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Study of Turbulence in a Motored Four-Stroke Internal Combustion Engine

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A theoretical-experimental study of a motored, reciprocating, four-stroke, internal combustion (IC) engine is presented. The experimental data were obtained by means of an LDV system, in a transparent cylinder using smoke as seeding material. The theoretical program solves the Navier-Stokes equations in cylindrical coordinates and two additional (k/ϵ) equations for turbulence. The theoretical-experimental results show that in the intake stroke a high-speed jet strikes the valve, is radially deflected along the cylinder head, and forms a recirculation zone with almost constant axial velocity behind the valve. Production of turbulence occurs at the shear layers of the incoming jet. In the compression and power strokes, the piston drives the flow and the velocity, and turbulent kinetic energy profiles become uniform. During the exhaust stroke, a high-speed outgoing jet is formed; local generation of turbulence occurs at the shear layers of this jet and above the valve. Comparison between theory and experiment shows satisfactory agreement for the four strokes. The major reasons of disagreement are due to the lack of knowledge of the boundary conditions at the intake port which are the main input parameters to the theoretical model.

Introduction

DUE to the increasing demands for more efficient and less polluting internal combustion (IC) engines, the last decade has been characterized by a large amount of theoretical and experimental work. Unfortunately, most of the experimental work was performed in real engines of complex geometries, and its extrapolation to other engines is a subject of current research. Semenov[1] performed hot-wire measurements in motored IC engines and found that the turbulence at top dead center (TDC) of the compression stroke was almost homogeneous and mainly controlled by the intake-generated turbulence. Lancaster[2] applied hot-wire anemometry to study the effects of engine speed, volumetric efficiency, and compression ratio at TDC. He concluded that this turbulence was almost isotropic in both scale and intensity, and was controlled by the intake stroke. Witze,[3] also using hot wire, performed experiments in a motored engine and showed that the turbulence at TDC of the compression stroke appears to be dominated by the decaying intake-generated turbulence. The turbulence at TDC showed a slight increase due to the piston motion, i.e., the so-called piston-generated turbulence occurs. Using an experimental set-up similar to that used in Ref. 3, Witze[4] showed that the intake-generated turbulence rapidly decreases; this decrease is followed by a high turbulence increase as the piston approaches TDC, but Witze could not conclude if the increase was shear-generated turbulence or compression-enhanced intake turbulence. More recently, Gosman et al.[5] experimented in an ideal two-stroke engine at 2000 rpm and with a centrally-located orifice at the cylinder head. Their results show that the intake-generated turbulence decays during the exhaust stroke, but generation of turbulence also occurs in a localized region close to the exhaust port. Morse et al.[6] performed analogous LDV experiments in the same geometry and at 200 rpm, and similar conclusions were obtained. They also studied the effect of an inclined annular orifice located at the cylinder head, and showed that turbulence is generated at the shear layers of the incoming jet and where the jet strikes the cylinder wall. Ramos et al.[7] studied two-stroke cycles where the valve remains at a fixed location inside the cylinder. Four-stroke cycles were examined by Gany et al.[8] in an axisymmetric piston-cylinder arrangement, provided with a 45 deg beveled valve that moves inside the cylinder.

In this paper, we compare the experimental data of Gany et al.[8] with the theoretical model of Ramos and Sirignano[9] for a four-stroke, axisymmetric piston-cylinder configuration at 31.25 rpm. Other authors have also studied the flowfield in a four-stroke IC engine. Griffin et al.[10,11] solved the inviscid and viscous flowfields, but they did not consider the turbulence. In their studies, the valves were simulated as orifices located at the cylinder head. A similar approach was followed by Ramos and Sirignano,[12] who also studied a time-dependent annular orifice located at the cylinder head. Gosman, et al.[13] also have performed turbulent calculations in an IC engine. This paper, however, presents the first study of an axisymmetric piston-cylinder configuration with an operating valve, where both theory and experiments are compared.

In the following section, the experimental apparatus is described. This section is followed by a brief description of the theoretical model, which has been described in detail elsewhere (cf. Ref. 9). A discussion of the theoretical and experimental results for the four-strokes of the motored engine is also presented, where the potential capabilities of the theoretical model are discussed.

Experimental Apparatus

The IC engine simulator used in these experiments consists of a reciprocating engine of 7.62 cm internal diameter plexiglas cylinder. The compression ratio is 7 and the clearance 1.27 cm. The air flow is drawn into the cylinder

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Table 1 Characteristics of optical arrangement

Half angle of intersecting beams	5.73 deg
Fringe spacing	3 μm
Diameter of intersection zone	0.2 mm
Length of intersection zone	0.8 mm
Number of stationary fringes	66

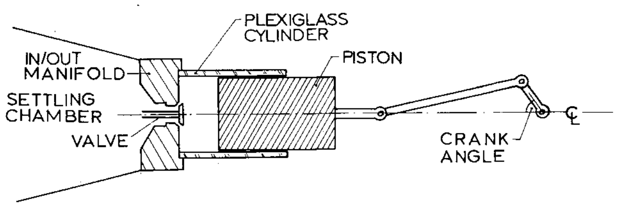

Fig. 1 Schematic diagram of the piston-cylinder assembly.

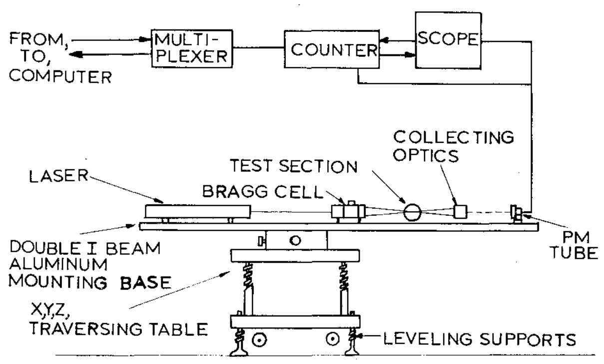

Fig. 2 Laser Doppler velocimeter and electronic instrumentation.

through a 1.57 cm inside diameter orifice, which is used for both the intake and exhaust strokes. This orifice is connected to a smoke generator, which consists of a burning cigar mounted in a glass flask. The suction created by the piston during the intake stroke accumulates the smoke in the plenum chamber. To avoid particles of diameter greater than 5 μ a filter was set before the plenum chamber. Before actually obtaining the experimental data, the engine was run for about 30 min to get enough seeding material in the plenum chamber. Figure 1 shows a diagram of the experimental apparatus. The valve is beveled at 45 deg with respect to the cylinder axis and is 2.16 cm in diameter. The valve rod diameter is 0.3925 cm. The engine rotational speed is 31.25 rpm which, as demonstrated by both the theory and experiments, is large enough to produce turbulence within the cylinder.

The measuring system employs a 15 mW He-Ne Spectra Physics Laser (Model 124A) with a wavelength of 633 nm. The beam produced by the laser is split into two beams by means of transmitting optics (Thermo Systems 900 Series V Optics) and focused to the point of interest inside the cylinder. The principal characteristics of the optical arrangement are given in Table 1. To resolve the directional ambiguity that results from the reversal flow zones, a Bragg cell (2 MHz frequency shifter) with electric down-mixer is incorporated in the transmitting optics. The LDV signals, that result from the scattered light of the test point, are amplified by the photomultipler (PM) tube, processed, without filtering, by a Sci-Metric Counter (Model 800A) and shown on a Tektronix oscilloscope (Model 7603). Validated signals are transmitted to the HP21MX minicomputer via a multiplexer. The laser, transmitting and collecting optics, and photomultiplier are mounted on an aluminum, double T beam which is mounted on a table with three degrees of motion. Figure 2 shows a diagram of the main components of the LDV optics and signal transmission systems. The cylinder axis is normal to the plane of Fig. 2. As just mentioned, incorporated into the line driver there is an eight-input, 16-bit wide, eight channel multiplexer. The first channel is wired directly to the signal processor (counter), while the second channel is connected to the output of a clock driven by a switch-selectable digital generator (range 10-2000 μs). The other channels were not employed in this study. An external zero reset of the clock is triggered by a signal from a TDC indicator that allows for identification of individual cycles and relates the detected signals to the proper crank angles.

Measurements and Data Processing

Measurements of the axial velocity component have been separately taken at 31.25 rpm and in a four-stroke, axisymmetric, motored engine. The Reynolds number, based on the bore, mean piston velocity, and atmospheric density and viscosity, is 610, while the Reynolds number, based on the orifice diameter and mean axial velocity at the port, is 2980. (The mean axial velocity at the port is given by the product of the mean piston velocity and the area ratio between the cylinder and the port orifice.) Twenty to twenty-five radial data points were obtained for each axial location. Measurements were made on both sides of the cylinder axis and proved that the flow is practically axisymmetric.[8,14] Each individual test consisted of 10,000 data points (at each axial location) taken over a number of engine cycles. Recent experiments[15] with 5000, 10,000 and 20,000 data points show that the results with 5000 data points are statistically inadequate, while those with 10,000 and 20,000 points differ by less than 3 and 7%, in the mean axial velocity and rms, respectively; however, the running time is doubled when 20,000 data points are used. From these experiments, it was concluded that 10,000 data points is always large enough to give an adequate statistical sampling, while keeping a feasible running time.

The most important achievements of these experiments is the study of a four-stroke axisymmetric engine where the valve moves inside the cylinder. Problems arose in this configuration when the valve was operated, e.g., poor sealing, pressure resistance, mechanical stresses, and water condensation during the expansion stroke. With some effort these problems were solved. By regulating the amount of smoke in the intake flow, we were able to receive good Doppler signals during the whole four strokes. The automatic data processing was used to show the interesting features of the fluid dynamics, the operation characteristics during each test, and the quality and other characteristics of the data collected. Thus, the following can be obtained for the axial velocity component:

1) Scattering plots were obtained including all data points taken for an individual test (raw data).

2) Composite curves and listings giving the mean axial velocity at each crank angle were obtained. We have used a phase-time average[16] wherein the instantaneous value at a specific crank-angle position is averaged over many cycles of the engine. In this phase-time average, the time average is done over a finite crank-angle window of 1 deg interval, in such a way that the mean velocity at the center of the averaging angle is defined as the mean of the instantaneous velocities recorded in several cycles. The instantaneous fluctuating velocity is defined as the difference between the instantaneous velocity and the phase-time averaged velocity. The square root of the phase-time average of the square of the instantaneous fluctuating velocity is called U' and presented in the results. We have also used angle intervals of 1 and 2 deg, and showed that the mean velocities are affected only slightly by the angle interval, but the rms values are overestimated for 2 deg interval when the variation of the velocity with the crank-angle is larger. Similar conclusions were obtained by Morse et al.[6] who showed that the phase-time average introduces an error in the turbulent quantities. This error is negligible except during the highly transient periods of the intake stroke. Our rms values have not been corrected in the intake stroke.

3) Rms of the velocity fluctuations (plots and listings) as a measure of the turbulent intensity were obtained. The experimental evaluation of the rms was mentioned before.

4) Plots of data taken at one or a number of individual cycles to indicate fluctuations within a cycle and possible cycle-to-cycle variations were also obtained.

An important aspect of the engine operation that has been investigated is the comparison among different cycles. Such a comparison has revealed that, in general, the scatterings of the data from different cycles do not exhibit any specific trends, and that the data points of different cycles overlap each other. Thus, it was concluded that at 31.25 rpm, cycle-to-cycle variations are not proven to be important in our motored engine design. The same conclusions have been obtained at 260 rpm.[15] Note that phase-time averaging for the purpose of describing the turbulence is performed over many cycles. There is not sufficient data in one cycle to measure turbulence fluctuations with statistical accuracy. Our evidence, however, indicates that fluctuations from cycle-to-cycle are no greater than fluctuations within a cycle.

Sources of experimental error include the uncertainty in the position of the measurement, flow asymmetry, variation in the speed of the engine, velocity gradient, and crank angle broadening. As we mentioned before numerous tests were conducted to ascertain that the flow was axisymmetric, while the crank angle broadening was reduced by averaging in intervals of 1 crank angle degree. The variation in the speed of the engine results from the cyclic torque requirement over the cycle. These cyclic torque fluctuations were smoothed by the flywheel of the engine, so that the variations in the rotation velocity were not more than $\pm 1\%$. The positional error has been estimated as 0.1 mm and thus is negligible relative to the separation between measurements located in the radial and axial directions (2 and 25.4 mm, respectively).

Theoretical Model

To study the axisymmetric flowfield in a four-stroke, motored engine a theoretical model was developed for a sudden valve[17] and for an annular valve with and without swirl.[12] Later, these authors extended this model to study the presence of a moving valve inside the cylinder.[9] Since the model has already been reported in Ref. 9 with additional details given by Ramos,[18] here we only consider a brief summary of this model. Basically the Navier-Stokes equations in cylindrical coordinates are solved together with the energy equation and two additional equations, k and ϵ, for the turbulence; where k and ϵ represent the turbulent kinetic energy and its dissipation rate. The Reynolds stresses are modeled as gradients of the mean velocity. The valve motion inside the cylinder is studied by a novel technique,[9] which consists of a fixed domain wherein the valve lies and a moving domain that extends up to the piston face. These two domains were required to transform the moving boundary-value problem associated with the piston motion into a fixed boundary-value problem. The governing equations are then written in terms of the appropriate transformed coordinates, and integrated along cell-volume elements. This integration procedure produces a set of fully implicit finite-difference equations that are solved line-by-line using an iterative process. The pressure is calculated, after the solution of the two momentum equations, in such a way that the continuity equation is brought into balance. Further details have already been given by Ramos and Sirignano.[17,19] In the theoretical model the 45 deg beveled valve used in the experiments is simulated as an infinitely thin disk of 1.57 cm in diameter without rod. Here a compromise has been made. The valve shape employed in these experiments was required to ensure sealing during the compression and expansion strokes. Such a shape, however, is not represented conveniently as a boundary in a numerical calculation, so that some dissimilarity exists. Of course, the major features of the flow outside of the immediate vicinity of the valve should be somewhat similar in both theory and experiments. Thus, three major differences between the theoretical and experimental valve should be noted: 1) the valve seat angle, 2) the valve diameter, and 3) the valve rod.

Initial and Boundary Conditions

Initial and boundary conditions must be prescribed to fully specify the input parameters in the theoretical model. Initially, all the flow variables were set to zero for the hydrodynamic field. The temperature and pressure were initially specified at 300 K and 1 atm, respectively, while the wall temperature was set at 350 K. Since at the end of the first numerical cycle, the flowfield is unlikely to be zero everywhere, the process has to be repeated, taking these final values as initial conditions for the next numerical cycle for as many cycles as required to get periodic solutions. The results

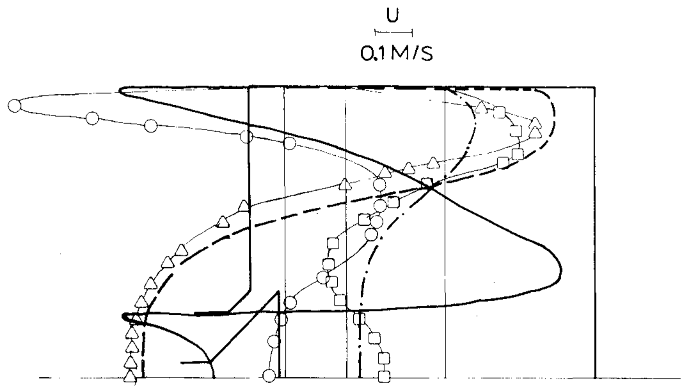

Fig. 3 Profiles of mean axial velocity at 90 deg for a four-stroke, motored engine: —, theory at 0.635 cm; – – – –, theory at 2.54 cm; – · – · –, theory at 5.08 cm; – · · – · · – · · –; theory at 7.62 cm; ○, experiments at 0.635 cm; △, experiments at 2.54 cm; □, experiments at 5.08 cm; ●, experiments at 7.62 cm.

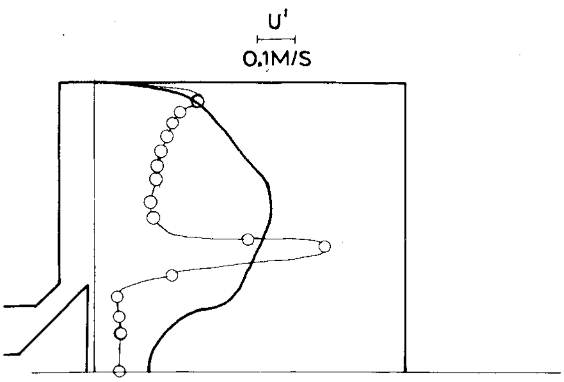

Fig. 4 Profiles of the axial rms at 90 deg for a four-stroke, motored engine: —, theory at 0.635 cm; ○, experiments at 0.635 cm.

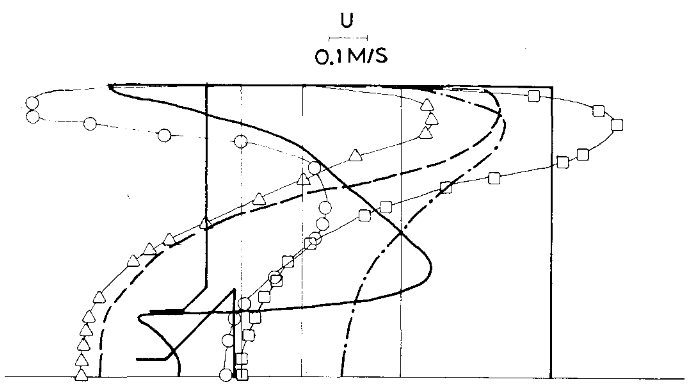

Fig. 5 Profiles of mean axial velocity at 120 deg for a four-stroke, motored engine (for legend see Fig. 3).

reported here were calculated with a 31×31 mesh, a 6 deg crank angle step and required four numerical cycles.

The boundary conditions at the intake port used air at 300 K, a 3% turbulent level, a zero radial velocity, and an axial velocity given by the incompressible continuity equation during the intake stroke. The dissipation rate of turbulent kinetic energy was also specified.[18] At any solid wall, the normal velocity to the wall was set to zero, while the parallel velocity to the wall and the temperature were calculated by means of the law of the wall; the normal gradient of the turbulence energy was set to zero and the dissipation rate of turbulent kinetic energy ϵ was assumed equal to the production of turbulence kinetic energy. In the exhaust stroke, the normal gradient of the radial velocity, temperature, turbulent kinetic energy, and its dissipation were set to zero, while the axial velocity is again given by continuity.

A parametric study was conducted to assess the influence of the boundary conditions at the port in the flowfield within the cylinder. From this parametric study we selected the aforementioned boundary conditions during the intake and exhaust strokes because they provided the best overall agreement with the experiments. Thus, when comparing the theoretical and experimental studies, one should keep in mind that although the valve geometry is different, the boundary conditions were specified to give good agreement in the region between the valve and the piston face. It is also worth noting that if we knew the experimental values of the flow properties at the valve seat, these boundary conditions could be used to remove the details associated with the valve geometry; however, at the present time these experimental values are not available. They could be obtained using backscatter in future experiments.

Results and Discussion

In this section, we present mean axial velocity profiles and axial rms of the axial velocity fluctuations at different, fixed, axial locations inside the cylinder. In the following figures, we have shown the real valve used in experiments instead of the infinitesimally thin disk used in the theoretical model. The rms profiles were calculated at only one axial location, i.e., at 0.635 cm.

Figure 3 shows the mean axial velocity profiles at three axial locations inside the cylinder and at 90 deg in the intake stroke. An air jet is drawn into the cylinder, follows a 45 deg inclined line with respect to the cylinder axis, strikes the cylinder wall, and forms a recirculation zone with almost uniform axial velocity behind the valve. The flow trends are

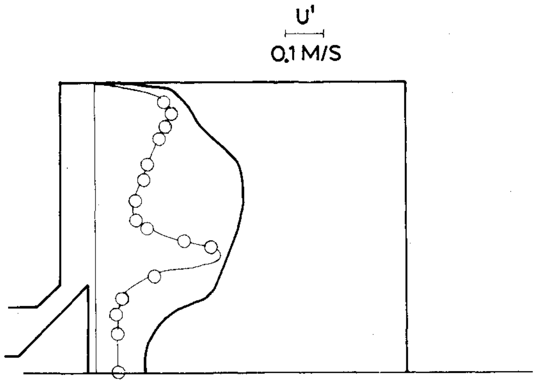

Fig. 6 Profiles of the axial rms at 120 deg for a four-stroke, motored engine (for legend see Fig. 4).

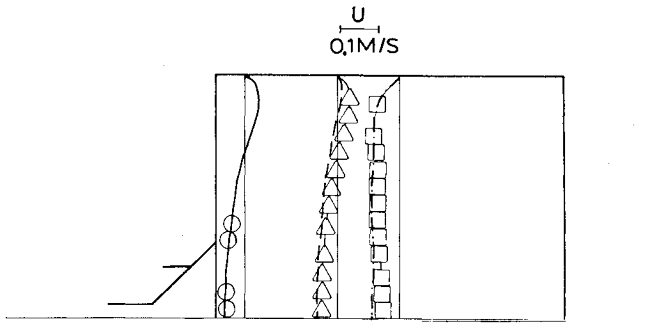

Fig. 9 Profiles of mean axial velocity at 240 deg for a four-stroke, motored engine (for legend see Fig. 3).

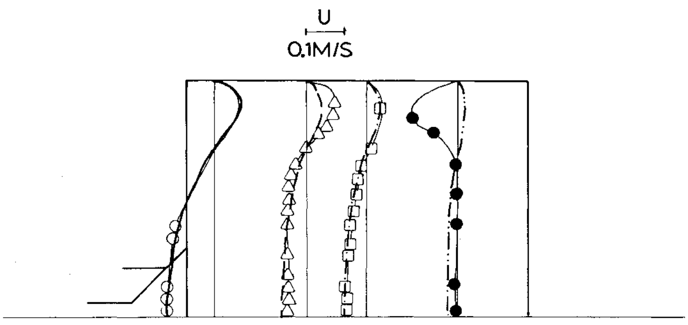

Fig. 7 Profiles of mean axial velocity at 210 deg for a four-stroke, motored engine (for legend see Fig. 3).

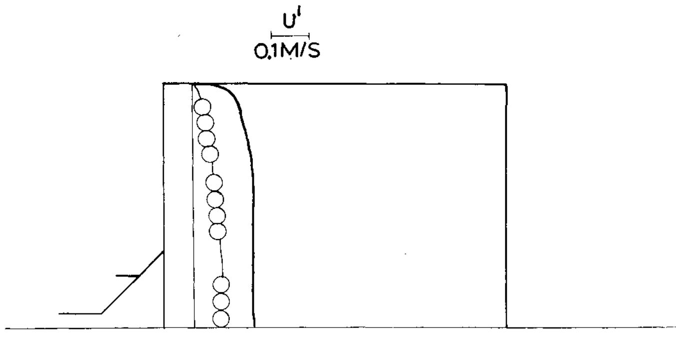

Fig. 10 Profiles of the axial rms at 240 deg for a four-stroke, motored engine (for legend see Fig. 4).

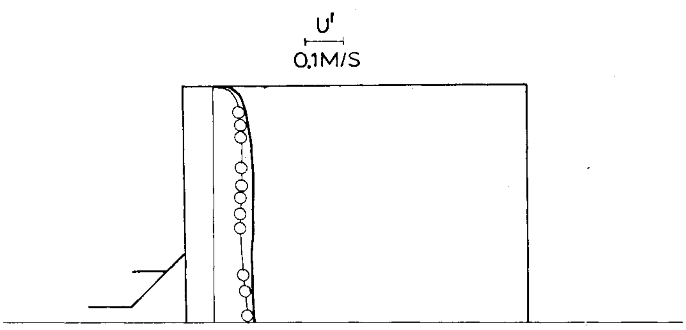

Fig. 8 Profiles of the axial rms at 210 deg for a four-stroke, motored engine (for legend see Fig. 4).

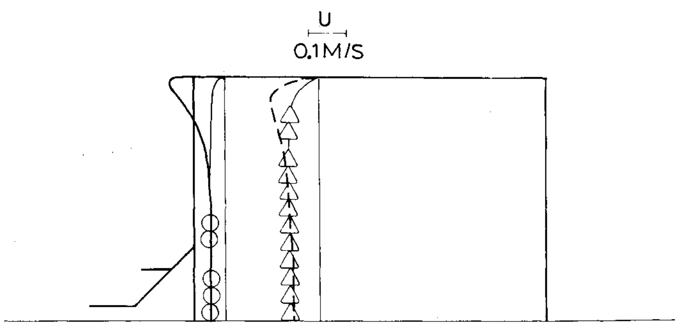

Fig. 11 Profiles of mean axial velocity at 300 deg for a four-stroke, motored engine (for legend see Fig. 3).

predicted except at the axial location at 0.635 cm because of the valve geometry. Once more we emphasize that the boundary conditions at the port were chosen so that the best overall agreement resulted. Figure 4 shows the rms of the fluctuating axial velocity. Agreement only at the roughest qualitative level is indicated in this figure, except close to the cylinder wall where the incoming jet strikes. This is again associated with the geometry of the valve. In particular, the real jet is drawn into the cylinder along a 45 deg inclined line, i.e., the valve angle. We note that Fig. 3 presents a very steep velocity gradient just above the valve. This velocity gradient produces high levels of turbulence at the same location (Fig. 4). Additional turbulence is generated where the indrawn jet strikes the cylinder wall. The comparison between the experimental and theoretical rms, calculated with the assumption of isotropic turbulence, is not good. This is again associated with the different valve geometry. In particular, while the experimental valve produces a 45 deg inclined jet, the theoretical valve predicts a jet that is drawn along the cylinder axis, strikes the valve and moves along the cylinder head. This jet strikes the cylinder wall where more turbulence is generated.

Figure 5 shows the mean axial velocity profiles at three different axial locations and at 120 deg during the intake. The main trends of the flow are again predicted except at the axial location at 0.635 cm where the details of the valve geometry are important. Behind the valve there is a recirculation zone with almost a constant mean axial velocity. As shown in Figs. 3 and 5, there is very strong vortical motion behind the valve. This vortical motion has been detected in our theoretical models.[18] One could ask if these vortical structures can be unsteady with a characteristic frequency and how much they contribute to the turbulence observed by point measurements. This question of flow instability is presently under consideration. In these experiments the Reynolds number based on the bore and mean piston velocity is 610, and one could question if this Reynolds number is large enough to have turbulent flow or if the flow is in a transitional stage. We also point out that the k/ϵ model of turbulence is based upon a high Reynolds number assumption, and its applicability to this low Reynolds number experiment is not clear in spite of its fair agreement with the experimental results.

Figure 6 shows that the agreement between theory and experiments is fair in a qualitative sense. At 210 deg in the compression stroke the theory compares fairly well with the experiments for the axial velocity profiles (Fig. 7) except at the axial location at 7.62 cm and close to the cylinder wall where the experiments are probably in error. This error is apparently associated with the accumulation of cigar smoke at the cylinder wall.

The rms profiles are slightly overpredicted at 210 deg (Fig. 8). Apparently the intake generated turbulence decays faster than the theoretical model predictions. Figures 9 and 10 show the mean velocity and rms profiles at 240 deg in the compression stroke. The flow trends are predicted but the rms value is overpredicted by a factor of two. The same characteristic trends can be observed in the deceleration part of the compression stroke (Figs. 11 and 12), while the expansion stroke shows an almost unidirectional fluid motion (Fig. 13). The theory in this case underpredicts the rms values (Fig. 14). This is due to the fact that in our theoretical model the production of turbulence energy contains a term that subtracts kinetic energy when the flow is expanding. In the exhaust stroke (Figs. 15 and 16), the flow is pushed out of the cylinder by the piston. This flow moves around the valve and

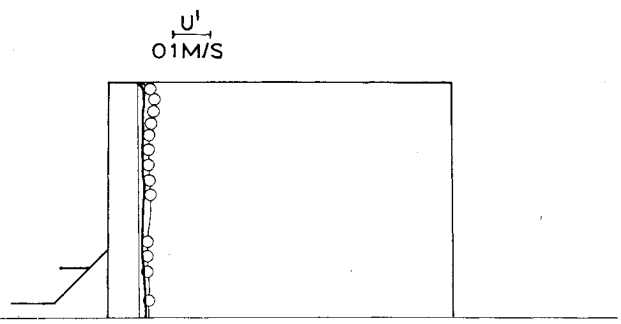

Fig. 14 Profiles of the axial rms at 480 deg for a four-stroke, motored engine (for legend see Fig. 4).

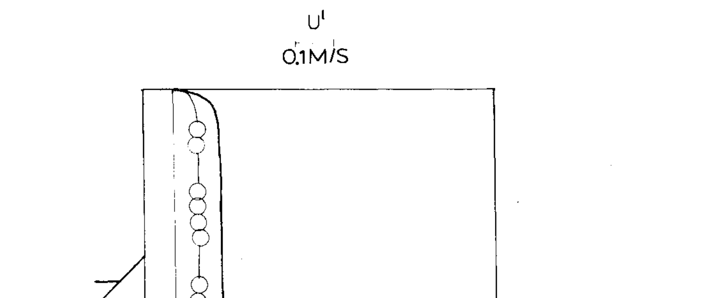

Fig. 12 Profiles of the axial rms at 300 deg for a four-stroke, motored engine (for legend see Fig. 4).

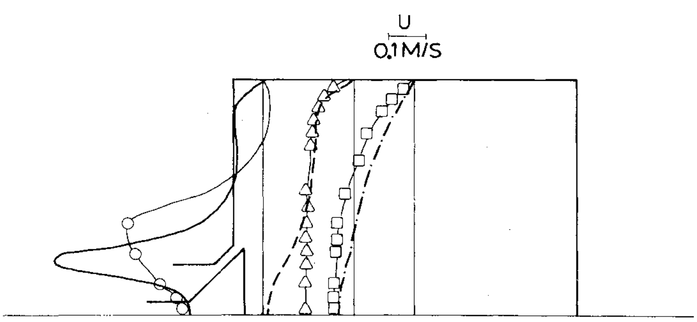

Fig. 15 Profiles of mean axial velocity at 600 deg for a four-stroke, motored engine (for legend see Fig. 3).

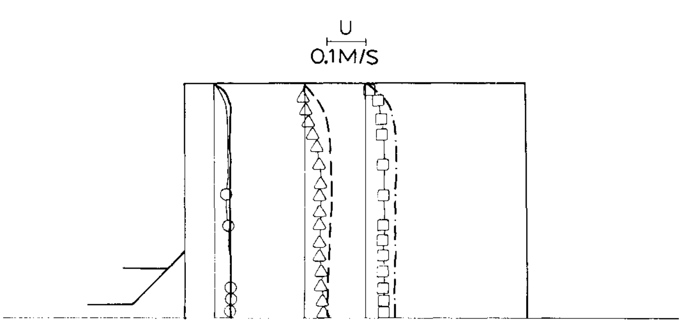

Fig. 13 Profiles of mean axial velocity at 480 deg for a four-stroke, motored engine (for legend see Fig. 3).

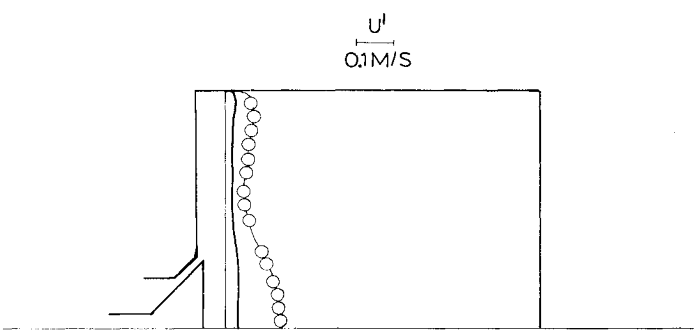

Fig. 16 Profiles of the axial rms at 600 deg for a four-stroke, motored engine (for legend see Fig. 4).

thus produces a velocity peak just above the valve. This velocity peak is smaller than theoretically predicted. Turbulence generation occurs close to the valve, because the flow is decelerated there (Fig. 16). The theoretical model underpredicts this increase of turbulence.

Conclusions and Areas for Future Research

The flowfield in a four-stroke, motored, reciprocating IC engine has been theoretically and experimentally studied at 31.25 rpm. The results of the study show that in the intake stroke a high-speed jet is drawn into the cylinder, strikes the cylinder wall, and forms a recirculation zone behind the valve, which has an almost uniform axial velocity. Generation of turbulence occurs at the shear layers of the incoming jet and where the jet strikes the cylinder wall. In the compression and power strokes, the flow is driven by the piston, the axial velocity profiles are almost uniform, while the turbulence is diffused and dissipated. In the exhaust stroke, a high-speed jet is created; this jet turns around the valve and locally produces turbulence.

The comparison between theory and experiments shows a fair agreement although the recirculation zone behind the valve is slightly overpredicted in the deceleration part of the intake stroke. The mean flow is overpredicted in the expansion and exhaust strokes. The decay of the intake-generated turbulence is slower in the compression stroke and faster in the expansion stroke because of the flow expansion.[9] Several areas of this study deserve further attention, in particular the effect of the vortical structures and their contribution to the measured turbulence. These vortical structures have also been shown by other investigators.[20-22] The phase-time average used to calculate the mean velocity profiles gives an error in the determination of the rms values, whose effects in the resulting rms are to be determined. The rotational speed (31.25 rpm) used in these experiments is not typical of internal combustion, neither is a single-valved engine; however, theoretical calculations[9] show the flow characteristics are similar. In particular, the turbulence at TDC of the compression stroke seems to be a linear function of the rpm. Similar results were also obtained by Semenov.[1] We are presently studying the flowfield in the same configuration at 260 rpm (almost the limit of our present configuration), and preliminary results[15] show that the flow characteristics are similar, and that the intake-generated turbulence is proportional to the square of the rpm. The theoretical model has used to predict the flowfield at 1000 rpm and has shown the same flow characteristics as those indicated here. However, the experimental apparatus is unable to stand operation at 1000 rpm, so a new design would be required, that would permit us to analyze and compare the flowfields at more realistic higher rpm. Also, the theoretical model has to be modified to handle more realistic valve geometries, that are required to perfectly seal the port during the compression and expansion strokes.

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