Formation and Destruction of Vortices in a Motored Four-Stroke Piston-Cylinder Configuration

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Abstract

THE effects of the compression ratio, engine speed, bore-to-stroke ratio, and valve seat angle on the turbulent flowfield within an axisymmetric piston-cylinder configuration have been studied by means of an implicit finite difference method which solves the conservation equations of mass, momentum, and energy, and two additional equations for the turbulent kinetic energy and its dissipation rate. The numerical results indicate that the turbulent intensity at top-dead-center of the compression stroke is independent of the rpm and decreases with decreasing compression ratio. The turbulent intensity also increases when decreasing the bore-to-stroke ratio. It has been found that the valve seat angle has the most important effect on the flowfield. A valve seat angle of 45 deg produces two vortical structures which break down and merge by the end of the compression stroke, and persist into the expansion stroke. For smaller valve seat angles there is no vortex breakdown and the vortical structures created during the intake stroke disappear by the end of the compression stroke. The importance of vortical structures on fuel mixing and turbulence levels within an internal combustion engine is also discussed.

Contents

The purpose of this paper is to report some numerical results concerning the effects of the compression ratio, bore-to-stroke ratio, rpm, and valve seat angle on the formation and destruction of vortices, and turbulent levels within an axisymmetric piston-cylinder configuration equipped with a centrally located valve. The calculations reported here were performed with an implicit finite difference algorithm that solves the conservation equations of mass, axial and radial momentum components, and energy. Turbulence has been modeled by means of a two-equation model for the turbulent kinetic energy and its dissipation rate. The motion of the valve has been accounted for by means of the two-domain technique developed in Ref. 1.

Calculations were performed at several engine speeds, i.e., rpm, and showed that as long as the valve seat angle is less than 45 deg (this angle is measured with respect to the engine cylinder centerline) the flowfield in the intake stroke is characterized by the formation of two vortical structures. One of these structures is located at the corner between the cylinder head and the cylinder wall and is referred to as the cylinder vortex. The other vortical structure is located between the valve and the piston and is referred to as the valve vortex. These structures are shown in Fig. 1. This figure presents the two-dimensional velocity field in a piston-cylinder configuration characterized by a bore-to-stroke ratio of 1, a compression ratio of 7, and a valve seat angle of 0 deg. The configuration was operated at 1000 rpm.

Figure 1 shows that both the valve and cylinder vortices are created in the intake stroke. The size of the valve vortex increases during the intake stroke, but is reduced in the compression stroke. By about 240 crankshaft angle deg the valve vortex merges with the cylinder vortex and disappears by the end of the compression stroke, i.e., 360 deg. In the expansion stroke, the piston drives the almost unidirectional flow shown in Fig. 1 at 450 deg. A similar flow exists in the exhaust stroke, except near the exhaust valve and port as indicated in Fig. 1 at 702 deg.

Flow trends similar to those shown in Fig. 1 have been observed at different rpm, compression ratios, and bore-to-stroke ratios as long as the valve seat angle is less than 45 deg. An increase of the engine speed yields higher air velocities. However, the turbulent intensity at top-dead-center of the compression stroke is almost independent of the rpm. This is because the turbulent kinetic energy is a quadratic function of the engine speed, while the mean flow velocity depends linearly on the rpm. Thus, a characteristic fluctuating velocity is of the order of the square root of the turbulent kinetic energy and increases with the rpm. If a length scale equal to the cylinder bore is taken, an eddy diffusivity proportional to the rpm results.

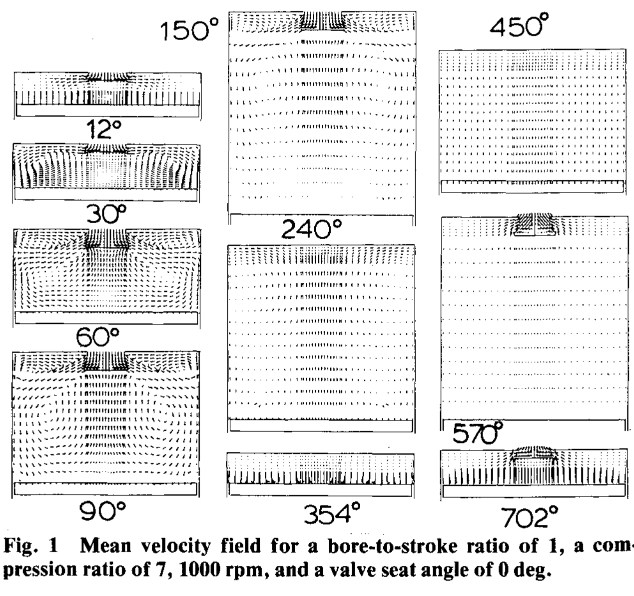

Fig. 1 Mean velocity field for a bore-to-stroke ratio of 1, a compression ratio of 7, 1000 rpm, and a valve seat angle of 0 deg.

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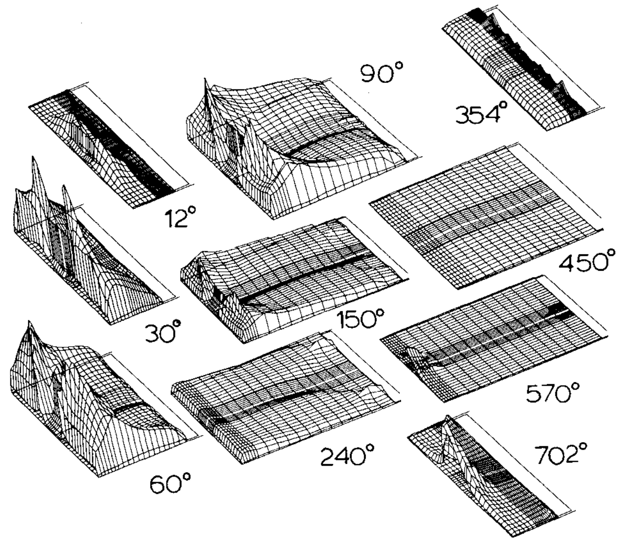

Fig. 2 Turbulent intensity field for a bore-to-stroke ratio of 1, a compression ratio of 7, 1000 rpm, and a valve seat angle of 0 deg.

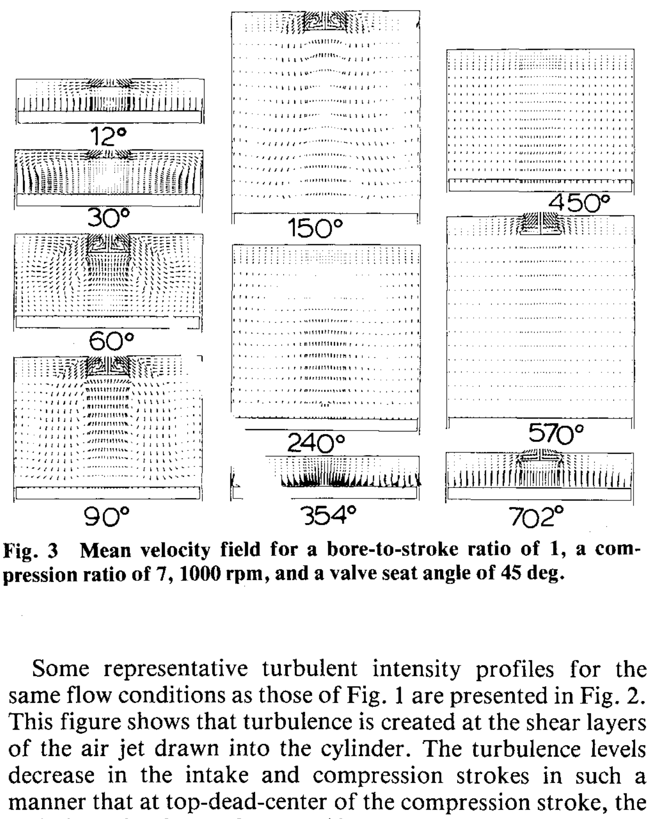

Fig. 3 Mean velocity field for a bore-to-stroke ratio of 1, a compression ratio of 7, 1000 rpm, and a valve seat angle of 45 deg.

Some representative turbulent intensity profiles for the same flow conditions as those of Fig. 1 are presented in Fig. 2. This figure shows that turbulence is created at the shear layers of the air jet drawn into the cylinder. The turbulence levels decrease in the intake and compression strokes in such a manner that at top-dead-center of the compression stroke, the turbulence levels are almost uniform. Turbulence decreases in the expansion stroke as shown in Fig. 2 at 450 deg. In the exhaust stroke, turbulence is generated at the exhaust port and near the valve because of the flow acceleration there.

The effects of the compression stroke have been studied in geometrical arrangements with the same engine stroke and indicate that large clearance volumes, i.e., smaller compression ratios, give rise to lower turbulence levels and lower velocities within the engine cylinder. This is due to the larger surface area available for friction and dissipation. The effects of the bore-to-stroke ratio were studied by varying the stroke in such a way that the compression ratio remained constant. Since the piston velocity is proportional to the stroke, a decrease in the bore-to-stroke ratio results in larger air velocities and larger turbulent intensities. In addition, for bore-to-stroke ratios smaller than 1, the vortical structures created in the intake stroke persist longer into the compression stroke than do those for bore-to-stroke ratios greater than 1.

The most important effect on the velocity and turbulent intensity fields is due to the valve seat angle. This effect has not been analyzed previously. The numerical results presented here indicate that there exists a critical valve seat angle beyond which the vortex dynamics and vortex breakdown and merging are substantially different from those existing when subcritical valve seat angles are employed. Figure 3 shows the mean velocity profiles for the case of a valve seat angle of 45 deg and indicates the formation of cylinder and valve vortices in the intake stroke. The cylinder vortex, however, is stretched in the intake stroke and located along the cylinder wall (cf. Fig. 3 at 90 deg), rather than at the corner between the cylinder head and the cylinder wall as shown in Fig. 1 at 90 deg. The valve vortex breaks down into two new vortices at 150 deg, which then merge with the cylinder vortex, giving rise to a complicated vortical flow pattern which is still present at 354 and 450 deg. In the expansion stroke, the vorticity is dissipated and the flow becomes unidirectional as it was found for a valve seat angle of 0 deg. (cf. Fig. 1). The flowfields in the expansion stroke, i.e., at 702 deg, are very similar for valve seat angles of 0 and 45 deg as indicated in Figs. 1 and 3. Figure 3 also shows the toroidal vortex created between the valve and the intake port. In the calculations reported here, the valve has been represented as an infinitesimally thin disk.

Flow patterns similar to those shown in Fig. 1 were found for a valve seat angle of 22.5 deg. The numerical results also indicate that there is a critical valve seat angle beyond which the valve vortex breaks down into two smaller vortices.[2] For valve seat angles smaller than the critical value, the valve vortex does not break down, but merges with the cylinder vortex to form a new vortex which rotates in the same direction as the original valve vortex.

The turbulent intensity profiles calculated with a valve seat angle equal to 45 deg are different from those calculated with a valve seat angle of 0 deg. The main differences are due to the turbulence generation associated with vortex breakdown and the larger levels of turbulence throughout the entire cylinder.[2] The results also indicate that turbulence is generated at the shear layers of the indrawn air jet and at the vortex edges. The turbulence levels at top-dead-center of the compression stroke are higher for a valve seat angle of 45 deg than for one of 0 deg. This is due to the turbulence generation associated with vortex breakdown. This result has not been previously reported in the literature and indicates that vortex dynamics plays an important role in establishing the turbulence levels within the engine cylinder. Not only are the turbulence levels higher for a valve seat angle of 45 deg, but the turbulence is also less homogeneous than for a valve seat angle of 0 deg. This has an important effect in determining the rates of turbulent mixing and combustion. In order to obtain better engine efficiency, it is necessary to have higher turbulence levels. These can be achieved by increasing the valve seat angle or decreasing the clearance volume, i.e., by increasing the compression ratio. Higher turbulence levels can also be achieved by using bore-to-stroke ratios less than 1.

The numerical results reported here are in qualitative agreement with the experimental data of Ekchian and Hoult.[3]

References

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