A Numerical Study of the Effects of Piston Head Configurations on Stratified Mixture Formation in Gasoline Direct-Injection Engines

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In this paper, the characteristics of flow and spray motions affected by from piston head configurations were investigated numerically. Calculations were carried out from intake process to the end of compression. GTT (Generalized Tank and Tube method) code, which includes a third order upwind Chakravarthy-Osher TVD scheme and κ-ε turbulence model with fuel spray analysis was used for the calculations. As a results, piston heads with smaller radii of curvature were found to give stronger reverse tumble than those with larger radii of curvature. Similar results are shown in the convection and diffusion of fuel sprays.

Key Words: Gasoline Direct-Injection, Stratified Mixture, Piston Head Configurations, Reverse Tumble, Fuel Spray

1. Introduction

In the last decade, a variety of approaches have been put forward to realize stratified combustion in spark-ignition engines and as a result, direct-injection gasoline engines have been developed (Schapertons et al., 1991, Shiraishi et al., 1995, Shimotani et al., 1995). In the direct-injection gasoline engines, many parameters affect the processes of mixture formation and combustion, such as combustion chamber geometry, engine speed, engine load, overall air-fuel ratio, fuel injection timing and ignition timing. The effects of the parameters on the global phenomena are very complicated which need tremendous effort to characterize. As most of the parameters play a key roles in establishing the optimal global phenomena at the end of the compression stroke for late injection, the characteristics of gas flow and fuel sprays at the end of compression must be examined in detail.

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In this paper, the characteristics of flow and spray motions affected by the configurations of the piston head were investigated numerically. Dual intake-ports were set as straight ports for reverse tumble, and the curvatures of piston-cavities were changed. Flow calculations were performed from the intake process to the end of compression. The characteristics of flow are examined in detail near the end of the compression process, including the spray motions.

2. Numerical Analysis

2.1 Gas flow

Computations of gas flow were carried out using the GTT code (Wakisaka et al., 1990), based on the finite volume and fully implicit discretization methods with generalized curvilinear coordinates. Pressure-velocity coupling is accomplished using the SIMPLEC algorithm (Van Doormaal and Raithby, 1984). Higher-order schemes are essential as discretization schemes for the convection terms of the governing conservation equations in order to obtain accurate results with less numerical diffusion. In this study, the third order upwind Chakravathy-Osher TVD scheme was applied for the momentum transportation equations, and the CIP (Cubic Interpolated pseudo-Particle) scheme (Yabe and Takei, 1988) was used for the transportation equation of fuel vapor concentration. Concerning the turbulence model, the κ-ε turbulence model was used for calculating the spray behavior. However, the turbulent viscosity calculated using this turbulence model was not used to calculate the gas velocity, while the laminar viscosity was used as the viscosity in the Navier-Stokes equations. The reason for this is that if the turbulent viscosity is used, small vortices are not reproduced, because of its over-estimated viscosity. As the wall boundary conditions for velocity, enthalpy, κ and ε, the law of wall is adopted. No slip boundary conditions were specified in all boundaries except the piston surface for the velocities. Along the piston surface the velocity of fluid is set equal to the piston velocity. The validity of the above-mentioned method of calculating gas velocity was confirmed by comparing the calculated results of the velocity field in an engine cylinder with the experimental ones. (Wakisaka et al., 1995)

2.2 Spray model

Liquid fuel sprays were numerically analyzed using the Discrete Droplet Model (DDM), as in the KIVA code (Amsden et al., 1985). The bag and stripping breakup model by Reitz et al. (1987) was used as a droplet breakup model and the models of droplet coalescence and evaporation in the KIVA code were used. Concerning the interaction between the gas phase and the fuel droplets, the vaporized mass, momentum and enthalpy lost by the droplets are given to the source terms in their respective conservation equations as to the gas phase. The model proposed by one of the authors (Wakisaka et al., 1994) is incorporated into the GTT code as a model for the impingement of fuel sprays on the wall. A high-pressure swirl injector, which generates hollow-cone sprays, was used as an injector. A schematic of the nozzle outlet is shown in Fig. 1. The liquid fuel in the injector undergoes swirling motion upstream from the nozzle hole, and this swirling motion causes the liquid to form a liquid film on the surface of the hole. Then, the liquid fuel is injected with a swirl velocity component and a cone of liquid sheet is formed. Since the swirl injector generates very small droplets in the downstream, it is inferred that this liquid sheet is very thin and breaks up rapidly. Other researchers have reported that the breakup length of the liquid sheet is very short (Iwamoto et al., 1997) which was confirmed by an experiment that the breakup length of the sheet lies in the range between 1mm and 2mm (Yamauchi et al., 1998). The injection cone angle and swirl angle were set as 60 and 30 degrees, respectively, in this study. These conditions were determined on the bases of the experimental results and some estimations in the previous study (Yamauchi and Wakisaka, 1996) so that the spray characteristics may coincide with the measured ones.

Table 1 shows major specifications and operating conditions of an engine used for the calculations. The fuel supply pressure was set at 5MPa, and the pressure and temperature of ambient gas (air) were set at 0.1MPa and at room temperature, respectively. The amount of injected fuel

Table 1 Specifications and operating conditions for the calculation

Engine type	4-stroke, 4-valve
Bore, Stoke	78mm, 70mm
Compression ratio	9.8
Fuel pressure	5 MPa
Engine speed	1500 rpm
Overall A/F ratio	34/1

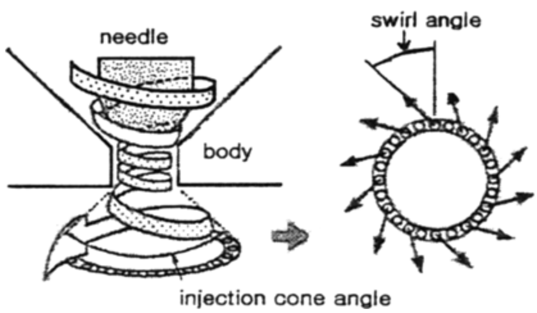


Fig. 1 Schematic of the Nozzle Outlet

mass is 13mg/stroke and injection duration 1.1 msec. The grid system for the calculation is shown in Fig. 2. To realize the reverse tumble, an upright straight intake-port is selected, and the spark plug is located in the center of the cylinder. The test types of piston head are shown in Fig. 3. The cavities of the piston head have different radii of curvatures with the same top-view. The radii of curvatures are 0.025m, 0.03m and 0.035m, respectively. The cavity of type 1 is deeper than that of the others. The computations are carried out during intake stroke (from TDC to BDC) at an engine speed of 1500rpm, and the results of the calculation were used as initial values for the grid system of the compression stroke.

The number of grid points was $41 \times 41 \times 37$ for the intake stroke and $41 \times 41 \times 17$ for the compression stroke. The time increment for the velocity calculation was set at 0.2 msec except for the early stage of the compression stroke. The droplet behavior and the fuel vapor concentration were calculated explicitly in the subcycles with a smaller time increment of 0.02 msec; the number of injected droplet parcels was 8000.

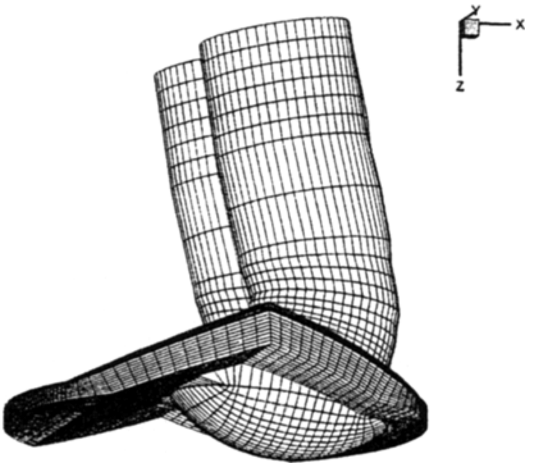


Fig. 2 Computational grid $(41 \times 41 \times 17)$

(a) Type 1

(b) Type 2

(c) Type 3

Fig. 3 Types of piston head

3. Numerical Results and Discussions

Figure 4 shows the velocity vectors of gas flow induced by a straight port in intake process (at 60° ATDC). It was observed that the reverse tumble was formed in cylinder. A reverse tumble induced by a straight port on the intake stroke is shown in Fig. 5. However, the reverse tumble ratio is rather low and the counter clockwise

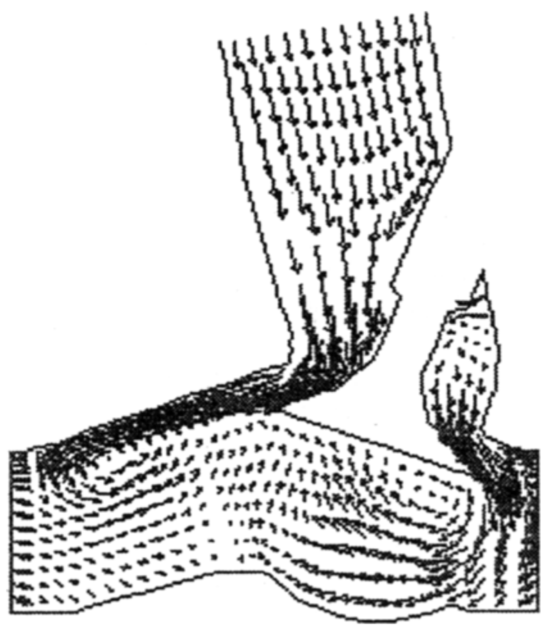


Fig. 4 Velocity vectors for the type 2

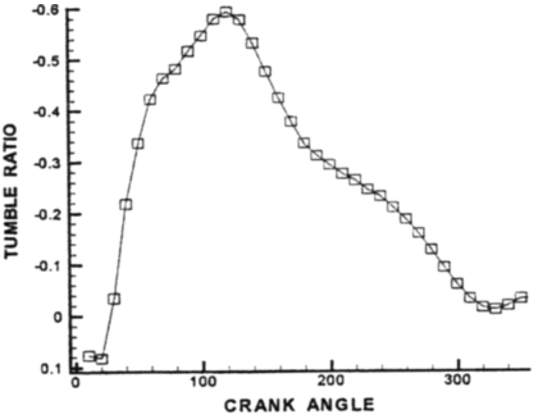


Fig. 5 Tumble ratio for the type 2

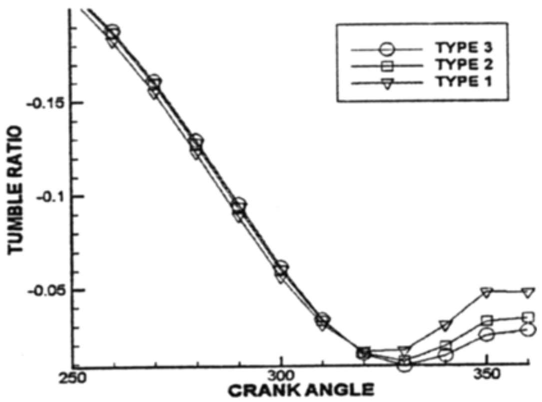


Fig. 6 Tumble ratios for three types

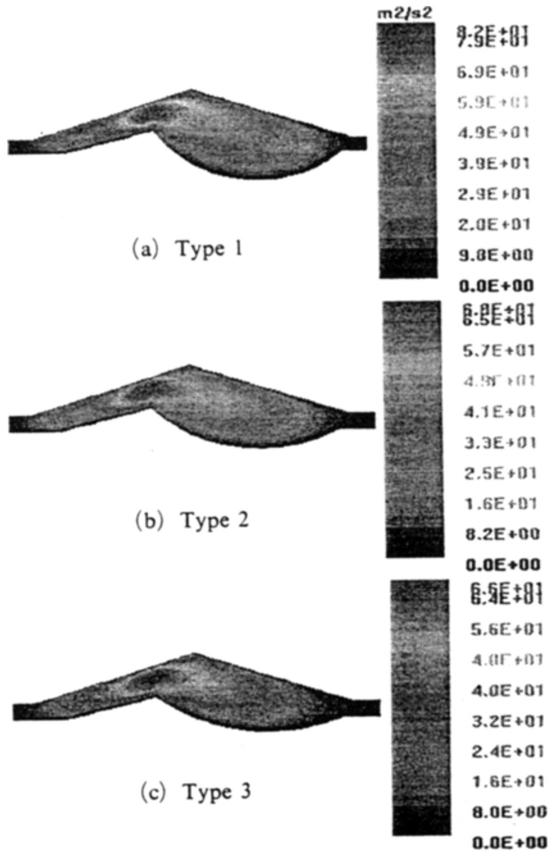


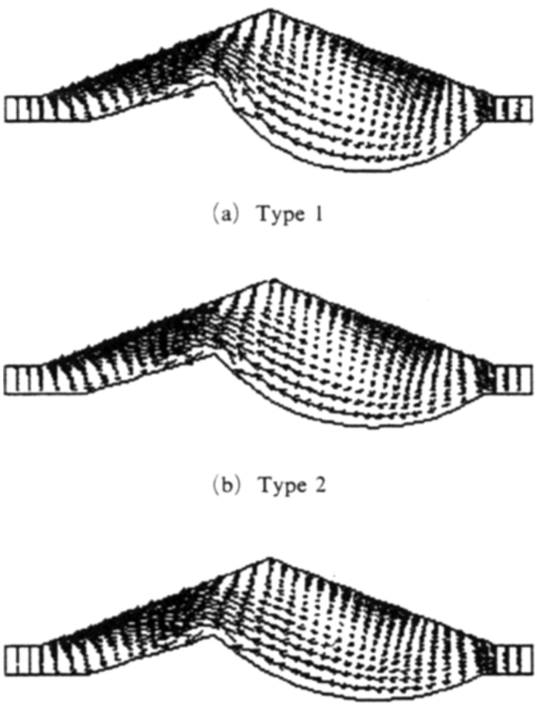
Fig. 7 The distributions of turbulent kinetic energy (at BTDC 10°)

tumble is shown at the beginning of the intake stroke, which means that the intake port is not designed optimally. It has been reported that the maximum reverse tumble ratio reaches 1.8 when the intake port is designed optimally (Kume et al., 1996). Putting the optimal design of the intake port aside, the trends of tumble ratio can be observed qualitatively.

Figure 6 shows that tumble ratios increase near the end of the compression stroke for the three types of piston head. Moreover, the deeper bowl enhances stronger tumbles in the bulk motion of the gas flow, which means the shape of the cavity parallel to the tumble motion has the advantage of producing a stronger tumble motion.

Quantitatively increasing amounts are small, but differences of tumble ratios depending on the types of piston head can be increased when the intake port is designed optimally.

As shown in Fig 7, the distributions of turbulent kinetic energy are also similar in all cases but values are higher in the case of Type 1. Figure 8 shows the gas flow at the 350 crank angle of degrees for three types of piston head, characterized by reverse tumble. Result of bulk motion can be confirmed by the behaviors of the fuel sprays. Figure 9 shows spray shapes for the type 2 of piston head. Figure 9(a) shows that injected fuel droplets generate a toroidal vortex due to the interaction between the spray and the bulk flow. As shown in Fig. 9(b), the fully developed spray, which is accompanied by the toroidal vortex, impinges on the piston cavity and is captured completely in the cavity. After the spray impinges on the cavity, the diffusion and convection of the air-fuel mixture proceed. This stage is shown in Fig. 9(c), where the number of fuel droplets

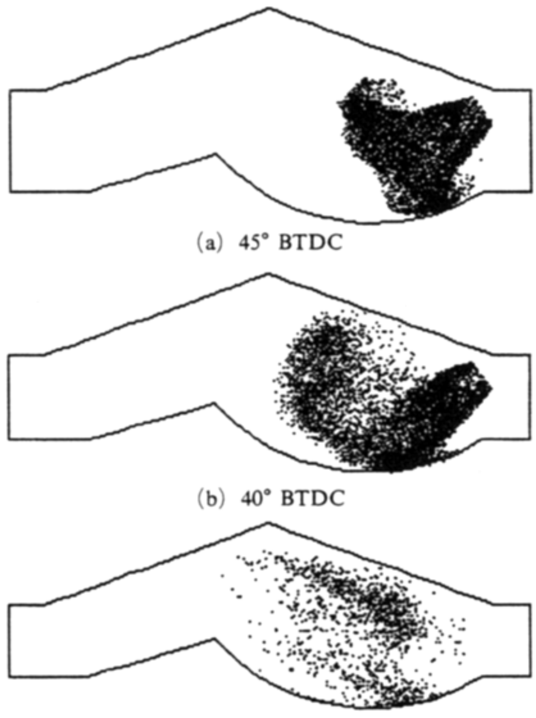


(a) Type 1

(b) Type 2

(c) Type 3

Fig. 8 Velocity vectors for three types (BTDC 10°)

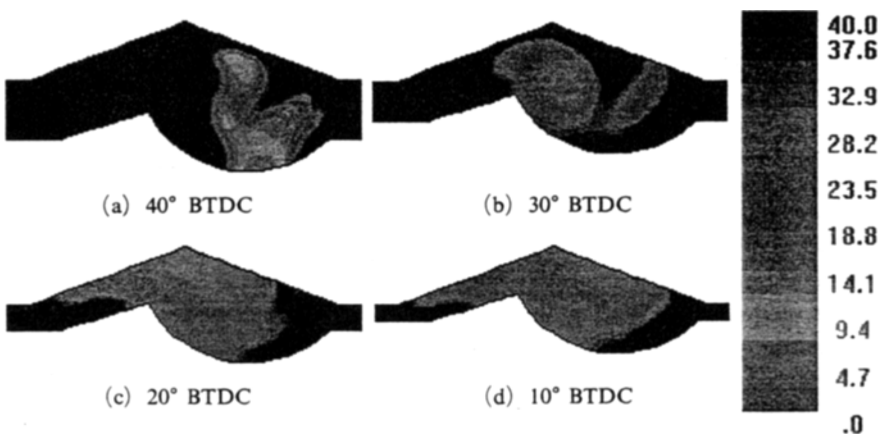


(a) 45° BTDC

(b) 40° BTDC

(c) 35° BTDC

Fig. 9 Calculated results of spray shapes for the type 2



(a) 40° BTDC

(b) 30° BTDC

(c) 20° BTDC

(d) 10° BTDC

Fig. 10 Contours of the air/fuel ratio in the vertical section with type 1 piston

significantly decreases due to evaporation. These processes of stratified mixture formation are clearly seen in Figs. 10, 11 and 12. These figures show the air/fuel ratios for each case. Fuel sprays are injected at 50° BTDC and completed at 40° BTDC. As in the previous description, the fully developed spray with a toroidal vortex impinges on the piston cavity and is captured in the cavity (at 40° BTDC). The toroidal vortex also rolls up toward the spark plug along the wall of cavity. If the cavity does not have such side wall as circular bank, it is inferred that the spray will spread over the whole piston cavity and the fuel vapor concentration will be weakened. Hence, it should be

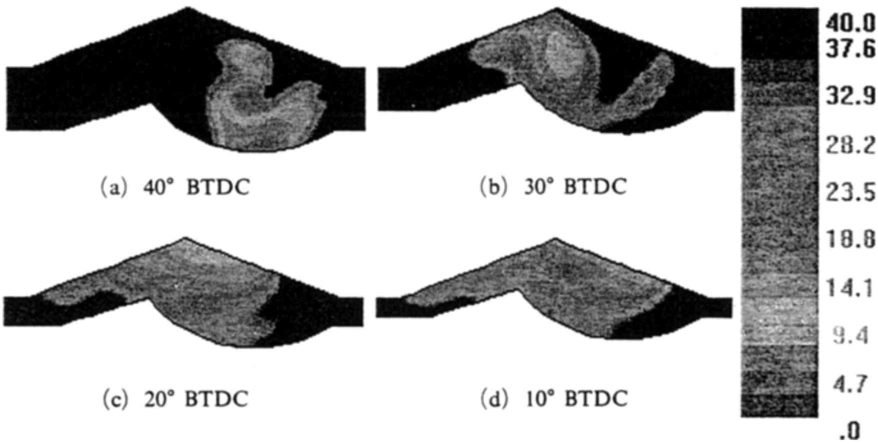


Fig. 11 Contours of the air/fuel ratio in the vertical section with type 2 piston

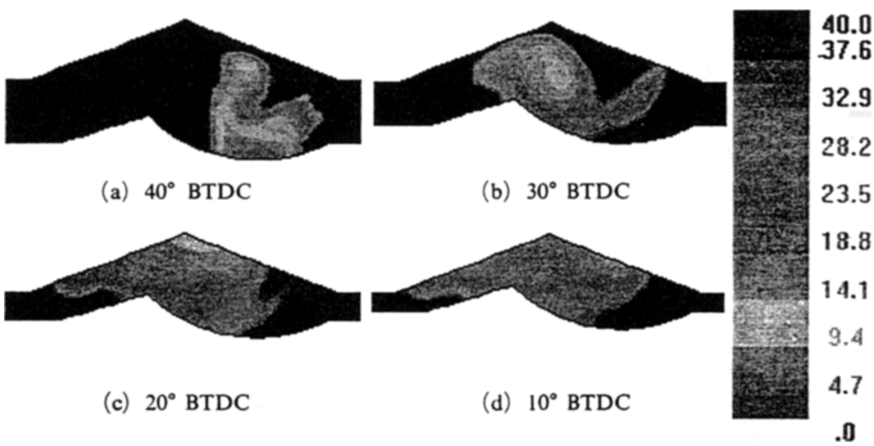


Fig. 12 Contours of the air/fuel ratio in the vertical section with Type 3 piston

noted that the cavity shape controls the outline of the spatial distribution of the air-fuel mixture. After the spray impinges on the piston cavity, the diffusion and convection of the air-fuel mixture proceed along the bulk motion of gas flow (at 30° BTDC, 20° BTDC and 10° BTDC).

In this stage, although the mixture volume becomes smaller due to compression, the overall fuel vapor concentration throughout the mixture region becomes leaner due to the diffusion, and a relatively rich mixture transported by the tumble flow is formed around the spark plug. Thus, the spatial distribution of the mixture, which is very important for stratified combustion, is largely influenced by the tumble flow. As shown in Figs. 10, 11 and 12, it was observed that the richest region of the air-fuel mixtures move following the tumble motion for all types, while the type 1 show stronger clockwise motion and the type 3 show wider spread. However, the formation of a rich mixture around the spark plug is common to all types of piston heads.

4. Conclusions

The characteristics of the flow and spray motions affected by the configurations of the piston head were investigated numerically using the GTT code in this study. By observing the reverse tumble ratios near the end of compression stroke, it was found that the shape of cavity parallel to the tumble motion enhances stronger tumble motion. It was also observed that the convection and diffusion of the mixtures follow the tumble motion, thus the mixtures of the type 1 piston head show stronger clockwise motion, while mixtures of the type 3 head are more spread. Through these computational studies, it was found that the process of stratified mixture formation proceeds mainly through interaction between the bulk flow (especially tumble flow) and spray after impingement. Spray behavior is much influenced by this interaction. Furthermore, spray behavior after impingement on the piston head is greatly influenced by the configurations of the piston head. It is expected that the discussions in this study on the characteristics of the flow and spray motions interacting with the configurations of the piston head can be utilized in designing any other types of direct injection gasoline engines.

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References

Amsden, A. A., Ramshaw, J. D., O'Rourke, P. J. and Dukowicz, J. K., 1985, "KIVA: A Computer Program for Two-and Three-Dimensional Fluid Flows with Chemical Reactions and Fuel Sprays," Los Alamos Report No. LA-10245-MS.

Iwamoto, Y., Noma, T., Nakayama, O., Yamauchi, T. and Ando, H., 1997, "Development of Gasoline Direct Injection Engine," SAE paper 970541.

Kume, T., Iwamoto, Y., Iida, K., Murakami, M., Akishino, K. and Ando, H., 1996, "Combustion Control Technologies for Direct Injection SI Engine," SAE Paper 960600.

Reitz, R. D. and Diwakar, R., 1987, "Structure of High-Pressure Fuel Spray," SAE paper 870598.

Schapertons, H., Emmennnthal K. D., Grabe, H. J. and Oppermann, W., 1991, "VW's Gasoline Direct Injection (GDI) Research Engine," SAE Paper 910054.

Shimotani, K., Oikawa, K., Horada, O. and Kagawa, Y., 1995, "Characteristics of Gasoline In-Cylinder Injection Engine," *Proceedings of the 12th Internal Combustion Engine Symposium*, Japan, pp. 289~294.

Shiraishi, T., Fujieda, M. and Ohsuga, Y., 1995, "A Study of the Mixture Preparation Process on Spark Ignited Direct Fuel Injection Engines," *Proceedings of 8th IPC*, pp. 235~240.

Van Doormaal, J. P. and Raithby G. D., 1984, "Enhancements of the SIMPLE Method for Predicting Incompressible Fluid Flows," *Numerical Heat Transfer*, Vol. 7, pp. 147~163.

Wakisaka, T., Shimamoto, Y., Isshiki, Y., Sumi, N., Tamura, K. and Modien, R. M., 1990, "Analysis of the Effects of In-Cylinder Flows during Intake Stoke on the Flow Characteristics near Compression TDC in Four-Stoke Cycle Engines," *Proc. Int. Symp. COMODIA 90*, pp. 487~492.

Wakisaka, T., Shimamoto, Y., Isshiki, Noda, T., Matsui, A. and Akamatsu, S., 1994, "Numerical Analysis of Spray Phenomena in Fuel Injection Engines," *Proc. Int. Symp. COMODIA 94*, pp. 403~409.

Wakisaka, T., Takeuchi, S. and Shimamoto, Y., 1995, "Numerical Analysis of the Behavior of Tumbling Vortices in Engine Cylinders Using Higher-Order Schemes," *Proc. 1st Asian Computational Fluid Dynamics Conference*, Vol. 1. pp. 227~233.

Yabe, T. and Takei, E., 1988, "A New Higher-Order Godunov Method for General Hyperbolic Equations," *Journal of the Physical Society of Japan*, Vol. 57, No. 8, pp. 2598~2601.

Yamauchi, T. and Wakisaka, T., 1996, "Computation of the Hollow-Cone Sprays from a High-Pressure Swirl Injector for a Gasoline Direct-Injection SI Engine," SAE paper 962016

Yamauchi, T., Wakisaka, T., Kaneko, T., Kaneko, M., Kato, S. and Hosaka, H., 1998, "Numerical Analysis of Stratified Mixture Formation in Direct-Injection Gasoline Engines," *Direkteinspritzung im Ottomotor*, ISBN 3-8169-1685-6, pp. 166~185.