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ABSTRACT

The purpose of this paper is to characterise barrel swirl behaviour in a production four-valve engine with pentroof chamber. Steady flow analysis showed that the insertion of tubes into the cylinder head's induction tracts increased the tumbling ratio of the in-cylinder flow field at intake valve closure. A comparison of LDA measurements, conducted along the spark plug axis, for tubes and no tubes inserted yielded the following conclusions.

The results indicated that the barrel swirl generated was not efficiently breaking down into turbulence but forming two counter-rotating vortices in the horizontal cylinder plane. The turbulence levels and cycle-to-cycle flow variations towards the end of the compression stroke increased with tumbling ratio. The former suggested faster combustion rates if applied to a lean burn engine, however, the latter suggested greater cyclic combustion instability and may limit lean burn capability. Both the rotational sense of the vortices and their cyclic stability appeared to be sensitive to tumbling ratio.

INTRODUCTION

Four-valve engines and pent roof chambers are currently being developed by the automotive industry, where both the large inlet-valve curtain area and centrally located spark plug offer improvements in engine torque and therefore driveability [1]*. Future power plants, however, must also meet new legislative combustion requirements of low emission levels (NO_x, CO and HC) combined with reduced fuel consumption [2].

Operating four-valve engines at part load with diluted mixtures through lean operation or exhaust gas recirculation is a solution for reducing both emissions and fuel consumption [3]. However the slow burning rate and poor mixture preparation that accompanies lean burn leads to cyclic variability of the combustion process which limits the lean burning capability of engines [4].

The enhancement of in-cylinder turbulence has significantly reduced the problems associated with lean burn [3,4]. However, lean burn technologies cannot meet future emission legislations without the aid of other emissions technologies. Excessive levels of turbulence can also lead to other undesirable effects, namely ignition difficulties and flame quenching [5]. Hence there is believed to be an optimum level and scale of turbulence inherent to a particular cylinder head design for reducing emissions and fuel consumption.

Methods of controlling the levels of turbulence in four-valve engines have involved variable valve timing or variable geometry manifolding [6]. However as these methods are bulky and expensive and consequently add to the cost of the vehicle, simpler alternatives are therefore desirable.

Recent studies have shown that barrel swirl can generate higher turbulence levels in four-valve engines without recourse to complex design [7-22]. These studies have been experimental, using laser Doppler anemometry (LDA) and laser sheet particle tracing techniques, and theoretical using CFD and phenomenological models. Barrel swirl (tumble) is an organised rotation of air motion around an axis perpendicular to the cylinder axis. During the compression stroke, angular momentum is conserved and then finally released as turbulent kinetic energy due to the high shear stresses that develop. The control of barrel swirl and its breakdown is believed to be a function of intake porting arrangement and chamber shape [19].

The measurement of turbulence in reciprocating engines is also a matter of definition. In-cylinder flow fields are unsteady and exhibit variation from cycle-to-cycle and thus preclude analysis by conventional statistical methods. It has therefore been usual to replace time-averaging by ensemble-averaging which phase averages the flow field at a particular crank angle position over many engine cycles. Ensemble averaging, however, must always be complemented by a cycle resolved analysis in order to differentiate between cycle-to-cycle flow variations and turbulence generation [23-25].

The present study focuses on barrel swirl behaviour in a production four-valve engine with pentroof combustion chamber. Steady flow rig analysis highlighted the effects of intake port modifications on the tumbling ratio at intake-valve closure and on breathing capacity. LDA measurements in a motored engine cylinder provided information about the degree of mean velocity and turbulence enhancement in the pentroof chamber. The implications of the flow fields generated for combustion are finally discussed.

* Numbers in parentheses designate references at end of paper

LITERATURE REVIEW

Gosman *et al* [7] in 1985 and Benjamin [8] in 1988, both suggested that the ideal situation for enhancing the combustion rate would be for the tumbling vortex to be completely broken up into relatively homogeneous micro turbulence.

Pischinger *et al* in 1990 [26], reported that in addition to turbulence enhancement some mean motion is also required (3 -5 m/s) to convect the flame kernel away from the quenching affects of the electrodes. In the same context, Mantel *et al* in 1992 [27] also showed that the ground electrode may have to be considered as a means of optimising the flame initiation phase of combustion.

The tumbling vortex ratio (TVR) measures the level of barrel swirl, defined as the rotational speed of the main vortex divided by the engine's rotational speed. Chapman *et al* in 1991 [28] and Arcoumanis *et al* in 1993 [29] both reported steady flow analysis procedures to estimate a value of TVR at intake valve closure. Arcoumanis *et al* in 1990 [11] and 1994 [21] also reported procedures to estimate the variation of TVR with crank angle from the integration of local velocity measurements in motored engines.

Gosman *et al* in 1985 [7], reported a combined experimental and computational study of the in-cylinder flow field in a disc chamber with a centrally located shrouded valve. Tests were conducted on a model engine, motoring at 200 rev/min with a compression ratio of 6.7:1. He showed that the shrouded valve generated a long-lived tumbling vortex which is sustained and amplified by the compression process which in turn caused amplification of the ensembled r.m.s turbulence levels towards top-dead-centre (TDC) compression (0.5 to 1.2 Vp). The r.m.s turbulence levels were normalised by the mean piston speed (Vp) because of their observed linearity with engine speed. His flow field predictions at TDC compression also showed evidence of swirling structures and other tumbling structures in the horizontal and vertical cylinder planes respectively.

Le Coz *et al* in 1990 [10] reported in-cylinder flow field predictions (CFD) and LDA measurements in a four-valve pentroof chamber. Tests were conducted on a production engine motoring at 1200 rev/min with a compression ratio of 9.5:1. No estimate of the TVR was reported. Both sets of results conducted at bottom-dead-centre (BDC) induction showed two inclined tumbling structures. At TDC compression, two counter rotating vortices in the horizontal cylinder plane were observed sharing a common stream line which runs from intake side to the exhaust side. He also applied a cycle resolved analysis using fast Fourier transforms and reported that barrel swirl generated turbulence was amplified at all frequencies.

Hadded *et al* in 1991 [12] reported LDA measurements along the spark plug axis in a four-valve pent roof chamber with low (TVR≈0.58) and high (TVR≈1.14) tumbling vortex ratios. Tests were conducted on a production engine, motoring at 1000 rev/min with a compression ratio of 10:1. For both cylinder head configurations, ensembled mean velocities at TDC compression, measured along the extent of the spark plug axis, were directed from intake to exhaust valves. Both mean velocity magnitudes and direction remained approximately constant throughout the expansion stroke. Ensembled r.m.s turbulence levels increased with TVR from 0.75 Vp to 1.0 Vp at 20 deg BTDC compression. He also conducted a cycle resolved analysis on these results and reached similar conclusions.

Saneyoshi *et al* in 1991 [13] applied stereoscopic photography to observe barrel swirl behaviour three dimensionally in a four-valve engine with pentroof combustion chamber. Tests were conducted on a production engine, firing at 1500 rev/min with a compression ratio of 8.7:1. No estimate of the TVR was reported, however he showed visually that a strong tumbling motion was present. At top-dead centre of the compression stroke he observed two contracting flows directed against each other and parallel to the ridge of the chamber together with a flow directed from the intake to the exhaust valves.

Hu *et al* in 1992 [15,16] reported LDA measurements along the spark plug axis in a four-valve engine with pent roof chamber with low (TVR≈0.4) and high (TVR≈1.7) tumbling ratios. Tests were conducted on a production engine, motoring at 1000 rev/min with a compression ratio of 10.5:1. Contrary to the results presented by Hadded *et al* [12] mean velocities, conducted along the extent of the spark plug axis, were found to decrease in value and change in direction at about 45 deg BTDC compression. The ensembled r.m.s turbulence levels were also found to increase to 1.6 Vp at the same crank angle instance, 15 deg earlier than those reported by Hadded. No cycle resolved analysis was conducted and therefore the magnitude of the turbulence increase is suspect. At TDC compression, a tumbling vortex was still apparent showing a gradual collapse up to 30 deg ATDC compression. He concluded that the centre of the tumbling vortex was moving through the measuring location and persisting after TDC compression due to the favourable shape of the pent roof geometry.

Kiyota el al in 1993 [17] applied a multi-colour laser sheet flow visualisation technique to a four-valve engine with pent roof chamber. Tests were conducted on a production engine, motoring at 1000 rev/min. The compression ratio was not reported. The tumbling ratio (TVR≈2.0) was calculated according to the procedure reported by Omori *et al* in 1991 [30]. Kiyota demonstrated the concept of lean combustion by Barrel-Stratification using the inherent characteristics of twin intake porting arrangements. Charge stratification was achieved by one-port fuel injection. He observed that the mixing rate of the fuel and air between the two intake jets was slow within the time frame of the engine cycle. Stratification was conserved until combustion where a richer region was prepared around the spark plug. He argued that the slow mixing rates were due the relatively small velocity component in the direction parallel to rotational axis of the tumbling motion.

Within the barrel stratified engine, Kiyota *et al* [17] also observed a triplet vortex structure at 15 deg BTDC compression, comprising of two counter rotating vortices in the horizontal cylinder plane and one vertical vortex. The common streamline of the horizontal vortices was from exhaust to intake valves and the vertical vortex rotated from intake to exhaust valves. He also stated that "At the beginning of the generation of the triplet vortex, kinetic energy was conserved by a shrinked vertical vortex and then distributed to the horizontal vortices through the common stream line". At TDC compression the triplet vortex structure decayed to many small eddies. Hu *et al's* 1992 [15,16] results ire therefore open to interpretation and he may in fact be measuring a vertical cross-section of the triplet vortex structure.

Kuwahara *et al* in 1994 [19], generalised Kiyota's triplet vortex structure as two 'wing like' vortices. He also investigated the effects of two vertical partitions located upstream of the intake valve stems on barrel swirl behaviour. These partitions were found to reduce flow separation down stream of the valve stem and intensify barrel swirl. In addition, the effect of a tumble control piston (pentroof shaped), in place of a flat piston, on barrel swirl behaviour was also investigated. This piston was found to suppress the 'wing like' structures and thus the bulk flow field around the spark plug at the time of ignition. He concluded that both intake port partitions and tumble control piston assisted optimisation of the flow field structure after distortion of the tumble to enable stable lean combustion.

Trigui *et al* in 1994 [20] reported CFD predictions in a four-valve pentroof chamber. The engine operating conditions were not reported. His predictions showed two counter rotating axial vortices towards the end of the compression stroke sharing a common stream line which runs from intake side to the exhaust side. The rotational sense of these vortices is the same as those reported by Le Coz *et al* in 1990 [10], but opposite in sense to those reported by Kiyota *et al* in 1993 [17] and Kuwuhara *et al* in 1994 [19]. He also stated that "the dominant tumbling vortex observed at the end of the intake stroke has been weakened and that some of its energy has been transferred to counter rotating secondary vortices". Hadded *et al's* 1991 [12] results are also now open to interpretation, and he may in fact be measuring a vertical cross-section of this structure.

Arcoumanis *et al* in 1994 [21] reported LDA measurements along the spark plug axis in a four-valve pentroof chamber with low (TVR≈0.65) and high (TVR≈1.7) tumbling ratios. His results and conclusions are similar to those reported by Hu *et al* in 1992 [15,16]. In addition, he stated that "a strong tumbling motion is expected from ports that are oriented as close as possible to the mid-plane between the valves, thus reducing any chances of any motions present in the horizontal plane to survive later in the cycle". This statement and Kuwuhara's [19] study, both indicate that four-valve intake configurations have a strong influence on barrel swirl behaviour.

Reeves *et al* in 1994 [22] conducted PIV measurements in the vertical plane in a pentroof chamber of a motored four-valve engine. Tests were conducted at 1000 rev/min with a compression ratio of 9:1. The engine type is similar to that tested by Hu *et al* in 1992 [15,16] with a TVR at intake valve closure of about 1.7. He observed two counter rotating vertical vortices present at the point of ignition in a plane that intersects the centre of the intake and exhaust valves. He also reported that the resolved flow structures resemble those presented by Kiyota *et al* in 1993 [17]. This comparison will therefore also apply to Kuwahara *et al's* study in 1994 [19]. It therefore appears that the 'wing like' vortex structure near to TDC compression is a common feature of four-valve engines operating at high tumbling ratios. However, this observation fails to explain the contradictory results presented by Hadded *et al* [12] at a lower TVR of about 1.14, Trigui *et al* [20] and Le Coz *et al* [10].

Concluding, there is a general agreement that as the piston approaches TDC compression, the barrel swirl vortex breaks up into smaller vortices. There are contradictions, however, as to the size, rotational sense and cyclic stability of the vortices. There also appears to be no precise explanation to why different types vortex structure actually exist during the late compression stage. The common factors appear to be twin intake port arrangements, tumbling ratio, pentroof chamber and flat piston.

Further investigations are therefore required into the effects of intake porting arrangement and / or combustion chamber geometry on barrel swirl behaviour in four-valve engines. This research may allow better control of the tumble generated turbulence levels and hence the combustion process for reducing emissions and fuel consumption.

EXPERIMENTAL SYSTEM

OPTICAL RESEARCH ENGINE - In-cylinder flow field measurements were made on a Rover M16i four-valve engine with pentroof combustion chamber. Figure 1 shows a schematic diagram of the chamber geometry and table 1, overleaf, shows the research engine specifications.

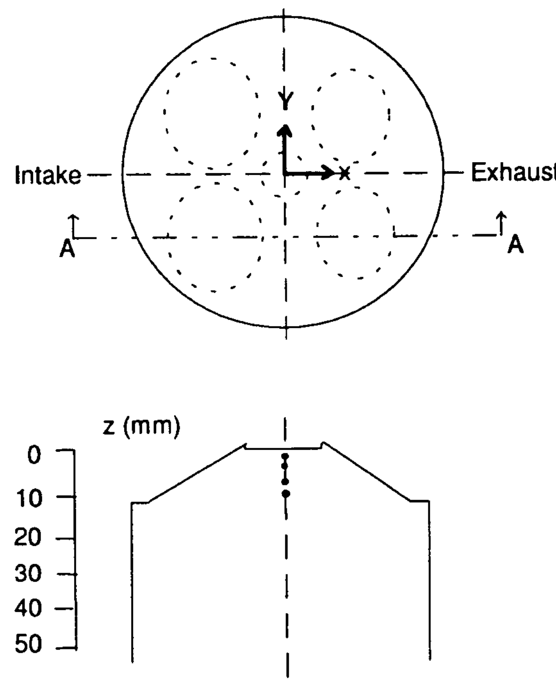

Figure 1 Four-valve pent roof chamber showing velocity components and sign conventions.

Optical access for LDA measurements in the pentroof chamber was provided by incorporating a small window in the spark plug hole. The compression ratio of the engine could be varied by inserting spacers of varying thickness between a piston extension and a flat topped screw on piston cap. A large plenum chamber was situated upstream of the throttle body, to permit the introduction of the necessary seeding for LDA measurements.

Table. 1 Research engine specifications

Bore		84.45 mm
Stroke		89 mm
Con rod length		148 mm
Compression ratio		6.2:1
Intake Valves: diameter		32 mm
	seat angle	45°
	open at	12 deg BTDC
	close at	52 deg ABDC
	Max. lift	8.712 mm

INTAKE PORT MODIFICATIONS - The tumbling vortex ratio at intake valve closure of the standard cylinder head configuration was known to be about 0.4, defined using the steady flow analysis reported by Chapman *et al* in 1991 [28].

To increase the tumbling ratio, the cylinder head was modified by inserting tubes with ramps into the two intake port tracts to direct the flow over the valves in varying degrees, as shown in fig. 2. The longest tube extension (L) that could be inserted into both ports was 30 mm, which just cleared the valve stem of the intake valve.

STEADY FLOW RIG - Steady flow measurements of the cylinder bead's tumbling vortex ratio (TVR) were undertaken on the flow rig shown schematically in fig.3. Arcoumanis *et al* in 1993 and 1994 [29,21] confirmed the validity of TVR measurements using this type of steady flow device by comparing with equivalent results obtained from a motored engine. Further details regarding the design of flow rig can be found in the literature although a short description is presented below; Seeley *et al* in 1993 [31] and Baker *et al* in 1994 [32].

A 84mm diameter cylinder fitted with a 90° circular adapter tube (referred to as the tumble to swirl conversion tube) was attached to the cylinder head. This arrangement converts the in-cylinder tumbling motion into swirl whilst allowing free passage of the air through the test rig. The piston crown is simulated by a flat plate placed at the bottom of the cylinder; this is crucial for the development of the tumbling motion. The tumble-to-swirl conversion tube was 6.5 cylinder bore diameters in length and connected to an outlet plenum chamber to ensure full development of the swirling motion. Both the cylinder and conversion tubes were made of perspex to permit LDA measurements of tumble and swirl distributions.

Air was sucked through the cylinder head arrangement by a centrifugal fan. A Ricardo viscous air flow meter was used to measure the air mass flowrate through the system. The pressure drop across the cylinder head arrangement was measured between inlet and outlet plenum chambers by a single limb water manometer, and controlled by a throttle attached to the outlet plenum chamber.

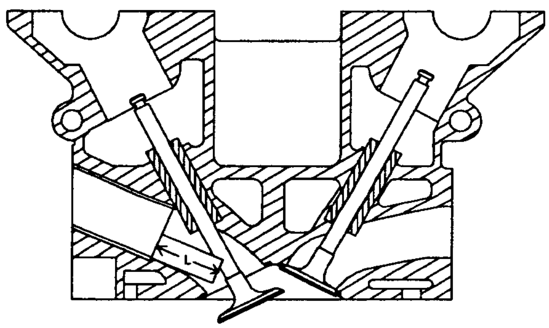

Figure 2 Four-valve cylinder head with pent roof chamber - Section A-A, showing tubes of length L

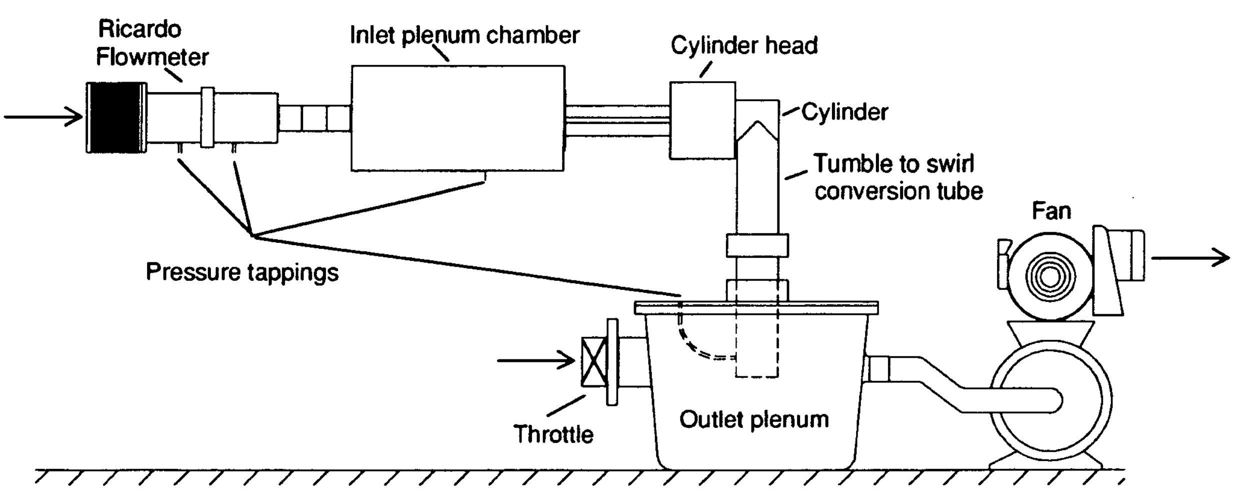

Figure 3 Schematic diagram of steady state air flow rig

EXPERIMENTAL TECHNIQUES

LDA OPTICAL CONFIGURATION - A single-component fibre-optic LDA system was used to measure the in-cylinder flow components (X and Y) at the various measuring locations along the spark plug axis (refer to fig. 1). Figure 4 shows the basic arrangement of the system which consisted of: (a) 5 Watt Argon-Ion laser, (b) an optical unit, built in-house incorporating miniature optics for beam splitting and miniature optics for frequency shifting, (c) both fibre-optically linked transmission and receiving optics. Table 2 shows specifications of the fibre-optic LDA system.

The fibre-optic probe was fixed to the cylinder head to eliminate spurious movements of the measuring volume resulting from engine vibration. The system was operated in back scatter where the light was scattered by particles of Titanium dioxide (TiO_2). The seeding was supplied from an air jet atomiser, which gave rise to an average particle diameter of 0.3 to 0.5 microns.

Table 2. Specifications of LDA system.

Wavelength	514 nm
Focal length	50 mm
Beam separation	8 mm
Beam intersection at half angle	4.07°
Measuring volume width	0.122 mm
Measuring volume length	1.52 mm
Calibration constant	3.23 ms^{-1} / MHz
Frequency shift	10 MHz

LDA SIGNAL PROCESSING AND DATA AQUISITION - The resulting Doppler signals were processed by a burst correlator (TSI IFA750) signal processor interfaced to a rotating machinery resolver (TSI 1990) and a microcomputer.

The burst correlator implements a double-clipped automatic correlation technique for measuring the frequency of coherent signals buried in back ground noise. This type of processor offers significant advantages over the counter processor which has been used in a majority of engine studies. Unlike other types of processor, a burst detector is used to locate signal bursts based purely on signal-to-noise ratio and not signal amplitude. Operation at a signal-to-noise ratios of at least -5dB was possible with this type of processor; a value that is far lower than the operating threshold of counter processors (4 dB). These advantages increase the potential of the LDA technique for performing a cycle resolved analysis of the velocity data.

The rotating machinery resolver was specifically designed to collect data from the IFA 750 signal processor during pre-set sectors of the engine cycle. The resolver was synchronised with the engine cycle using a crank shaft mounted optical encoder. A 500 pulse/rev, quadrature, incremental encoder was used to divide each cycle into 2000 equal angular segments providing a maximum resolution of 0.18 crank angle degrees.

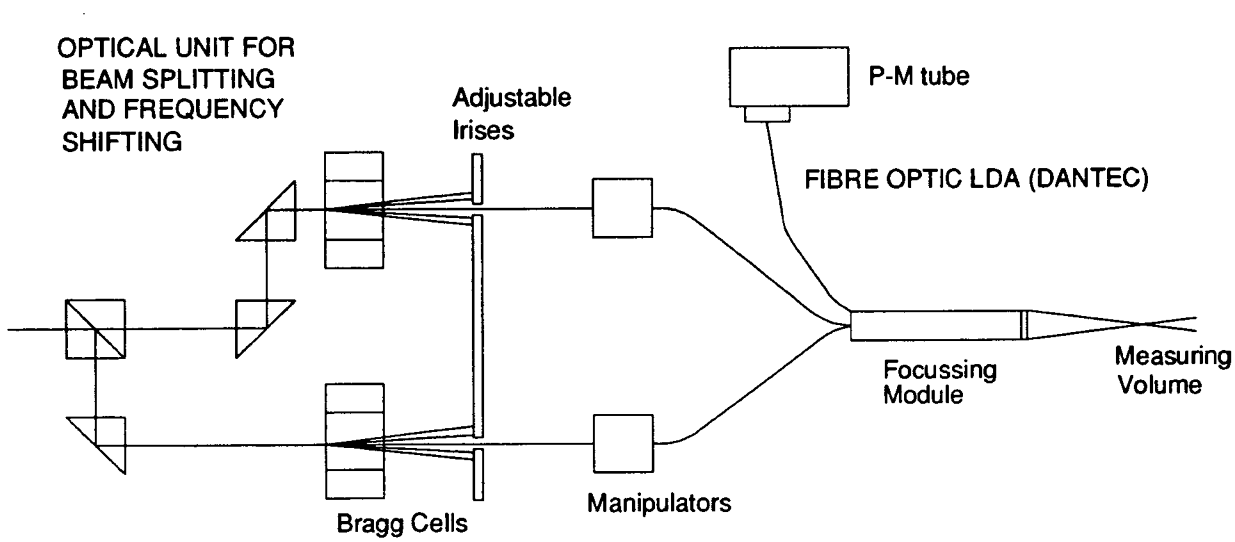

Figure 4 Schematic diagram of fibre-optic Laser Doppler Anemometer

RESULTS AND DISCUSSION

STEADY FLOW RIG TESTS - During this series of tests the pressure drop measured across the cylinder head was held constant at 64 mm of H_2O. This pressure drop equates to an equivalent engine speed of 1000 rev/min when considering mass flow rate measurements for various valve lifts and the relationship between the valve lift and crank angle for the camshafts supplied with the cylinder head.

EFFECTS OF MODIFIED PORTS ON TUMBLING RATIO - To define the degree of tumble, the tumbling vortex ratio (TVR) at intake valve closure was calculated according to the procedures developed by Arcoumanis *et al* in 1993 [29] and Baker *et al* in 1995 [32].

By integrating the time-averaged swirl velocity profiles (measured using LDA) obtained at various valve lifts a set of non dimensional tumbling ratios (NRTs) for each valve lift were calculated. NRT was defined as the ratio of the flows angular momentum to its axial momentum, normalised by cylinder bore. NRTs were then integrated over the intake valve opening period to estimate TVR_0 at inlet valve closure. Further steady flow measurements conducted by the authors can be found in the literature [31,32].

In order to assess the effect of the measuring technique on the results obtained, an equivalent set of tests were conducted on a water flow rig where a paddle wheel was used to measure NRTs at the various valve lifts for the same Reynolds Number.

Figure 5 shows plots of NRT versus valve lift for the no tubes inserted and 30 mm tubes inserted for the four-valve pentroof combustion chamber. The magnitudes of the NRTs for the 30 mm tubes inserted are higher at valve lifts of 4,6 and 8 mm for both air and water flow rigs. For the air flow rig NRTs, the scatter observed for no tubes inserted is due to difficulties in determining the flow centres for low degrees of tumble. For water flow rig NRTs, the relatively lower value at 2mm valve lift for the 30 mm tubes inserted may be due to paddle wheel friction. Despite this discrepancy, there is a close agreement between results conducted on both air and water flow rigs.

The tumbling ratio at intake valve closure for 30 mm tubes inserted was estimated to be 1.4. The value is comparable to steady flow tumbling ratio measurements conducted by Arcoumanis *et al* in 1994 (TVR≈1.7) and Hadded et a] in 1990 (TVR≈1.14) [21,12]. Hence, steady flow rig tests conclude that the insertion of tubes into the cylinder head's induction tracts increase tumbling capability.

EFFECT OF MODIFIED PORTS ON DISCHARGE COEFFICIENT - To assess whether incorporating the tubes effected the breathing capacity of the engine it was necessary to calculate the discharge coefficient. The latter was defined as the ratio between the measured mass flow rate through the inlet ports for a constant pressure drop at each valve lift, to the isentropic mass flowrate though a nozzle with a throat area equal to the inlet valve curtain area at the same pressure drop.

Figure 6 shows the variation of the discharge coefficient (Cd) for various valve lifts at 64 mm of H_2O for the cylinder heads with no tubes and 30 mm tubes inserted. The most important observation is that there is a net reduction in Cd with the addition of the 30mm inserts, showing that the tubes restrict the breathing capacity of the engine. This restriction may reduce power at full load, however the effect would be less significant at cruising conditions. One possible explanation for the increase at low valve lifts could be flow reattachment to the valve seat allowing an increase in the effective flow area and hence an increase in discharge coefficient.

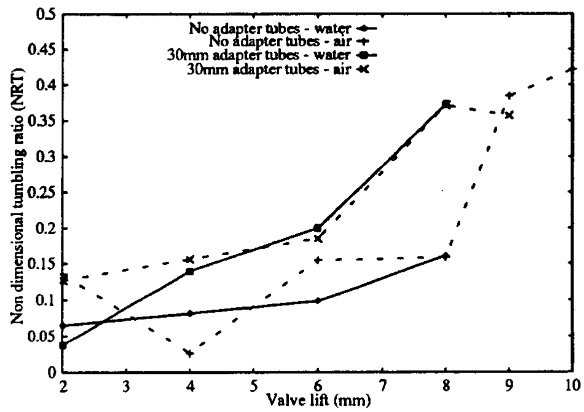

Figure 5 Non Dimensional rig tumble comparison for no tubes and 30 mm tubes inserted.

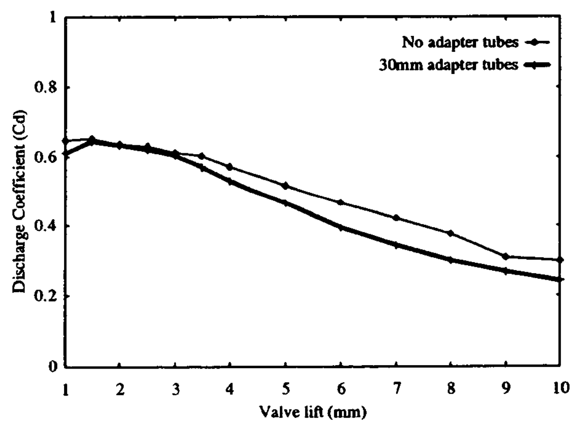

Figure 6 Discharge coefficient comparison for no tubes and 30 mm tubes inserted.

MOTORED ENGINE TESTS - The following results were mainly obtained at an engine speed of 1000 rev/min at wide-open-throttle with a compression ratio of 6.2:1. A few measurements were also conducted at 800 rev/min, 1600 rev/min in order to examine the effects of engine speed on the bulk and turbulent flow fields.

The bulk and turbulent flow fields were characterised by conventional ensemble arithmetic averages of the mean and r.m.s velocities respectively conducted within consecutive crank angle windows ($\delta\theta$ = 3.6 deg) over the total number of engine cycles (N_c = 50 - 130).

It must be noted that the definition of the ensembled r.m.s velocity assumes that cycle-to-cycle mean velocities remains constant and equal to the ensembled mean. Under certain flow conditions, however, this assumption is not valid and can lead to serious overestimates of the levels of turbulence. These flow conditions include measurements in regions where there are steep temporal variations and flow reversals. In order to differentiate between cycle-to-cycle flow variations and turbulence levels, the ensemble analysis was complemented by a cycle resolved analysis.

The main sources of statistical error in the LDA motored engine tests were velocity gradient and velocity biasing, crank angle broadening, and uncertainties due to finite sample size in regions of high turbulence intensity [33]. In this present study, velocity gradient and velocity biasing errors were difficult to determine. Experimental repeatability tests conducted at different data collection rates were therefore carried out to encompass these errors. Errors due to finite sample size were minimised by ensemble averaging over a 3.6 degree crank angle. These errors appeared as ripples superimposed on the mean and r.m.s velocity versus crank angle traces. The crank angle broadening error was found to be negligible for the 3.6 degree window, as it was found only to take effect on the ensembled r.m.s velocities at windows size above 14.4 degrees. Overall, both mean and r.m.s velocities could be measured to within an accuracy of ±10%.

EFFECT OF ENGINE SPEED ON ENSEMBLE MEAN AND R.M.S VELOCITIES - At realistic engine speeds it is well known that the in-cylinder flow field is fully turbulent and exhibit a Reynolds Number independence. There is also evidence that the in-cylinder flow field in a motored engine scales with engine speed in the absence of cycle-to-cycle flow variations [34]. For this reason, it is common to normalise the mean and r.m.s velocities by either; the mean engine speed or mean piston speed (Vp). The values of Vp at 800, 1000 and 1600 rev/min were 2.37, 2.97 and 4.75 m/s respectively.

Figures 7a and 7b shows a comparison of ensembled mean and r.m.s velocities normalised by the mean piston speed versus crank angle for both the induction and compression strokes at engine speeds of 800, 1000 and 1600 rev/min at a compression ratio of 6.2:1. The comparison is for the 30 mm tubes inserted, where velocity measurements are resolved in the X direction at the centre (z=10mm) of the TDC clearance height (refer to fig. 1) .

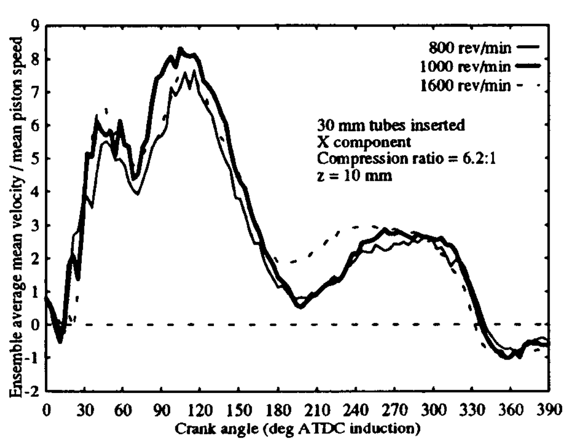

Figure 7a Effect of engine speed on mean velocities

The two sets of data show similarities in the ensemble mean and r.m.s velocities. The speed scaling for the mean velocities, however, is less apparent than for the r.m.s velocities. There are two possible explanations for any deviation of the mean flow from scaling with engine speed in motored engines; these are compressibility effects in the induction system at high speeds and back flow through the intake valves at low speeds [34].

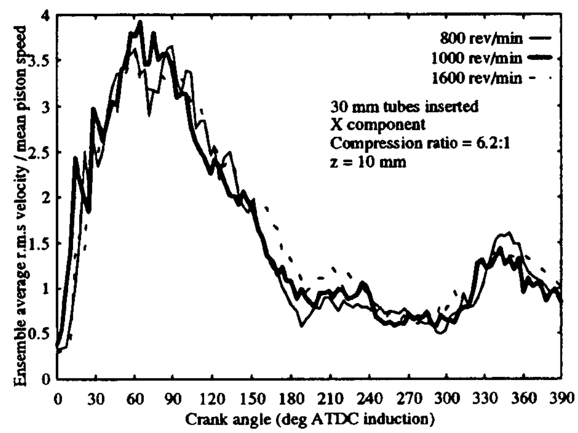

Figure 7b Effect of engine speed on r.m.s velocities

EFFECT OF MODIFIED PORTS ON ENSEMBLE MEAN VELOCITIES - Figures 8a and 8b show equivalent plots of the ensemble average mean velocity versus crank angle for both the X and Y components during induction and compression strokes at 1000 rev/ min.

In both cases, it can be seen that the X component is the dominant flow direction. A major observation, for the 30mm tubes inserted, is the flow reversal towards the end of the compression stroke where positive and negative values of the X component are indicative of a higher tumbling motion.

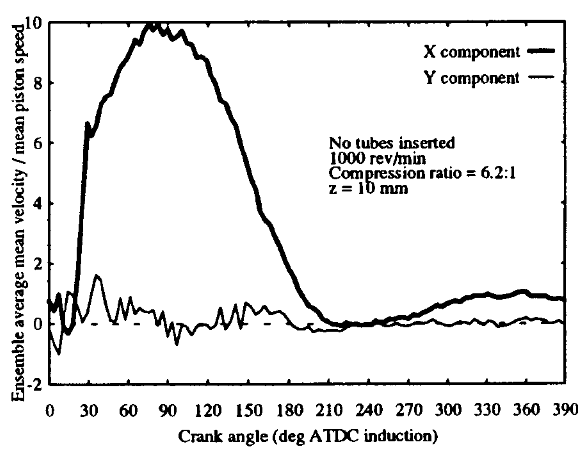

Figure 8a Comparison of mean X and Y components Pent roof chamber - no tubes inserted.

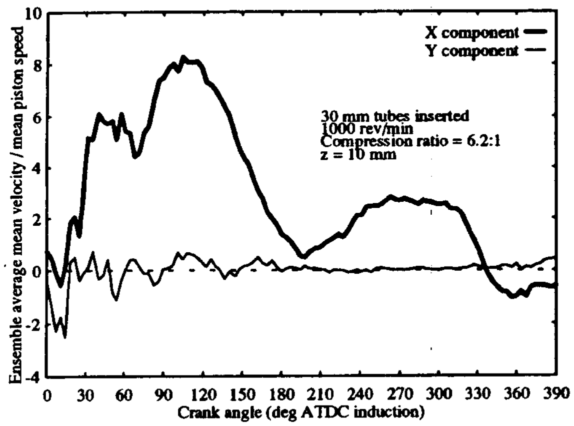

Figure 8b Comparison of mean X and Y components Pent roof chamber - 30 mm tubes inserted

Figures 9a and 9b show velocity vector diagrams at various crank angle instances with corresponding piston position superimposed. The X velocity measurements along the spark plug axis for the no tubes and 30 mm tubes inserted are plotted on this figure.

For 30 mm tubes inserted, it initially appears that flow reversals observed towards the end of the compression stroke are attributable to the passage of the tumbling vortex through each measuring location as it moves towards the apex of the pentroof chamber. However, at top-dead-centre of the compression stroke all velocity vectors measured along the spark plug axis reverse indicating a residual motion from the direction of the exhaust valves to the intake valves. On the other hand, for no tubes inserted, all velocity vectors measured are directed from intake to exhaust valves. Similar observations were also made by Hadded *et al* in 1991 [12]. In both cases, it is not natural for a flow to be present from exhaust valves to intake valves or vice versa without any returning flow.

In light of the literature review conducted by the authors, these LDA measurements can be open to interpretation. Figure 10 shows how the above observations can be explained if the tumbling motion is thought to be breaking down into two counter rotating vortices in the horizontal cylinder plane; Kiyota *et al* in 1993 and Kuwahara *et al* in 1994 [17,19]. For 30 mm tubes inserted, these vortices share a common stream line which runs from exhaust side to intake side. The results for the no tubes inserted suggest the opposite effect; Le Coz el al in 1990, Trigui *et al* in 1994 [10,20]. Hence it appears that the rotational sense of the vortices is sensitive to tumbling ratio. The break down of the tumbling vortices into this horizontal swirling structure would impose a limit on turbulence production.

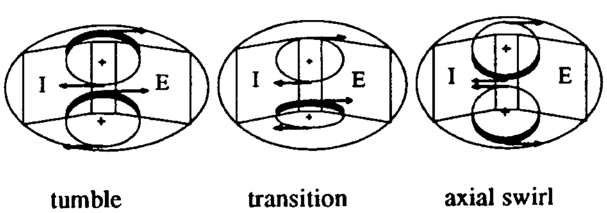

Figure 10 Illustration of barrel swirl breaking down into two counter rotating horizontal vortices

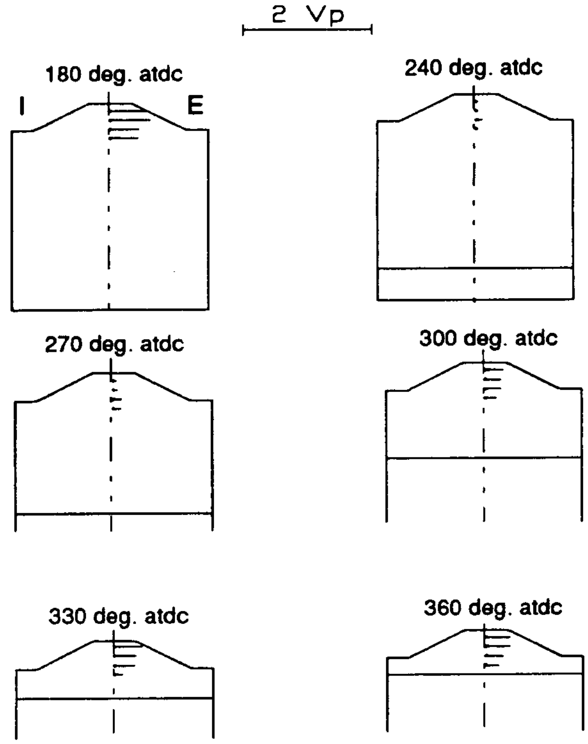

Figure 8b Comparison of mean X and Y components Pent roof chamber - 30 mm tubes inserted

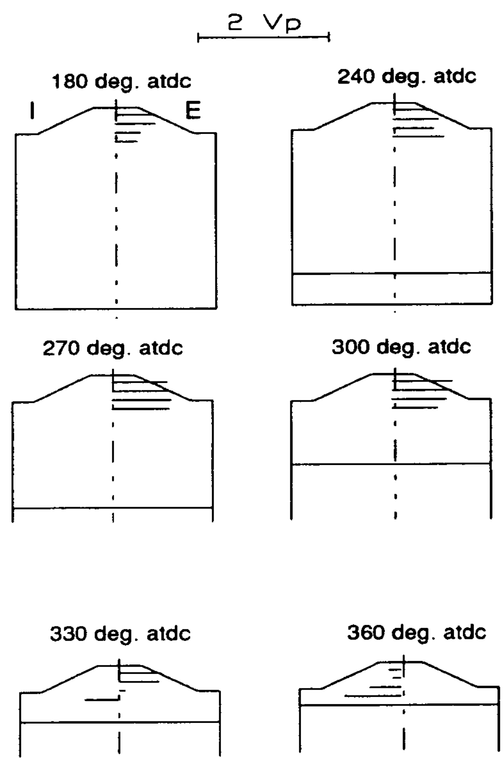

Figure 9b Velocity vector diagrams- 30 mm tubes inserted

EFFECT OF MODIFIED PORTS ON ENSEMBLE R.M.S VELOCITIES - Figures 11a and 11b show the equivalent data for ensemble average r.m.s velocities for the no tubes and 30 mm tubes inserted over induction and compression strokes. The velocity magnitudes are, once again, normalised by mean piston speed (Vp = 2.97 m/s) at 1000 rev/min.

For the 30 mm tubes inserted, the ensemble r.m.s velocities are seen to increase in value towards the end of the compression stroke. At this point the mean (X) velocities were seen to decrease and reverse towards the intake valves. The observed r.m.s velocity increase could therefore be turbulence generation due to the increase in the mean (X) spatial gradient, or due to cycle-to-cycle flow variations about the point of flow reversal.

It is interesting to note that the increase in the r.m.s velocity for the Y component occurs slightly later during the compression stroke than for the X component. This observation supported by the smaller spatial gradients in the Y component may indicate rapid distribution of turbulent kinetic energy from the X to Y components.

EFFECT OF MODIFIED PORTS ON TURBULENCE LEVELS - The instantaneous velocity data was resolved on a cycle-to-cycle basis to determine if the cycle-to-cycle mean remains constant and equal to the ensemble averaged mean.

Figures 12a and 12b show a comparison of the instantaneous velocity versus crank angle traces for four consecutive engine cycles for both no tubes inserted (fig. 12a) and 30 mm tubes inserted (fig. 12b).

Traces representing the ensembled mean (Emean) and Emean ± ensembled r.m.s (Erms) over the crank angle range of interest are also included on both figures. To show the differences between the intensity of in-cycle turbulent fluctuations with and without tubes inserted the Emean ± Erms bounds for no tubes inserted are plotted onto the cycle traces for 30 mm tubes inserted.

The most important observation is that the cycle-to-cycle flow variability at the measuring volume is higher with 30 mm tubes inserted than for no tubes inserted. This may be due to: (i) phase changing in the barrel swirl breakdown, (ii) precession of the counter rotating vortices in the horizontal cylinder plane. The rotational sense of the counter rotating vortices and cyclic stability therefore both appear to be sensitive to tumbling ratio. This observation has not been predicted by CFD calculations in the literature and has not been reported in previous experimental work.

It can also be seen that the intensity of turbulent fluctuation with 30 mm tubes inserted is higher than for no tubes inserted. Hence, the turbulence levels have increased with tumbling ratio. There is, however, very little evidence of turbulence enhancement towards the latter stages of the compression stroke for the 30 mm tubes inserted. This observation indicates that the previously observed increase ensemble average r.m.s (see figs. 11a,b) towards the end of the compression stroke is weighted by cycle-to-cycle flow variation and thus provides an overestimate of turbulence intensity.

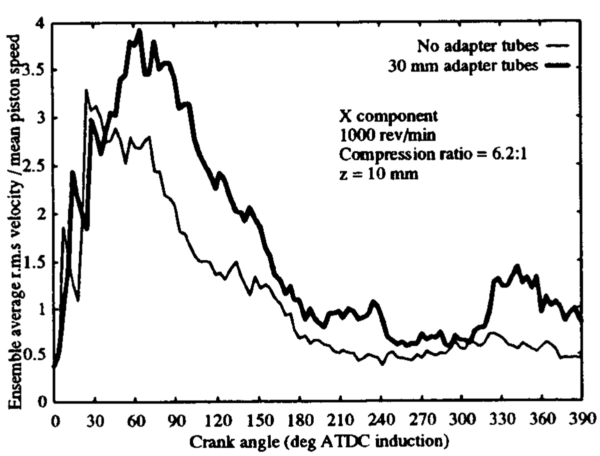

Figure 11a Effect of modified ports on ensemble r.m.s velocities - X component.

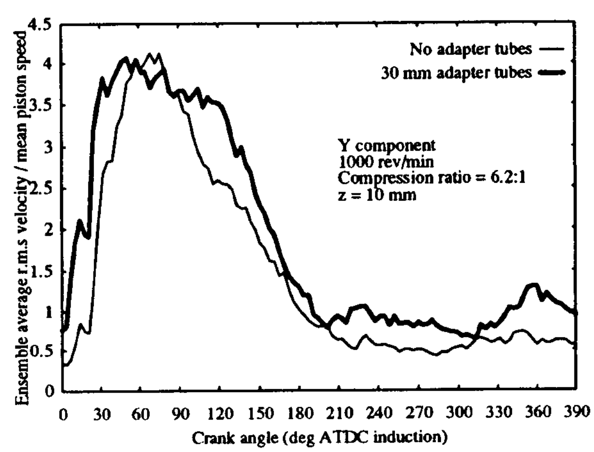

Figure 11b Effect of modified ports on ensemble r.m.s velocities - Y component.

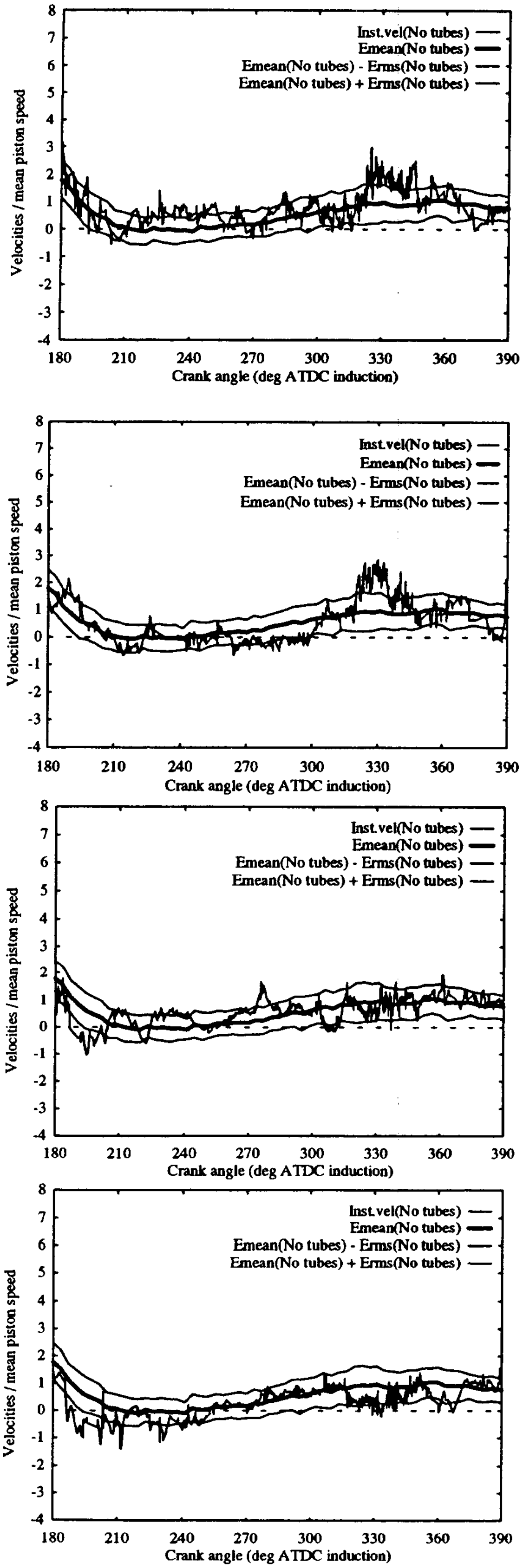

Figure 12a Four consecutive engine cycles - no tubes inserted, 1000 rev/min, 6.2:1, z = 10 mm

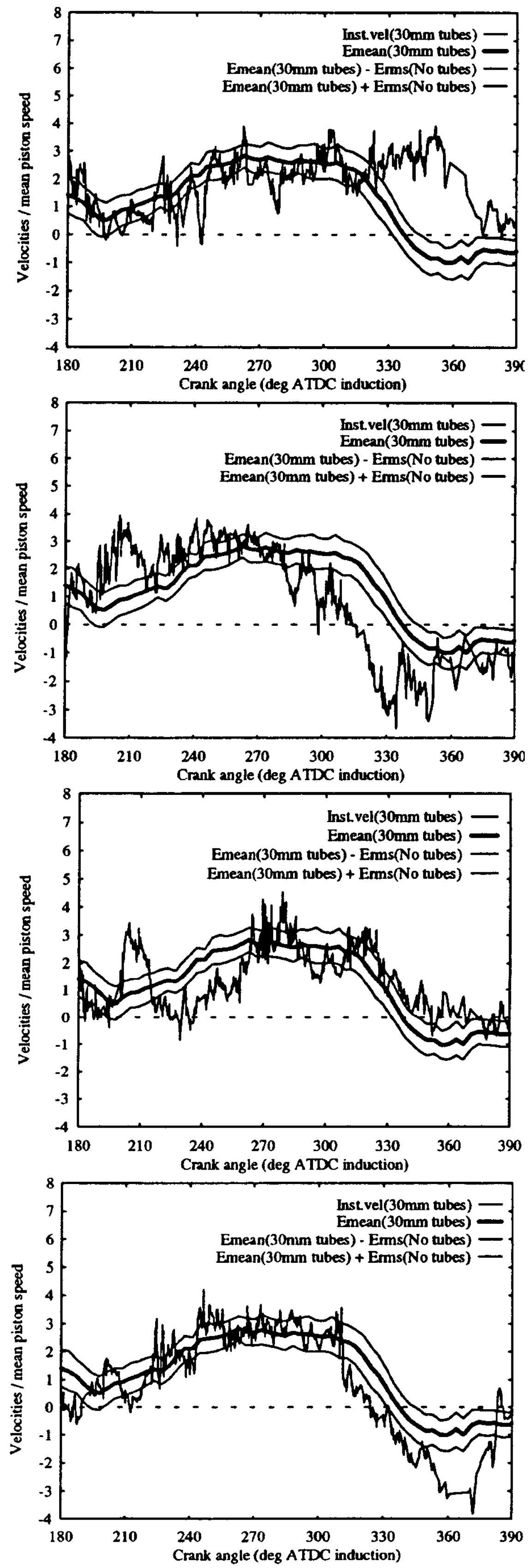

Figure 12b Four consecutive engine cycles - 30 mm tubes inserted, 1000 rev/min, 6.2:1, z = 10 mm

IMPLICATIONS FOR COMBUSTION AND BARREL SWIRL OPTIMISATION - The turbulence levels and cycle-to-cycle flow variations towards the end of the compression stroke increased with tumbling ratio. The former suggested faster combustion rates if applied to a lean burn engine, however, the latter suggested variations of ignition delay and thus greater cyclic combustion instability which may limit lean burn capability.

The resultant mean velocities measured towards the end of the compression stroke for 30 mm tubes inserted (refer to fig. 9b), are within Pischinger and Heywood's [26] recommended mean velocity range of 3-5 m/s at 1000 rev/min (equivalent to 1.0 - 1.7 Vp). Their recommendation was based on the argument that the local velocity magnitude should be large enough to convect the flame away from the electrodes but not so large as to quench the flame due to excessive stretching. The residual mean velocities associated with the two counter rotating vortices in the horizontal plane also scale with engine speed (fig. 7a). Above engine speeds of 1000 rev/min, the optimum velocity range for assisting burning rates may be reached (3-5 m/s) after which quenching and consequently engine misfire may result; Kuwahara *et al* [19].

The break down of the tumbling motion into two counter rotating vortices in the horizontal cylinder plane appears to be a phenomenon exclusive to four-valve engines. There is evidence in the literature that this may be due to momentum transfer from the decaying tumbling vortex to motions present in the horizontal cylinder plane; Trigui *et al* in 1994 [20] and Arcoumanis *et al* in 1994 [21].

It is suggested that the insertion of partitions into the 30 mm induction tubes may reduce secondary flow structures in the horizontal plane emanating from the intake porting arrangement; Kuwahara *et al* in 1994 [19]. If the two induction jets are oriented as close as possible to the mid-plane between the valves, turbulence mixing between the two induction jets may also improve and result in a purer tumbling motion which should break down more effectively into turbulence; Newman *et al* [35]. This may be achieved by using different geometrical variations and orientations of the ramps inserted into the cylinder head's induction system.

CONCLUSIONS

Steady flow analysis showed that the insertion of tubes into the cylinder head's induction tracts increased the tumbling ratio of the in-cylinder flow field at intake valve closure from 0.4 to about 1.4. The motored in-cylinder results revealed new insights into barrel swirl behaviour in four-valve engines.

1./ The results, supported by the literature survey, suggested that the barrel swirl generated was not efficiently breaking down into turbulence but forming two counter-rotating vortices in the horizontal cylinder plane. The rotational sense of the vortices appeared to be sensitive to tumbling ratio.

2./ The turbulence levels and cycle-to-cycle flow variations towards the end of the compression stroke also increased with tumbling ratio. The former suggested faster combustion rates if applied to a lean burn engine, however, the latter suggested variations in ignition delay and greater cyclic combustion instability which may limit lean burn capability. Hence, both the rotational sense of the vortices and their cyclic stability appear to be sensitive to tumbling ratio.

Optimisation of existing and new four-valve engine designs for meeting future emission targets may be achieved if barrel swirl can be broken down more effectively into turbulence. Further work is being conducted by the authors on the effects of intake port configuration / combustion chamber shape on barrel swirl behaviour. The effects of different geometrical variations / orientations of the ramps inserted into the cylinder head's induction system are currently being assessed.

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