylinder flow is able to be calculated by the method described in Sec. 4.

As to the inflow condition, it is assumed that air flows into the cylinder through the hole obliquely to the cylinder head at an angle \( \alpha \) of \( \pi/6 \), as shown in Fig. 3(a). The inflow velocity component parallel to the cylinder diameter passing on the center of the intake hole is set at zero, and the inflow velocity distribution over the intake hole is assumed to be uniform. The grid divisions of calculation region in \( r \), \( \theta \) and \( \zeta \) directions are 10, 12 and 10, respectively (Fig. 3(b)).

In order to treat the inflow boundary condition at the off-center intake hole in such a relatively coarse calculation grid, the Porosity method\(^\text{14}\) is used: If one face among the six faces of a control volume is partially blocked by the wall, only the unblocked area of that face contributes to the flux of each conservative quantity at the face.

The flow velocity at the inflow boundary is determined so that the continuity equation concerning all the control volumes in the cylinder may be satisfied. The pressure and temperature of air at the inflow boundary are given on the assumption that the outside air at the standard pressure and room temperature makes an isentropic change.

Some examples of the calculated results of flow velocity distribution are shown in Fig. 4. The resultant velocities \( \nu_A \) and \( \nu_B \) are shown in the \( r-z \) and \( r-\theta \) planes, respectively, with the vectors whose starting points are at the calculation points (for example, point \( P \) in Fig. 2) of those velocities. The velocities at the inflow boundary are not plotted for simplicity.

From Fig. 4(a)-(e), it is found that a complicated three-dimensional flow is generated in the cylinder and the vortex motion appearing in the \( r-z \) and \( r-\theta \) planes varies in many ways with time and space. The strong swirl generated in the \( r-\theta \) planes is not axisymmetric but its rotation center is located helically around the cylinder axis. The radial distribution of \( \nu_\theta \) varies considerably with the azimuthal position.

As mentioned above, it has been confirmed that a converged solution is obtained though the calculation grid is relatively coarse, and that the three-dimensional in-cylinder flow can be calculated by using the method described in Sec. 4 and this section.

5.3 Central intake valve model

This model has an intake valve at the center of the cylinder head, as shown in Fig. 5(a). The method for treating the boundary of the intake valve is investigated. It is assumed that air flows into the cylinder along the valve seat (the seat angle \( \alpha = \pi/4 \)) through the valve clearance. The tangential velocity component \( \nu_\theta \) is set equal to \( \nu_\theta = \nu_\zeta \). Since the

![Diagram of Central Intake Valve Model](image)

Fig. 5 Central intake valve model
inflow velocity distribution in $\theta$ and $\xi$ directions at the valve periphery is assumed to be uniform, the flow becomes axisymmetric. The calculation, however, is carried out in the three-dimensional manner for testing the axisymmetry of the calculated results. The expanding/contracting coordinate transformation in Sec. 3 is applied and the calculation region is divided into 14, 12 and 15 segments in $r$, $\theta$ and $\xi$ directions, respectively, as shown in Fig. 5(b). The flow velocity, pressure and temperature at the valve periphery are given in the same way as in the case of the off-center intake hole model.

Some examples of the calculated results are shown in Fig. 6. From Fig. 6(a)-(c), it is found that two remarkable vortices rotating in the $r$-$z$ plane are generated in the cylinder due to a high-speed flow issuing from the narrow valve clearance. This vortex motion develops gradually with an increase of crank angle, and after that it comes to decay. Since a strong swirling is generated in the $r$-$\theta$ plane as shown in Fig. 6(d), the relative pressure $\partial p = p - p_{rel}$ (where $p_{rel}$ = pressure at the inflow boundary) in that plane has such distribution as shown in Fig. 6(e); $\partial p$ becomes much lower near the cylinder axis. The radial distribution of $v_r$ in the $r$-$\theta$ plane near the middle of the cylinder is similar to the $v_\theta$ distribution in the Rankine vortex (Fig. 6(a)).

As mentioned above, it has become possible to calculate the process of inflow through the valve without any difficulty of treating the moving boundary by using different expanding/contracting coordinates for the two regions above and below the intake valve.

5.4 Off-center intake valve model

This model has an intake valve at the off-center position of the cylinder head as shown in Fig. 7(a). It is assumed that air flows into the cylinder through the valve clearance along the valve seat (the seat angle $\alpha = \pi/4$). As to the pattern of the inflow velocity distribution around the intake valve, the following three models N, D and H are considered:

Model N has a uniform inflow velocity distribution without any tangential velocity component (Fig. 7(b)). Model D, which is modeled after the inflow velocity distribution for a directional port, has such a nonuniform distribution of $v_r$ (Fig. 7(c)) as is expressed by the equation

$$ v_r = \begin{cases} 
1 + \frac{N_0 - 1}{N_0} (\beta' - \beta) & (v_r)_{max}, \\
\beta - \pi \leq \beta' < \beta, \\
\frac{N_0 - 1}{N_0} (\beta' - \beta) & (v_r)_{max}, \\
\beta \leq \beta' < \beta + \pi
\end{cases} $$

where,

- $v_r$ = radial component of the inflow velocity vector in the cylindrical coordinates ($r'$, $\theta'$, $z'$) whose $z'$-axis is set on the valve axis
- $N_0 = (v_r)_{max}/(v_r)_{max} \approx 3$
- $\beta$ = angle between the $r'$-axis and the $(v_r)_{max}$ vector.

Fig. 6 Distributions of $v_\theta$, $v_r$, etc. in central intake valve model

Fig. 7 Off-center intake valve models (Models N, D and H)
The angle $\beta$ is set at $\pi/3$, $\pi/2$, $2\pi/3$ or $5\pi/6$ (each is designated Model D-60, D-90, D-120 or D-150, respectively). Model II, which is modeled after the inflow velocity distribution for a helical port, has a uniform inflow velocity distribution with the tangential velocity component (Fig. 7(d)) whose deflection angle $\gamma$ is set at $\pi/8$, $\pi/6$ or $\pi/4$ (each is designated Model H-22.5, H-30 or H-45). In every model, the axial component $v_r$ of inflow velocity vector at the valve periphery is prescribed as $v_r = v_r \tan \alpha$.

The calculation region is divided so that the grid points around the intake valve can be allocated as near as possible in the relatively coarse grid as shown in Fig. 7(e). The grid divisions in $r$, $\theta$, and $z$ directions are 10, 16, and 14, respectively. The symbol $l$ in Fig. 7(e) represents the slab number counted from the upper end of the cylinder.

A pipe with a bend is attached to the intake hole (the center line length of the pipe is about 193 mm). The atmosphere at the standard pressure and room temperature is sucked from the pipe inlet. The pressure and temperature of air at the valve periphery are assumed to distribute uniformly. Their values and the intake mass flow rate are calculated in advance by the one-dimensional characteristic method. The boundary conditions at the intake valve are treated by the Forsivity method similarly to the case of the off-center intake hole model.

Calculation is carried out until BDC of intake stroke. Typical calculated velocity distributions are shown in Figs. 8-12 (the positions of the planes in these figures are indicated in Fig. 7(e)). It is found that there occurs a very complicated vortex motion at 90° ATDC in the $r-z$ planes of Models H-45, D-90 and N. The rolling up of flow below the intake valve is most remarkable in Model H-45 (Figs. 8-10; (a) and (b)). In Model N, naturally, there occurs no rotating motion around the cylinder axis (Fig. 10(d)). A swirl is generated in either of Models H-45 and D-90, but its generation process in each model differs greatly from each other. In the case of Model H-45, although the swirl already develops to a considerable extent at 90° ATDC in both of the lower $r-\theta$ plane and the middle $r-\theta$ plane, the swirl center exists considerably away from the cylinder axis (Fig. 8(c),(d)); it is located helically around the cylinder axis. At BDC, however, the swirl center is nearly on the cylinder axis in every $r-\theta$ plane, and the $v_r$ distribution varies little from plane to plane (Fig. 11). On the contrary, in the case of Model D-90, the velocity distribution in each $r-\theta$ plane at 90° ATDC varies greatly with the axial position of the

Fig. 8
Distributions of $v_r$ and $v_z$ in Model H-45 ($\theta=90^\circ$ ATDC)

Fig. 9
Distributions of $v_r$ and $v_z$ in Model D-90 ($\theta=90^\circ$ ATDC)

60 m/s ($v_r$ and $v_z$ in Figs. 8-12)

60 m/s ($v_r$ in Figs. 11 and 12)
plane; Although the rotating motion is active in the lower region of the cylinder (Fig. 9(c)); the motion is disorderly in the middle region (Fig. 9(c)). At BDC, the rotating motion becomes orderly, though the \( n_s \) distribution varies a little with the axial position of the \( r \theta \) plane (Fig. 12). Swirls are generated also in Models D-60, D-120 and D-150, and their flow patterns are considerably disorderly. The computer CPU time needed for calculating the process from 0° to 180° ATDC in the above model is 30-35 minutes.

The ratio of the mean rotation speed of swirl to the engine speed, namely, swirl ratio \( n_s \), is calculated for all the off-center intake valve models except Model N by the following equation:

\[
    n_s = \frac{\int_0^{z_m} \int_0^{\pi} \int_0^{2\pi} \rho r^2 v_r \rho \, rd\theta \, dz}{\int_0^{z_m} \int_0^{\pi} \int_0^{2\pi} \rho r^2 \rho \, rd\theta \, dz}
\]

where,
\( r_c \) = cylinder radius
\( \omega_e \) = angular velocity of engine rotation.
The region right above the intake valve is excluded from the integration range of the above equation.

The variation of \( n_s \) with \( \varphi \) is shown in Fig. 13 (data are plotted at every 10° crank angle). In every model, \( n_s \) increases rapidly with an increase of crank angle until 70°-80° ATDC, and after that, \( n_s \) increases slowly and reaches the maximum. Then, it becomes nearly constant or decreases a little. The value of \( n_s \) at BDC of intake stroke becomes larger in the order of Models D-150, H-22.5, D-120, D-60, D-90, H-30 and H-45. From the variation of \( n_s \) with crank angle, it can be found that there exist some differences in the process of swirl development between the group of Model H's and that of Model D's: The swirl ratios for the group of Model H's tend to reach the maximum at earlier crank angles than those for the group of Model D's.

In order to examine the contribution of the angular momentum of air in each slab (Fig. 7(e)) to the total angular momentum of air in the cylinder, the angular momentum contribution of the \( i \)-th slab (\( n_{si} \)), is calculated by the equation

\[
    (n_{si}) = \frac{\int_0^{z_f} \int_0^{\pi} \int_0^{2\pi} \rho r^2 v_r \rho \, rd\theta \, dz}{\int_0^{z_f} \int_0^{\pi} \int_0^{2\pi} \rho r^2 \rho \, rd\theta \, dz}
\]

where,
\( \Delta z_i \) = thickness of the \( i \)-th slab.
The region right above the intake valve is excluded from the integration range of the above equation.

Shown in Fig. 14 are the variations of \( n_{si} \) with \( \varphi \) in Models H-30 and D-90 which have nearly the same \( n_s \) at BDC of intake stroke (data in the slabs of even number are plotted). In both models, the contribution of the angular momentum in the slabs near the intake valve is large in the early period of intake stroke. Then, the swirl develops with an increase of crank angle in the slabs at the lower part of the cylinder and the value of \( n_{si} \) in these slabs become larger at about 40° ATDC than those in the upper slabs. The variation of \( n_{si} \) after that crank angle differs between the two models. Here, the difference between the maximum and the minimum of \( n_{si} \), below the value (5 ≤ \( i \) ≤ 14) is designated \( \Delta n_{si} \), which becomes the maximum at 80°-90° ATDC in both of Models D-90 and H-30. This maximum value of \( \Delta n_{si} \) in Model D-90 is larger than that in Model H-30, and in the former model the development of swirl delays especially in the slabs near the middle of the cylinder (1 = 8-10). In the group of Model D's, the value of \( \Delta n_{si} \) at 90° ATDC increases with an increase of angle \( \beta \) (Fig. 7(c)) as shown in Fig. 15. At BDC, the values of \( n_{si} \) in both of the group of Model D's and that of Model H's mostly become larger as the position of slab comes nearer the piston. The values of \( \Delta n_{si} \) at BDC in the group of Model H's except Model H-22.5 are smaller than those in the group of Model D's. Thus, the variation of swirl intensity with the axial position is smaller in the case of the inflow velocity distribution of helical-port type than in the case of that of directional-port type. This tendency has also been found from the measured results in a practical engine. As shown

![Fig. 13 Swirl ratios \( n_s \) in off-center intake valve models](image)

![Fig. 14 Angular momentum contributions \( n_{si} \) in Models H-30 and D-90](image)
in Fig. 15, the value of \( m_s \) for each of Model D's varies convexly with angle \( \beta \) and becomes the maximum at \( \beta \) of about \( \pi/2 \). On the other hand, the value of \( n_s \) for each of Model H's increases with an increase of angle \( \gamma \) and the rate of increase in \( n_s \) becomes slightly larger with an increase of \( \gamma \).

As mentioned above, the calculated results which seem to be qualitatively valid have been obtained. Their accuracy will be examined hereafter by comparing them with some experimental data.

6. Conclusions

A method for performing a three-dimensional numerical analysis of in-cylinder flows during intake stroke has been examined. As a result, it has become possible to calculate the process of induction swirl generation under the conditions of a relatively large time increment and a coarse calculation grid by using the Tank and Tube method into which Crank-Nicolson's method is introduced and by applying the expanding/contracting coordinate and the Porosity method.

It is found that the in-cylinder flow during intake stroke in the off-center intake valve model is much more complicated than that in the simpler two models and the flow pattern varies in many ways with the inflow velocity distribution around the intake valve. As to the models having the inflow velocity distributions of the helical-port and directional-port types, the effect of the difference in inflow velocity distribution on the swirl ratio is predicted and it is revealed that the process of induction swirl generation differs between these two types.

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References

(11) p. 126 in Ref. (9).
(12) p. 52 in Ref. (9).