Experimental study on an IC engine in-cylinder flow using different steady-state flow benches

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Abstract
In-cylinder air flow structures are known to strongly impact on the performance and combustion of internal combustion engines (ICE). Therefore the aim of this paper is to experimentally study an IC engine in-cylinder flow under steady-state conditions. Different methods can be used to characterize the in-cylinder flow which are optical engines and laser diagnostics, computational fluid dynamic and steady-state flow bench. Here we are concentrating on two different types of flow benches. The first (Ricardo) uses the impulse torque meter method while the other (FEV) uses the paddle wheel technique. The experiments were carried out on the same cylinder head and the same pressure difference across the inlet valves of 600 mmH2O in order to compare the results. The experimental results are presented in terms of the measured air flow rate, flow coefficient, discharge coefficient and non-dimensional rig tumble. Moreover, number of modifications were conducted on the FEV flow bench in order to apply particle image velocimetry measurements on the vertical tumble plane, which passing through the middle of the cylinder at different valve lifts. The results show that a reasonably good level of agreement can be achieved between both methods, providing the methods of calculations of the various parameters are consistent.

1. Introduction
It is generally acknowledged that there are two major challenges facing the automotive industry, continuous increase in fuel prices and restricted emission regulations [1,2]. Therefore, the automotive industry has put extensive energies over the last decades in assessing air flow during the intake stroke and air flow within the cylinder as it is generally accepted that it has significant effect on the in-cylinder mixture preparation and hence the combustion performance. Large scale flow structures such as swirl and tumble increase the level of turbulence at the time of ignition which, in turn, strongly affect both pollutant emissions and fuel efficiency [3–5].

Fundamentally, in-cylinder flow is divided into two main categories, swirl and tumble. Swirl is the in-cylinder flow where the flow rotational axis is parallel to the cylinder axis while tumble is the flow with an axis perpendicular to that of the cylinder. Steady-state flow benches have been utilized in the
and to compare the various cylinder heads designs [6,7]. There are three air motion axes along which steady-state flow bench measurements are taken. These are swirl, normal tumble (sometimes referred to as barrel swirl) and side tumble as shown in Fig. 1. Swirl measurements is most commonly applied to two valves (2 V) and four valves (4 V) diesel engines and to two valves gasoline engines, while normal tumble (referred to as tumble) is most commonly applied to pent roof four valves gasoline engines [8]. The main principle of steady-state flow bench was described in details by Heywood [9]. With the appearance of intake generated tumble in SI engines, the standard steady flow benches had to be modified to measure the new type of flow.

The development and adaptation of steady flow rig “barrel swirl” measurements was described by [10]. The authors established a standard test procedure for multi-valve tumble engines and compared their results with hot wire anemometry (HWA) measurements. In their opinion, steady flow rigs could be used reliably as standard tools for the measurement of barrel swirl in engines. Several years later, the variations in different non-dimensional parameters such as swirl ratio and flow coefficient at various throttle opening and valve lifts were studied using a steady-state flow bench by Ramesh Kumar [11]. The results indicated that a higher swirl ratio and swirl coefficient can be achieved with shrouded valves and twisted tapes but with a penalty on the flow coefficient. Unfortunately no standardized testing methodology exists at present and great care has to be taken when comparing data coming from different sources [12–14]. Hongming Xu illustrated a comprehensive review of the most widely adopted techniques [15].

Many measurement and visualization techniques have been used to study numerous features of the in-cylinder flow of GDI engines, particularly Particle Image Velocimetry (PIV) because of the need for whole field flow visualization and measurement of the characteristics of the complex unsteady in-cylinder flows. PIV is a class of methods used in experimental fluid mechanics in order to determine instantaneous velocity vector fields by measuring the displacement of numerous fine particles that accurately follow the motion of the fluid. Basically, almost all quantitative measurements of fluid flow before the advent of PIV were carried out using single-point probes that measure different quantities in the flow for instance, velocity, and temperature and pressure measurements [16,17]. These single-point measurements have number of demerits because of measurements are carried out only at single-point at a time and information about the underlying flow pattern is missing. Typically the experimental set-up of a PIV system involves several subsystems, seed particles, illumination source, imaging system and processing unit (PIV Processor). Mainly, the flow has to be seeded with a suitable type of tracer particles having a similar density to that of the fluid. Within a short time interval, these tracer particles have to be illuminated twice using high power double pulsed laser. The scattered light from the particles has to be captured either on double frames or on single frame using Charged Coupled Device (CCD) camera. The image frames are sub-divided into smaller areas called “interrogation area” and the displacement of the tracer particles between the light pulses has to be determined through means of statistical methods (auto- and cross-correlation). The velocity associated with each interrogation area is the displacement over the time between the two exposures. Previous PIV measurements in IC engines have characterized turbulence properties [18], analysed spatial flow structure [19], investigated influences of cycle-to-cycle variations [20], as well as characterized flows during injection and ignition [21]. B.M. Krishna carried out an experimental investigation in order to study the in-cylinder tumble motion during the intake and compression strokes at different engine speeds using PIV. The results showed that, the tumble ratio was not significantly affected by the engine speed but mainly changed with the crank angle [22]. The behavior of the in-cylinder flow pattern under steady-state conditions and different air flow rates was characterized by maximum turbulent kinetic energy (TKE) and tumble ratio by B.M. Krishna [23]. It was found that the tumble ratio increased with inlet valve opening but was not influenced much by the variation in air flow rates.

The first objective of this paper is to compare two different methodologies for evaluating in-cylinder flow using the same cylinder head, reference area and pressure difference. While the second objective is to illustrate the evolution of tumble motion using particle image velocimetry (PIV).

2. Experimental set up

2.1. Cylinder head general specifications

The cylinder head specifications are shown in Table 1. A four valves pent roof cylinder head was used on both flow benches. All port flow calculations had to be based on one characteristic dimension which could be applied to all ports, thus the inner seat diameter was used as reference. Moreover the pressure drop used to undertake the tests was selected based on Reynolds number criteria. 600 mmH2O pressure difference was selected depending on valve inner seat diameter of 33.7 mm to insure fully turbulent flow. As it is well known that the tumble motion became the dominant motion in four valves spark ignition engines so this study was concentrated only on tumble.
measurements. In industry the most widely adopted instruments for tumble measurement are the paddle wheel and impulse torque-meter.

2.2. Torque meter steady-state flow bench

In this rig the flow was blown through the valves to assess inlet ports, by providing pressurized air at the manifold inlet and discharging to atmosphere through the valve and cylinder liner. A schematic diagram of the flow test equipment is shown in Fig. 2 for the tumble test set up. The test equipment consisted of a centrifugal fan driven by a continuously variable speed direct current (DC) motor. The volume flow rate was measured using a Viscous Flow Air Meter (VFAM). A pressure box was attached to the cylinder head manifold face which was designed to act as a reservoir of air under conditions approaching stagnation. It contained flow straighteners to eliminate any directional flow influence from the flexible pipe. Pressure and temperature were measured in the entry box and were used as the reference conditions for the calculations of flow parameters. Bulk air motion was quantified by an impulse swirl meter (ISM). For Tumble Measurement-Procedure, the cylinder head was mounted with the gas face vertical, and a tumble rig was attached via an adaptor plate mounted on the gas face of the cylinder head. The ISM is mounted on the top of the tumble tube. At the lower end of the tube a second piece of honeycomb element was mounted to produce an equal flow resistance to the swirl meter element on the top of the tumble tube.

2.2.1. Parameters used in port performance analysis [8]

- **Flow Coefficient** ($C_f$)

\[
C_f = \frac{Q}{A_{seat} \times V_o} \tag{1}
\]

\[
V_o = \sqrt{\frac{2 \times \Delta P}{\rho}} \tag{2}
\]

- **Coefficient of Discharge** ($C_d$)

\[
C_d = \frac{Q}{A_V \times V_o} \tag{4}
\]

\[
A_V = n \times \pi \times D^2 \times \left(1 + \frac{L}{D} \sin \varnothing \cdot \cos \varnothing\right) \tag{5}
\]

- **Non-dimensional rig tumble** ($N_T$)

\[
N_T = \frac{8 \times G}{m \times V_o \times B} \tag{6}
\]

2.3. Paddle wheel steady-state flow bench

The principal operation and design of FEV steady tumble test rig is shown in Fig. 3. For the measurements, the cylinder head (10) is mounted to a cylindrical liner (7) having the same diameter as the cylinder bore. Air is sucked by a centrifugal compressor (1) through the inlet port (21), the cylindrical tube, the compensation tank and through the rotary piston gas meter (5). The valve lift can be adjusted manually using micrometer (6). The measurements are carried out at a constant pressure difference between cylindrical liner and atmosphere. The pressure adjustment, which is required for different valve lift settings is done by a bypass (2) with stepper motor (3) with additional measurement of air temperature and pressure drop. The tumble level is gained by assessing the rotational speed of the paddle wheel anemometer (7). The test bench is equipped with a test bench computer for electronic acquisition (8) (air volume flow rate, paddle wheel rotation speed, temperatures, pressures) and a test bench control (bypass with stepper motor). For tumble measurement-procedure, the test method is based on the paddle wheel anemometry with a horizontal axes of rotation in order to determine the rotation of a vortex perpendicular to the cylinder axis (tumble). Due to the fact that the piston crown plays an important role for the development of the tumble vortex, the piston (13) is included in the test rig.
2.3.1. Parameters used in port performance analysis [24]

- **Flow Coefficient**

The flow coefficient ($C_f$) is defined as the ratio of the empirically obtained mass flow rate and the theoretical mass flow rate:

$$C_f = \frac{\dot{m}_{\text{real}}}{\dot{m}_{\text{theor}}} \quad (7)$$

The actual mass flow rate is measured on the test bench:

$$\dot{m}_{\text{real}} = Q \times \frac{P_1}{R \times T_1} \quad (8)$$

The theoretical mass flow rate ($\dot{m}_{\text{theor}}$) for a defined cross sectional area ($A_{\text{Seat}}$) is obtained as:

$$\dot{m}_{\text{theor}} = A_{\text{Seat}} \times \rho_s \times C_s \quad (9)$$

The flow velocity ($C_s$) is calculated with the formula for isentropic flow:

$$C_s = \sqrt{\frac{2 \times k}{k - 1} \times \frac{R \times T_1}{P_2} \times \left[ 1 - \left( \frac{P_2}{P_1} \right)^{\frac{k-1}{k}} \right]} \quad (10)$$

Likewise, the density under isentropic conditions is computed as:

$$\rho_s = \frac{P_1}{R \times T_1} \times \left( \frac{P_2}{P_1} \right)^{\frac{k}{k-1}} \quad (11)$$

- **Coefficient of Discharge ($C_d$)**

The discharge coefficient ($C_d$) is defined as the ratio of the empirically obtained mass flow rate and the theoretical mass flow rate:
The theoretical mass flow rate \( m_{\text{theor}} \) for a defined orifice area between valve head and seat at low valve lifts \( A_V \) is obtained as:

\[
m_{\text{theor}} = A_V \times p_s \times C_s
\]  

### Non-dimensional rig tumble

In order to describe the measured tumble intensities independently from the mass flow, the tumble ratio is indicated as a non-dimensional quantity \( C_T \) (\( C_T \): circumferential velocity of the tumble motion, while \( C_s \): mean axial velocity in the cylinder).

The linear circumferential speed of the tumble motion is computed as follows:

\[
C_T = 2 \times \pi \times N \times R_{MFL}
\]  

Likewise, the axial velocity of the air flow in the cylinder is calculated as:

\[
C_s = \frac{m_{\text{real}}}{\rho_{\text{cyl}}} \times A_V
\]  

\[
\rho_{\text{cyl}} = \text{air density inside the cylinder, Kg/m}^3.
\]

2.4. Particle image velocimetry (PIV)

Number of modifications was essential on the FEV steady-state flow bench before applying PIV measurements. These included the removal of the paddle wheel, and designing of acrylic box to connect between the cylinder head and the tumble rig to allow enough extended area for the whole stroke measurements as shown in Fig. 3. Full velocity vector maps were obtained in one vertical (tumble) plane (passing through the centre of the cylinder, which was located between the two intake valves) at different valve lifts 2 mm, 5 mm, 8 mm and 9 mm valve lift. Titanium Dioxide was used as seeding particles and generated by means of a solid particle seeder and mixed with air through the inlet port. A doubled pulsed Nd: YAG laser with a wavelength of 532 nm, 15 Hz of maximum laser pulse frequency and a capacity of 400 mJ/4 ns was used as light source. For every valve lift, 208 pairs of images were acquired using a digital camera (Flow Sense 2M) running in double frame mode. The final PIV images (1600 \( \times \) 1200 pixels, 8 bit grey scale) were saved directly to the hard drive for analysis. The post processing of the acquired images was carried out using Dynamic studio V3.41 software in order to obtain the velocity vector fields.

Non-dimensional tumble for PIV data was derived from the angular momentum equation around the center of rotation of the paddle wheel \( (x_c, y_c) \) used for FEV flow bench experiments in order to calculate the circumferential velocity of the tumble motion. While the axial velocity was calculated same as in flow bench experiments.

\[
\text{Circumferential Velocity} = \sum_{i=1}^{m} \sum_{j=1}^{n} \left( w_{ij} \times (x_c - x_{ij}) - u_{ij} \right) \times (y_c - y_{ij})
\]
3. Results and discussions

All experiments were carried out at the same pressure difference, 600 mmH₂O. The valve lift was changed from 1 to 9 mm in 1 mm step and the experimental results are presented in terms of measured air flow rate, flow coefficient, discharge coefficient and non-dimensional rig tumble.

3.1. Measured air flow rate

The variation of air volume flow rate as a function of valve lift for both Ricardo and FEV steady-state flow benches at 600 mmH₂O pressure difference are shown in Fig. 4. In general, as can be seen from the figure, air volume flow rate increased as the valve lift increases. There is an acceptable agreement between the two curves till 6 mm valve lift but the difference increase further as the valve lifts increases. This is mainly due to choking of the flow due to the small discharge area in case of the FEV flow bench. This is thought to be due to the recommended value of the air discharge port being 35% of the bore diameter.

3.2. Flow coefficient

The flow coefficient was measured using both steady-state flow benches at different valve lifts as illustrated in Fig. 5. The flow coefficient increased monotonically from zero with valve lift since the effective flow area through the valve increased with lift. As more air entered the cylinder at higher valve lift, the effect of the flow coefficient to the engine breathing capacity was significant. However, the small scale of flow coefficient at lower valve lift could not reflect clearly the difference in flow capacity. It is clear from the figure that there was also a good agreement between both curves especially till 6 mm valve lift.

3.3. Discharge coefficient

The dependence of the discharge coefficient which, identify flow restriction by valve and seat lips, on valve lift can be understood from Fig. 6. At low valve lifts, the inlet air jet was attached to both the seat and the valve, and thus was affected by viscous shear. If the jet was attached, then the discharge coefficient decreased slightly with increasing the Reynolds number since the viscous effects in the jet decreased. At high valve lifts, the fluid inertia prevented the flow from turning along the valve seat, so the flow broke away, forming a free jet. There is also a good matching between both curves.

3.4. Non-dimensional rig tumble

The variation of non-dimensional rig tumble versus the valve lift is shown in Fig. 7. At low valve lift, more air came from one side of the valve seat producing negative value (clock wise direction). After that, a symmetric flow distribution occurred at the valve seat area which resulted in no tumble at about 5 mm valve lift for paddle wheel flow bench and at about 4 mm for impulse torque meter flow bench. At higher valve lifts, an asymmetric flow distribution occurred again with the flow directed more towards the exhaust valves. Intensified by the deflection at the piston crown this jet flow led to the generation of a strong tumbling motion within the cylinder with positive values (counter clock wise direction). The difference between both curves was related to the fact that different techniques were used for tumble measurements, paddle wheel for FEV flow bench while an impulse-torque meter for Ricardo flow bench. Moreover, in the case of the torque meter method (Ricardo method) there was no piston but in the case of the paddle wheel the piston was there. The rate of increase of non-dimensional rig tumble for FEV flow bench became very...
high starting from 6 mm valve lift. This was mainly because of the arrival to choking condition therefore more air trapped inside the cylinder which, in turn, increased the rotation rate of the paddle wheel.

3.5. Particle image velocimetry results

3.5.1. Ensemble average velocity vectors for in-cylinder flow pattern

Fig. 8 shows the ensemble average velocity vector fields at different valve lifts. At valve lift 2 mm, there was a domination for the right air jet coming through the intake valves and the area behind the intake valves was the area where the max velocity was concentrated. This might explain the rotation of paddle wheel during the flow bench experiments in clock wise direction with lower negative non dimensional rig-tumble. At valve lift 5 mm, there was a symmetric velocity distribution behind both intake valves and exhaust valves, this also might
Fig. 9  Non-dimensional Rig-Tumble for FEV Flow Bench and PIV at different valve lifts.

Fig. 10  The variation of average TKE at different valve lifts.

Fig. 11  Vorticity magnitudes at different valve lifts.
explain the rotation stoppage of paddle wheel at this valve lift. With increasing the valve lift more, the domination of the left air jet became clear as more air directed towards the exhaust side the deflected by the cylinder liner then the flat piston, this led to a counter clock wise vortex motion (tumble). The tumble motion started firstly at valve lift 8 mm at the right bottom side of the cylinder then at valve lift 9 mm the tumble motion became more stronger and the center transferred towards the center of the cylinder.

3.5.2. Non-dimensional Rig-Tumble

Fig. 9 illustrates the non-dimensional rig-tumble calculated form both FEV flow bench and PIV experiments. It can be seen from the figure that the trend of non-dimensional rig tumble was the same for both cases. The difference might be related to the fact that PIV experiments were carried out at two dimensional plane while the paddle wheel experiments included all dimensions. Moreover, the calculated non-dimensional rig-tumble from PIV data depended on the ensemble average of all images at each valve lifts.

3.5.3. Average turbulent kinetic energy

One of the key factors responsible for air-fuel mixing specially in direct injection engines, flame speed and heat transfer is the turbulent kinetic energy (TKE) [25]. It is quantified by the average turbulent kinetic energy (Avg.TKE) which was calculated for the in-cylinder flow using Equation 18 of Reuss [26]. The Avg.TKE of the flow was calculated from the root mean square (RMS) velocity vector fields, which were acquired from ensemble average velocity fields by Dynamic studio V3.41 software, which in turn were gained from 200 instantaneous velocity vector fields.

\[
TKE = \frac{1}{2} \rho V^2_u = \frac{1}{2} \rho (u^2_{rms} + v^2_{rms}) \tag{18}
\]

where \(u_{rms}\) and \(v_{rms}\) are the RMS velocity components in x and y directions respectively, while the third component for 2D tumble motion was neglected [B]. The air density in this study \(\rho\) was assumed to be 1 Kg/m³.

Fig. 10 shows the plot of average turbulent kinetic energy at different valve lifts. It is clear from the figure that the magnitude of the average turbulent kinetic energy is increasing with increase in valve lift. This might be because of the increased air flow rate and velocity at higher valve lifts. That was finally lead to that there was a strong tumble motion by the end of intake stroke at higher valve lifts with high value of the average turbulent kinetic energy which was expected to break down into small scale structures by the end of compression stroke. Higher level of turbulence at the time of ignition is definitely expected to increase the flame propagation especially for direct injection engines.

3.5.4. Vorticity

Fig. 11 shows the vorticity magnitudes at different valve lifts. Green area represents the flow rotation in counter clock-wise direction while blue area represents the flow rotation in clock wise direction. It can be seen that at lower valve lifts, clock wise vortices existed behind the intake valves, with vorticity strength increased with the valve lift increasing. However, the domination of the left jet led to the formation of counter-clock wise vortices behind the exhaust valves firstly at lower valve lifts and then transferred to the lower center of the cylinder.

4. Conclusion

The aim of the present work was to investigate the in-cylinder tumble motion in, a four valves pent roof cylinder head using two different steady-state flow benches. The same pressure difference and reference area were selected for comparisons. Moreover, the FEV flow bench was modified in order to apply particle image velocimetry (PIV). Full velocity vector maps were obtained in one vertical (tumble) plane (passing through the centre of the cylinder, which was located between the two intake valves) at different valve lifts 2 mm, 5 mm, 8 mm and 9 mm valve lift.

From the experimental results the major conclusions that can be drawn are:

- A reasonably good level of the agreement was achieved between the two flow benches till about 6 mm valve lift.
- After 6 mm Valve lift, the flow inside the FEV flow bench started to choke, which in turn affected the value of both flow coefficient and discharge coefficient.
- This choking condition led to an increase in the intensity of rotation of the paddle wheel and resulting in higher rig tumble. This might also be due to the fact that in this method the piston was positioned inside the cylinder while the torque meter method did not have a piston.
- At higher valve lifts a strong tumble motion was generated with high value of non-dimensional rig-tumble, average turbulent kinetic energy and vorticity magnitude.
- A reasonably good level of the agreement was achieved between data obtained from the two flow benches and PIV experiments.

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[1] Commission, E. Regulation (ec) no 715/(2007) of the european parliament and of the council on type approval of motor vehicles with respect to emissions from light passenger and commercial vehicles (euro 5 and euro 6) and on access to vehicle repair and maintenance information.


