A Method for Predicting Three-Dimensional Flow Characteristics during the Intake Process in Four-Stroke Cycle Internal Combustion Engines

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Abstract

In this paper, a practical method of analysis is described for numerically predicting the three-dimensional flow characteristics such as swirl during the intake process in four-stroke cycle engines. The method is based on the one-dimensional calculation of the gas exchanging process and on the three-dimensional calculation of the gas flow in each region of the intake port and cylinder. The effect of the intake port configuration on the induction swirl intensity and volumetric efficiency can be examined efficiently by this method. Some calculations have been carried out for the case where the configuration of a helical port has been changed systematically. The feasibility has been shown through simulating the variation of the flow characteristics at the intake valve outlet owing to the change of the intake port configuration.

1. Introduction

Gas motion during the gas exchanging process in four-stroke cycle engines is one of the main phenomena which affect the engine performance. In the intake process, the volumetric efficiency and the characteristics of the in-cylinder gas flow, such as swirl, which are influenced by the configuration of the intake port[1], are principal factors. Therefore, the intake port configuration is one of the important design parameters. The velocity distributions at the outlet boundary of the intake valve have been investigated precisely by experiment[2)-4)]. Alternatively, a few attempts to numerically analyze the flow in the intake port have been recently made[5), 6)]. Also, the three-dimensional numerical simulation of the

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in-cylinder flow has been carried out[7),8)].

In designing the intake ports, attention has to be paid to the flow characteristics at the valve outlet. In high-speed diesel engines, for example, an adequate induction swirl has to be generated in the cylinder and, moreover, a high volumetric efficiency has to be realized. Recently, some attempts have been made in order to improve the engine performance by controlling the induction swirl intensity according to either the engine speed or the load with some variable mechanisms installed in the intake ports[9)-12)]. The investigation of the effect of the intake port configuration on the flow characteristics thus becomes more and more important.

When a new intake port is developed, usually its optimum configuration is searched for experimentally, through a trial and error procedure which takes much time and labor, with the steady flow test rig to evaluate the swirl generating ability. Therefore, a method which will enable designers to develop intake ports efficiently is expected.

Under the above-mentioned background, the authors have made some studies[13),14)] in order to develop a computer program by which the three-dimensional flows in four-stroke cycle engines can be numerically predicted. The authors have applied the technique of the numerical analysis of the three-dimensional flows to the gas flows in the intake port and cylinder. When such a computer program is completed, it will become possible to develop the intake ports efficiently by utilizing the program as a tool for aiding the designer in the early designing stage.

The purpose of this paper is to propose a practical method of analysis, that is, a simulation method applicable to the actual engine conditions with an easiness in changing the intake port configurations. Then, it is hoped that this method will reduce the calculation time and the computer memories required for predicting the three-dimensional flow characteristics during intake stroke. The purpose is also to confirm the feasibility of predicting the variation of the swirl intensity and volumetric efficiency owing to the change of the intake port configuration with this method.

It is necessary to analyze simultaneously the whole flow field of the intake system, including the intake port and cylinder, for predicting the induction swirl accurately. At present, however, it seems that the simultaneous three-dimensional analysis of the whole flow field is impractical by existing computers, from the standpoint of the calculation time and the computer memories. Therefore, the authors have chosen to adopt the following method:

Firstly, the one-dimensional calculation of the gas exchanging process is executed. Using these calculated results as the boundary conditions, the effect of the intake port configuration on the flow characteristics at the intake valve outlet is investigated by calculating the three-dimensional unsteady flow in the intake port separately. Next, the three-dimensional in-cylinder flow is numerically analyzed using the calculated velocity distribution around the intake valve. Such separated calculations make it possible to predict the flow characteristics efficiently.

In this paper, some calculations are carried out with this method for the case where the configuration of a helical port is changed systematically to investigate its effect on the swirl intensity and volumetric efficiency.

2. Main Notations

L_v	: valve lift
n	: engine speed
p_a	: atmospheric pressure
p_c	: cylinder pressure
S_C, S_{cyl}	: swirl ratios
x, y, z	: Cartesian coordinates
η_v	: volumetric efficiency
Θ	: crank angle measured from TDC of intake stroke

Superscript

* : normalized value

3. Method of Analysis and Its Feature

3.1 Calculation procedure

The calculation procedure is shown in Figs. 1 and 2. The computer program consists of three codes – "EXCHANGER" (Gas **Exchang**ing Process Analy**zer**), FLARE – PORT (**Flo**w Analyzer in **R**eciprocating **E**ngines for Intake **Port**) and FLARE–CYL (**Flo**w Analyzer in **R**eciprocating **E**ngines for **Cyl**inder).

The calculation is put forward through the following steps using each code:

(i) To begin with, the main specifications of an engine to be developed and the configuration of a prototype intake port are prepared.

(ii) A one-dimensional calculation of the gas exchanging process in the intake and exhaust systems, including the cylinder, is carried out, and the state quantities at the inlet of the intake port and in the cylinder are obtained at each

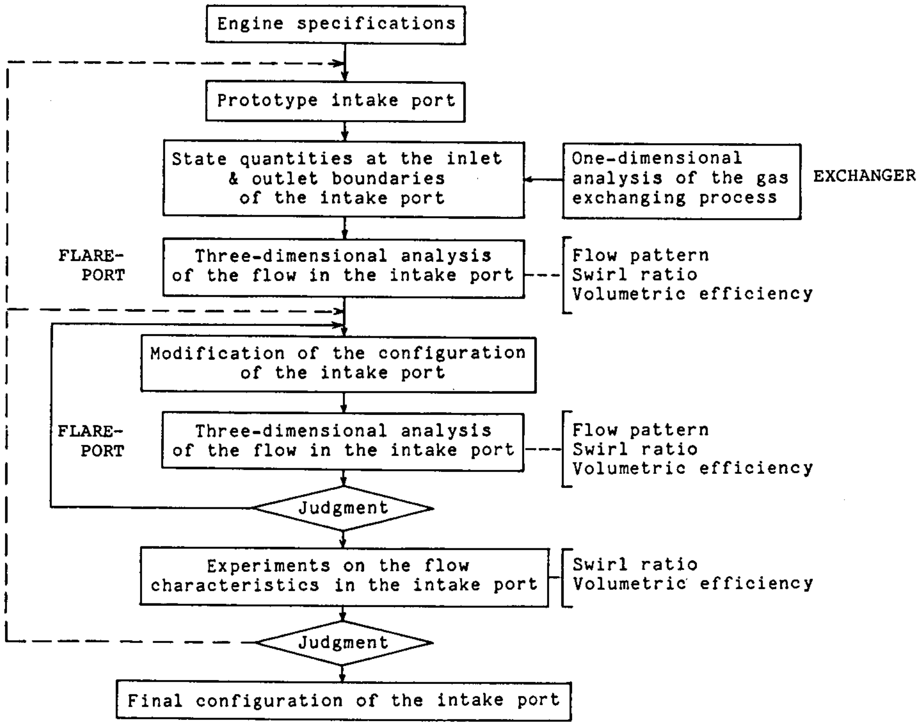


Fig. 1 Procedure of the determination of the intake port configuration with the technique of the numerical analysis of gas flows

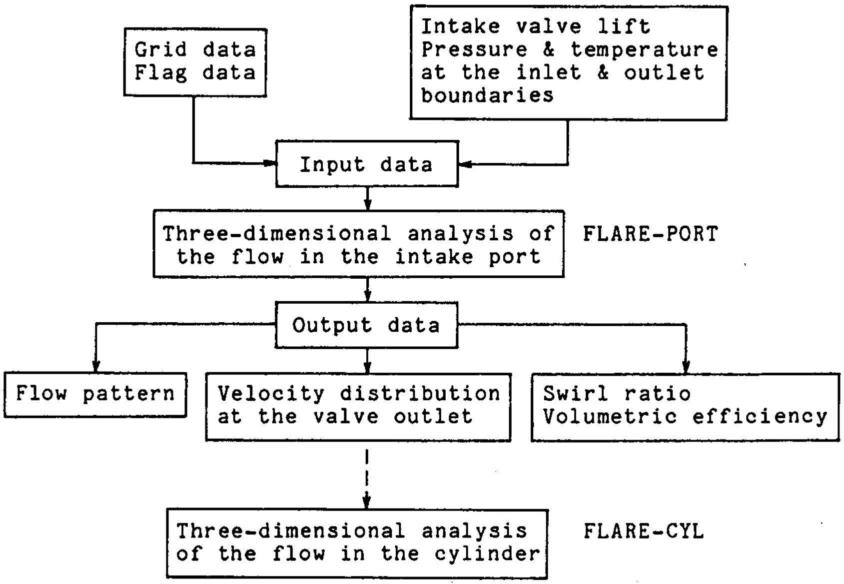


Fig. 2 Procedure of the three-dimensional numerical analysis of the gas flows in the intake port and cylinder

crank angle.

(iii) A three-dimensional numerical analysis of the gas flow in the intake port is performed using the results obtained in step (ii) as the boundary conditions. Therefore, the flow in the intake port can be analyzed in the unsteady state which approximates the actual engine conditions. From the calculated results, the flow pattern in the port, the swirl ratio and the volumetric efficiency are obtained.

(iv) The configuration of the prototype port is modified. On this occasion, the configuration can be easily changed with this method in which the regions of the gas and wall are discriminated by flags representing the configuration of the port.

(v) With the modified port, the three-dimensional calculation of the flow is carried out to obtain the swirl ratio, etc. Here, it is assumed that the boundary conditions calculated in step (ii) for the prototype port may be applied also to the modified port.

By repeating steps (iv) and (v) in the above procedure, the relationship between the port configuration and the flow characteristics is investigated so that the configuration of the intake port, which possesses the desired characteristics, is decided.

(vi) With the port thus decided, it is necessary to investigate experimentally the flow characteristics in order to ascertain the quality of the port. If the characteristics are confirmed to be the desired ones, it follows that the final configuration of the intake port is obtained.

(vii) Next, if it is required to investigate the swirl behavior in the cylinder, the three-dimensional numerical analysis of the in-cylinder flow during the intake stroke is performed, using the pattern of the velocity distribution around the intake valve and other quantities calculated in the above steps. (Fig. 2)

Examining the port configuration through this procedure can reduce the number of the ports which have to be experimentally tested. This seems to save effectively the time and labor necessary for developing an intake port.

3.2 One-Dimensional Analysis of Gas Exchanging Process

In the code "EXCHANGER"[15], the intake and exhaust systems, including the cylinder, are modeled as the connections of pipes, vessels and throttles. The state of gas in each vessel is assumed to be uniform. Each pipe is approximately represented by a circular pipe whose cross-section changes smoothly. The whole process of the gas exchange is calculated by the one-dimensional characteristic method. This code can treat the gas exchanging process in a practical multi-

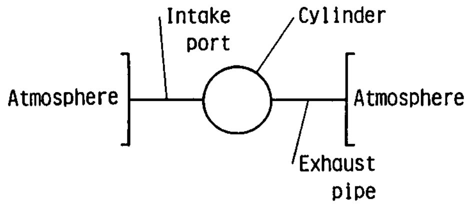


Fig. 3 One-dimensional model of intake and exhaust systems

cylinder engine, that is, the process in the intake and exhaust systems equipped with an air cleaner, silencer, muffler, etc. However, a simple model equipped with only an intake port and a short exhaust pipe, each of which faces the atmosphere as shown in Fig. 3, is considered in this paper.

3.3 Three-Dimensional Analysis of Gas Flow

Both of the codes "FLARE-PORT" and "FLARE-CYL" employ the following method of analysis:

The gas in the calculation region is assumed to be a compressible viscous fluid and is regarded as air conforming to the ideal gas law. The conservation equations for the mass, momentum and energy of the gas are discretized by the finite volume method. Using the SIMPLE algorithm[16], these discretized equations are solved iteratively to obtain the spatial distributions of the three velocity components, pressure, temperature and density. The turbulent viscosity has to be determined on the basis of some appropriate turbulence model. In order to simplify the calculation procedure, the authors have chosen to employ, for convenience, the turbulent viscosity determined by the formulation in the Sub-grid Scale model, though the flow simulation in this study does not belong to the category of the Large Eddy Simulation because of the relatively coarse grids used in the calculation. The details of the method of analysis have been described in Refs. 13) and 14).

In the code "FLARE-PORT", the Cartesian coordinates (x, y, z) are used as shown in Fig. 4, and the wall boundaries of an intake port (Fig. 4 (a)) are approximately represented by staircase-like boundaries (Fig. 4 (c)) in the following way:

Firstly, the three-dimensional grid (Fig. 4 (b)), which covers the intake port to be investigated, has to be prepared. Then, with regard to all of the rectangular prisms (the control volumes) formed by the grid lines, it is required to input the flag data which determine whether each control volume is within the gas region or within the wall region. In the case of the grid shown in Fig. 4 (b), the grid spacing in the y-axis direction (the region of negative y) at the induction part is made different from that at the helical part for a better representation of the configuration.

The grid spacing and flag data are the input data concerning the port configuration. Also, the valve lift curve has to be given (Fig. 2). For the boundary conditions at the port inlet, the gas temperature and total pressure is prescribed when the gas flows into the port, while the static pressure are prescribed when the gas flows out of the port inlet. With respect to the boundary conditions at the valve outlet, the gas pressure and temperature in the cylinder are given. The swirl ratio and the volumetric efficiency are calculated from the velocity distribution and density obtained at the valve outlet.

In the code "FLARE-CYL", cylindrical coordinates are used. As for the inflow boundary conditions, given in this code are the pattern of the velocity distribution around the intake valve (calculated by means of "FLARE-PORT") and the gas pressure, temperature and mass flow rate at the valve boundary

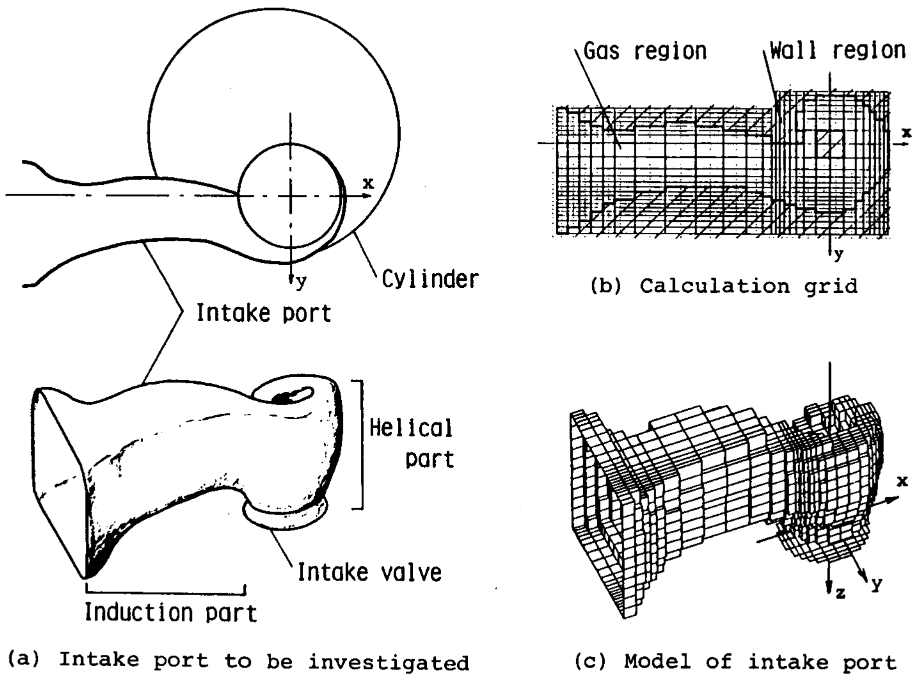


Fig. 4 Modeling of intake port

(calculated by means of "EXCHANGER").

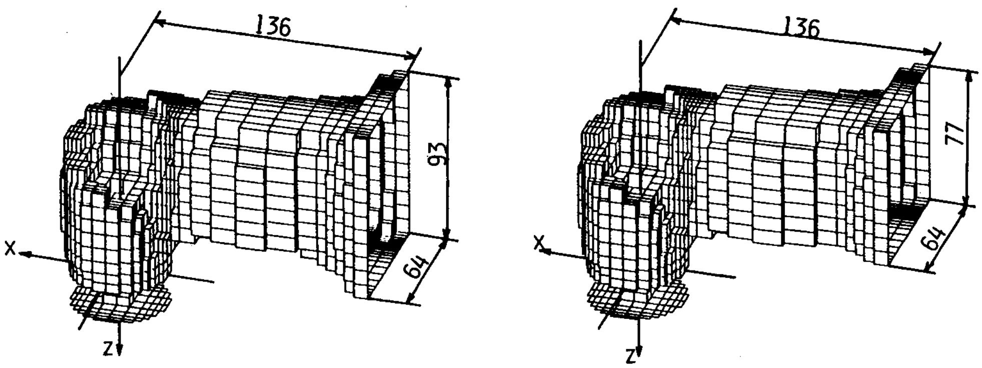
The above-mentioned method where each region of the intake port and cylinder is treated separately is an approximate one, unlike the method where the calculation is performed simultaneously in the whole region. In the former method, for example, the effect of the cylinder wall on the flow characteristics at the valve outlet is neglected. In view of practical use, however, it seems that the method in this paper is preferable.

4. Results of Calculation

The main specifications of the high-speed four-stroke cycle diesel engine to be investigated are shown in Table 1. A helical port model (designated Model P) shown in Fig. 5 (a) is employed as the prototype of an intake port model. The variation of the flow characteristics owing to the modification of the port configuration is numerically calculated.

Table 1 Main specifications of model engine

Bore	130 mm
Stroke	150 mm
Compression ratio	15.6
Intake valve opens	18° BTDC
Intake valve closes	52° ABDC
Intake port diameter at valve seat	52 mm
Intake valve seat angle	30°



(a) Model P (b) Model HP

Fig. 5 Intake port models

4.1 Examination of Swirl Ratio

Model HP (Fig. 5 (b)), whose inlet shape is a little different from Model P, is used here. The number of control volumes in the gas region is about 3200. The flows in the intake port and cylinder (the piston head is flat) during the intake process are calculated under the motoring condition at $n = 30\,s^{-1}$, using the method described in Section 3.

The cylinder pressure calculated by "EXCHANGER" is shown in Fig. 6 with the valve lift. Using these data, the three-dimensional gas flow in the intake port of Model HP is calculated. Then, the three-dimensional flow in the cylinder is analyzed, using the pattern of the velocity distribution around the intake valve as the boundary condition. As an example of the calculated velocity distributions, the projected velocity vectors in the port and cylinder at $\Theta = 102°$ are shown in Figs. 7 and 8. It is found from Fig. 7 that the rotating flow is generated in the helical part of the port. Consequently, the projected velocity vectors at the valve outlet contain the tangential component. This fact plus the non-uniform distribution of the radial velocity component result in generating the swirl. In fact, it is revealed from Fig. 8 that the counter-clockwise swirl is generated in the cylinder though the flow field is complicated.

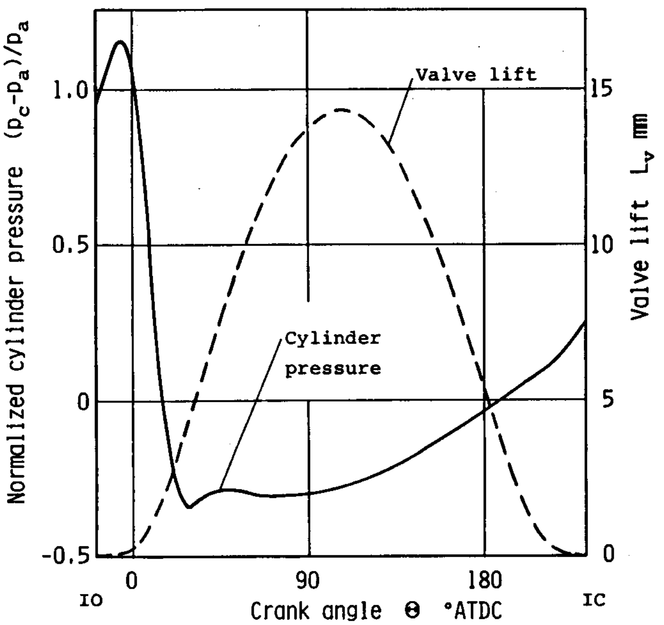


Fig. 6 Cylinder pressure and intake valve lift (Model HP, $n = 30\,s^{-1}$)

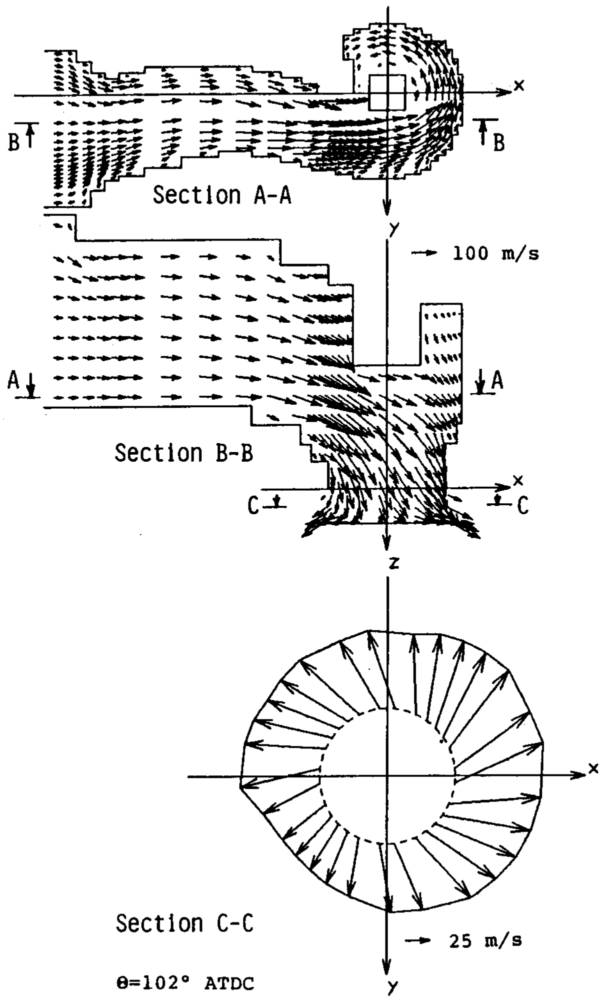


Fig. 7 Flow velocity distribution in the intake port (Model HP)

The swirl ratio should be calculated from the in-cylinder flow field. From a practical point of view, however, it is preferable to evaluate the swirl ratio from the flow characteristics at the intake valve outlet when the effect of the intake port configuration on the swirl intensity is investigated. Therefore, the swirl ratio S_C to be calculated from the flow characteristics at the valve outlet is defined in addition to the swirl ratio S_{cyl} to be calculated from the in-cylinder flow field. The swirl ratio S_C at any crank angle Θ is defined as

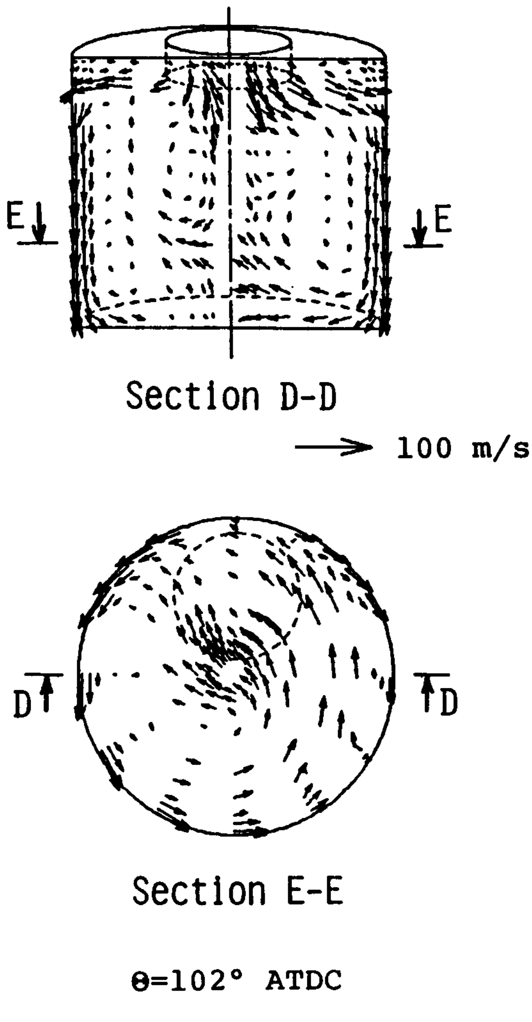


Fig. 8 Flow velocity distribution in the cylinder (Model HP)

$$S_C = \int_{\Theta_{IO}}^{\Theta} \frac{\pi}{180\omega_e} \Omega_C \, d\Theta \Big/ \frac{1}{2} R_c^2 M \omega_e \qquad (1)$$

where Ω_C : moment (about the cylinder axis) of the momentum flux of the jet issuing from the intake valve outlet at Θ

Θ_{IO} : timing of intake valve opening

ω_e : engine angular velocity

R_c : cylinder radius

M : initial mass of the gas at Θ_{IO} plus the mass of the gas flowing into the cylinder through the intake valve in the period from Θ_{IO} to Θ.

The swirl ratio S_{cyl} at any crank angle Θ is defined as

$$S_{cyl}=\int_0^{\xi_p}\int_0^{2\pi}\int_0^{R_c}\rho r^2 v_\theta dr d\theta d\xi/\omega_e\int_0^{\xi_p}\int_0^{2\pi}\int_0^{R_c}\rho r^3 dr d\theta d\xi \quad (2)$$

where r, θ, ξ : radial, circumferential and axial coordinates

ξ_p : distance between the cylinder head and the piston head at Θ

v_θ : circumferential velocity component in the cylinder at Θ

ρ : density at Θ.

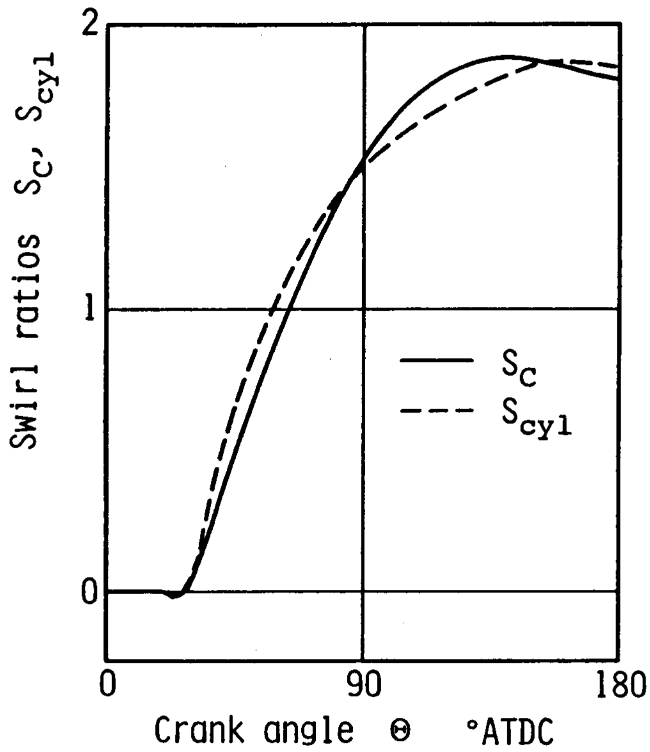


Fig. 9 Swirl ratios (Model HP)

Figure 9 shows the swirl ratio S_C calculated from the result of the flow analysis for Model HP and the swirl ratio S_{cyl}. This S_{cyl} is calculated from the in-cylinder flow field which is obtained by using the pattern of the velocity distribution around the intake valve for this intake port model. The fact that S_{cyl} is almost the same as S_C seems to mean that the swirl angular momentum in the cylinder scarcely decays during the intake process. Therefore, it may be said that the swirl ratio at the end of the intake process can be evaluated from the flow characteristics at the intake valve outlet.

4.2 Examination of the Effect of Port Configuration on the Flow Characteristics

Various intake port models are prepared by modifying systematically the configuration of the prototype port Model P (Fig. 5(a)). The number of control volumes is about 3000. Calculations are performed for these models under the motoring condition at $n = 30\,\mathrm{s}^{-1}$ to investigate the swirl ratio S_C and volumetric efficiency η_v at a crank angle just before the timing of the intake valve closing. The computation for one case (Model P) took about 7 minutes (CPU time) on a FACOM VP-200 computer at the Data Processing Center of Kyoto University.

4.2.1 Effect of the configuration of induction part

As shown in Fig. 10, four models, P-I1 through P-I4, are formed by gradually changing the configuration (side wall curve) of the induction part of Model P. The normalized swirl ratio S_C^* and the normalized volumetric efficiency η_v^* in the figure are based on the values for Model P, respectively. The value of η_v^* becomes smaller in the order of Models P-I1, P, P-I3, P-I2 and P-I4. This order corresponds approximately to the magnitude of the curvature of the induction part in each model. The normalized swirl ratio S_C^* becomes smaller in the order of Models P-I1, P, P-I3 and P-I2, whereas it becomes largest in the case of Model P-I4.

Actually, the cylinder pressure is affected by the change of the intake port configuration. In Section 4.2, however, the cylinder pressure for the case of Model P is used as the common boundary condition at the intake valve outlet for analyzing the three-dimensional flows in the intake port models having different configurations. Accordingly, the calculated volumetric efficiency deviates from the actual one, though it is possible to predict the tendency of the variation of the volumetric efficiency owing to the change of the intake port configuration. Since the volumetric efficiency affects the swirl ratio, it is necessary to eliminate the effect of the volumetric efficiency from the swirl ratio when

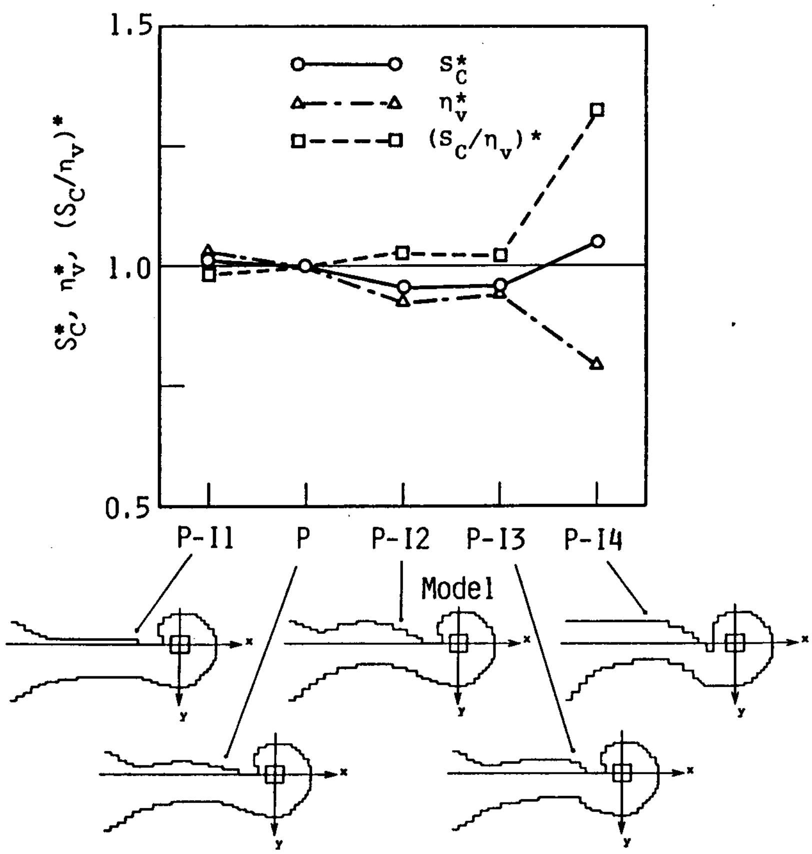


Fig. 10 Effect of the configuration of induction part on the flow characteristics at the valve outlet

only the effect of the intake port configuration on the swirl intensity is examined. The swirl ratio S_C is considered to be approximately proportional to the volumetric efficiency η_v. Therefore, the modified swirl ratio S_C/η_v is used tentatively in order to eliminate the effect of η_v. The normalized values of the modified swirl ratio, $(S_C/\eta_v)^*$, are shown in Fig. 10. The difference in $(S_C/\eta_v)^*$ among Models P, P-I 1, P-I 2 and P-I 3 is small in spite of the difference in the configuration. In contrast with these models, Model P-I 4 exhibits a considerably larger value of $(S_C/\eta_v)^*$.

4. 2. 2. Effect of the height of helical part

Shown in Fig. 11 are the calculated results for four models, P-H 1 through P-

H 4, which are formed by gradually lowering the height of the helical part of Model P in the order of Models P-H 1, P-H 2, P-H 3 and P-H 4. The value of η_v^* decreases while S_c^* increases with a lowering of the helical part height. Consequently, $(S_c/\eta_v)^*$ has a larger increase.

Thus, it has been found that the change of the helical part height causes a much greater variation of the swirl ratio and volumetric efficiency than the

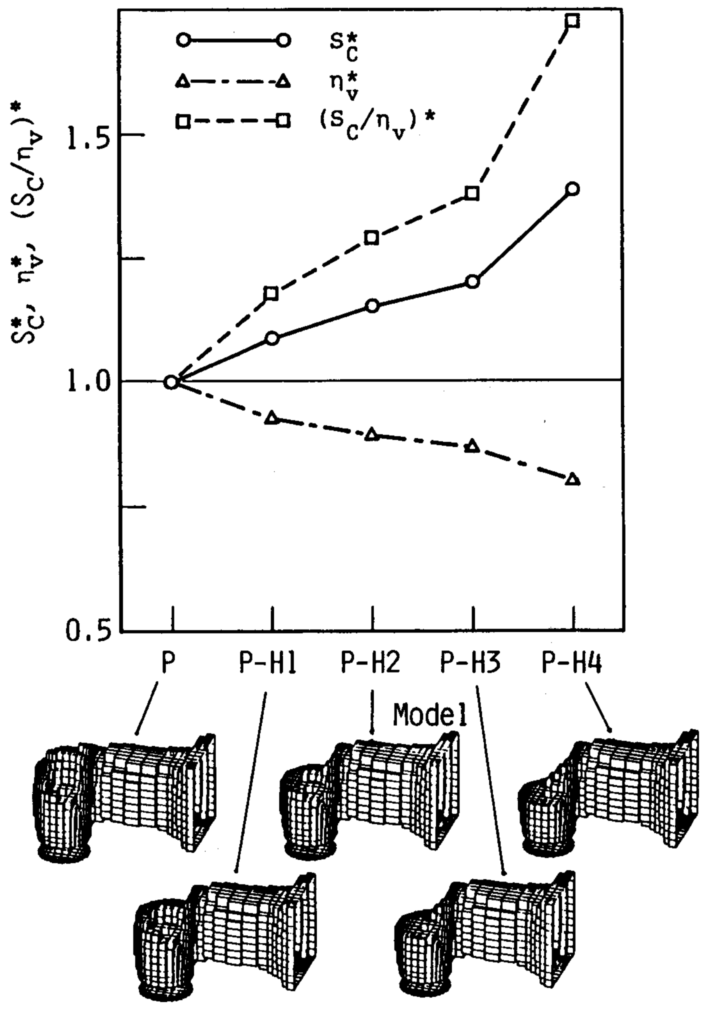


Fig. 11 Effect of the height of helical part on the flow characteristics at the valve outlet

change of the induction part configuration causes one.

4.2.3. Effect of the side wall configuration at helical part

As shown in Figs. 12 – 14, twelve models, namely, Models P-HA 1 through P-HA 4, Models P-HB 1 through P-HB 4 and Models P-HC 1 through P-HC 4 are shaped from Model P by moving each side wall at the quarter sectors A, B and C, respectively, close to or away from the valve axis.

In Fig. 12, as the side wall at Sector A is moved gradually closer to the valve axis with Model P-HA 1 as a starter, η_v^* decreases little by little, while S_c^* and

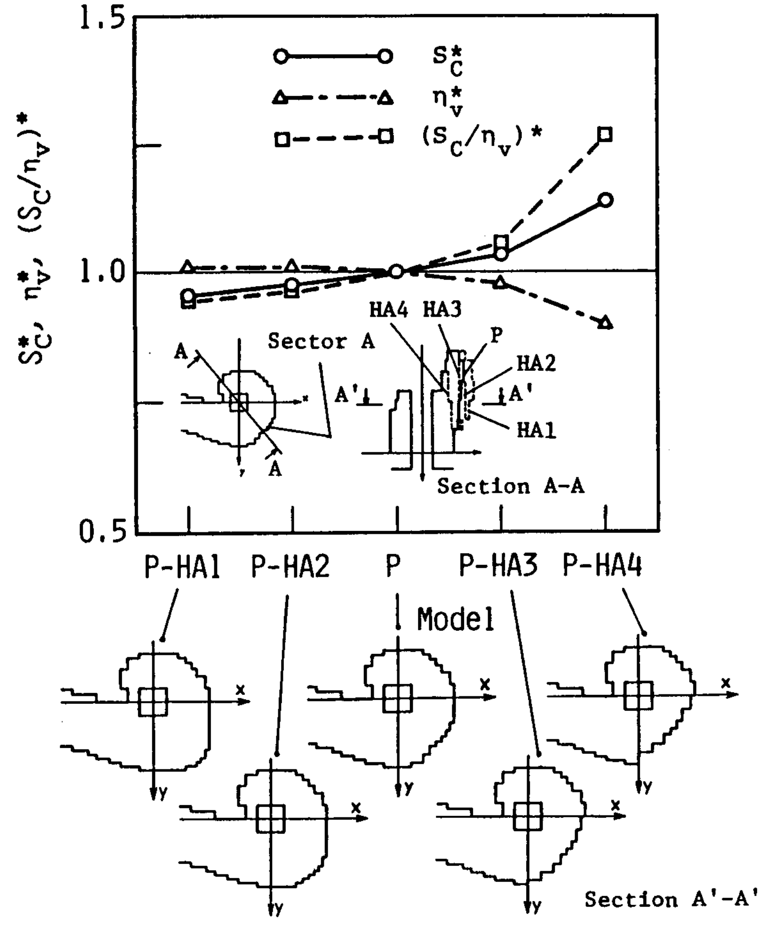


Fig. 12 Effect of the side wall configuration at Sector A on the flow characteristics at the valve outlet

$(S_C/\eta_v)^*$ become larger. In Figs. 13 and 14, as the side wall at Sector B or C is moved gradually closer to the valve axis in the same way as above, η_v^* for either case scarcely varies. The values of S_C^* and $(S_C/\eta_v)^*$, however, vary with the approach of the side wall to the valve axis. In the P-HB Models, S_C^* and $(S_C/\eta_v)^*$ become a little larger, while in the case of the P-HC Models, they tend to slightly decrease.

From the above-mentioned results, it has been found that in the case where the side wall configuration at Sector A is changed, the swirl ratio and the

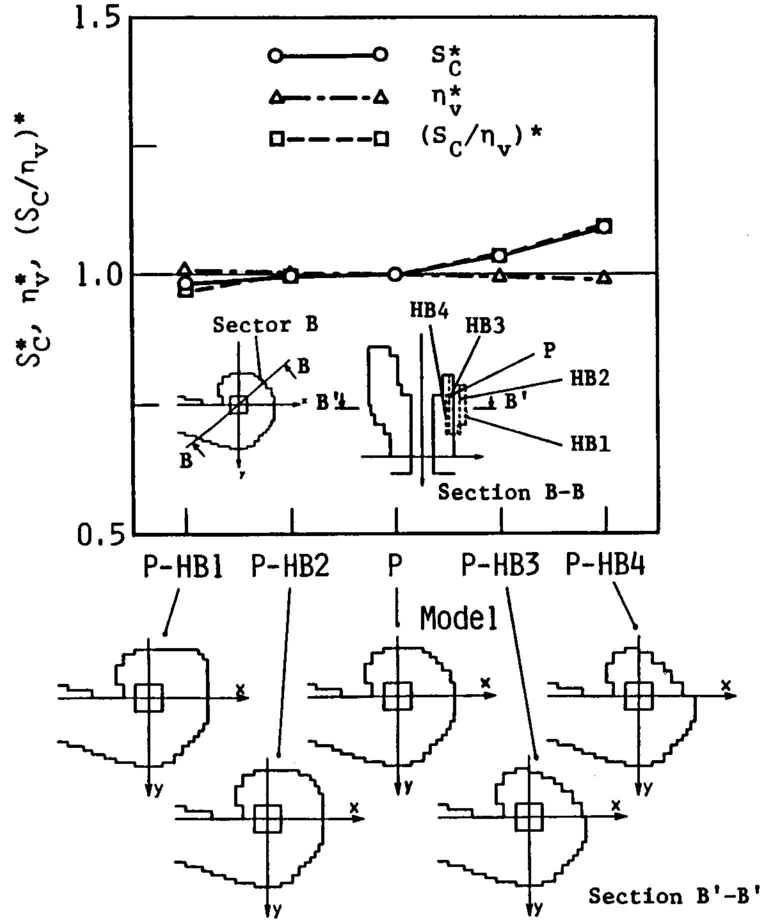


Fig. 13 Effect of the side wall configuration at Sector B on the flow characteristics at the valve outlet

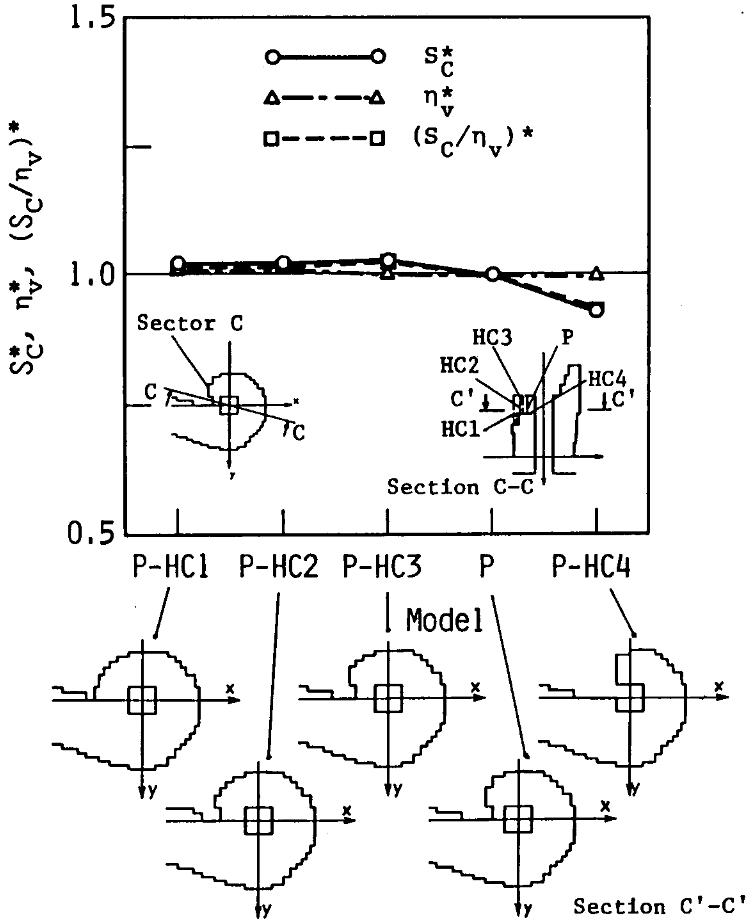


Fig. 14 Effect of the side wall configuration at Sector C on the flow characteristics at the valve outlet

volumetric efficiency vary considerably in comparison with cases where the side wall configuration at Sector B or C is changed.

5. Conclusions

A practical method of analysis has been presented for numerically predicting the three-dimensional gas flow characteristics such as swirl during the intake process in four-stroke cycle engines. In this method, the one-dimensional calculation of the gas exchanging process is executed under actual engine conditions.

Then, using these calculated results as the boundary conditions, the three-dimensional numerical analysis of the flow in the intake port is carried out separately from the three-demensional analysis of the in-cylinder flow.

It has been found that the swirl ratio may be evaluated with this method from the flow characteristics at the intake valve outlet. The effect of the port configuration both on the swirl intensity and on the volumetric efficiency has been demonstrated with regard to a helical port.

With this method, it may be feasible to assess relatively the effect of the intake port configuration, which is one of the important design parameters of an engine, on the flow characteristics during the intake process. The behavior of the swirl in the cylinder during the intake stroke may be also practically predicted.

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