# Effect of Cavity in a Pentroof Piston on the Engine In-cylinder Tumble Flows – An Investigation by Particle Image Velocimetry

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***Abstract*—This study deals with the investigations on the in-cylinder tumble flows of a single cylinder two-valve engine under motoring conditions with two pentroof pistons having different cavities at an engine speed of 1000 rev/min., during suction and compression strokes using particle image velocimetry (PIV). The two-dimensional velocity vector fields are analyzed on a vertical plane passing through the axis of the cylinder. To quantify the tumble flows, tumble ratio (TR) and average turbulent kinetic energy (ATKE) are estimated from the ensemble average velocity vectors obtained from PIV measurements. It is found that, the TR and ATKE are higher for pentroof-offset-cavity piston than pentroof-central-cavity piston. The present study will be useful in understanding the effect of piston-cavity and shape on the nature of the in-cylinder tumble flows in real engine conditions.**

***Index Terms* —Pentroof, PIV, TKE, Tumble flow, Velocity field.**

## I. Introduction

TOday, the direct injection, stratification and lean-burn concepts are becoming more promising ways to improve the fuel economy and reduce exhaust emissions in the modern spark ignition (SI) internal combustion (IC) engines. Generally, the lean burning is associated with higher cycle-by-cycle combustion variations due to the lower flame initiation and propagation rates. A practical approach to overcome the above problem is to enhance mean flow and turbulence of the in-cylinder flows [1]. The in-cylinder flow field structures have a major influence on the combustion, emission and performance of an IC engine. Now-a-days, in-cylinder flow fields can be measured and analyzed very accurately using optical tool like particle image velocimetry (PIV) which has many advantages viz., non-intrusive, high spatial and temporal resolution and instantaneous visualization of the whole flow field.

So far many experimental and computational techniques have been tried to understand the in-cylinder flows in the IC engines. The in-cylinder rotational flow components parallel and perpendicular to the axis of the cylinder are termed as swirl and tumble respectively. In direct injection SI engine, the tumble helps to deflect the injected fuel plume directly into the cylinder towards the spark plug by the guidance of piston cavity.

Reference [2] reported that generating a significant vortex flows in an IC engine cylinder during the intake process generates high turbulence intensity during the later stage of compression stroke. Reference [3] measured the in-cylinder flows using particle tracking velocimetry (PTV) and reported that the swirl and tumble flows should be optimized for achieving good combustion. Reference [4] reported that in-cylinder rotating flows can significantly increase the turbulence intensity during the combustion period resulting in reduced burning period and increased thermal efficiency in the premixed SI engines. The PIV measurements on various engines reported that the in-cylinder flow structures change substantially along the cylinder length due to the geometry of the intake port and tumble motion during the intake stroke [5-8].

Previous investigations showed that a good understanding of the in-cylinder fluid flow structures of an IC engine is very much necessary for the optimization of the engine parameters for better performance. The in-cylinder tumble flows are very much dependent on the shape of the piston top, location of the piston-cavity, orientation of the intake manifold, compression ratio, engine speed etc. However, there is very a limited study on the effect of piston-cavity on the in-cylinder flow characteristics. Therefore, the present study deals with the in-cylinder tumble flow analysis of an IC engine to see the effect of piston cavity using PIV.

## II. Experimental Setup and Procedures

The experimental investigations are carried out on a single-cylinder, two-valve engine under motoring conditions at an engine speed of 1000 rev/min (Experimental setup:Fig.1 and Photographic view: Appendix), using two pentroof pistons having central and offset cavities as shown in Fig.2 (hereafter designated as pentroof-central-cavity and pentroof-offset-cavity pistons). The engine specifications are given in the Table I. The original metal cylinder liner is extended to a

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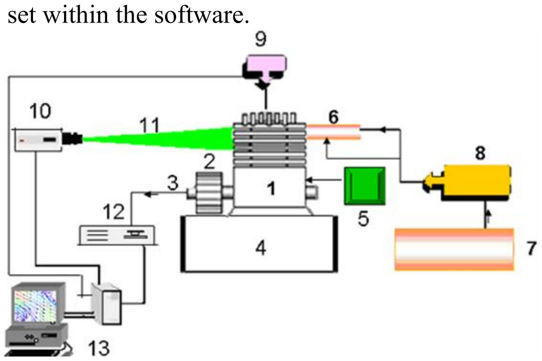
height of 35 mm by adding a transparent plexiglass liner to have an optical access to the combustion chamber. It facilitates a field of view (FOV) of size 87.5 x 35 mm for in-cylinder flow visualization in the combustion chamber. In order to maintain the required compression ratio of 10:1, the piston crown is raised accordingly (Fig.2). The intake manifold of the engine is connected to a plenum to mix and supply the air and seeding particles with uniform mixing.

TABLE I: ENGINE SPECIFICATIONS

| | |
|---|---|
| Bore x stroke (mm) | 87.5 x 110 |
| Compression ratio | 10:1 |
| Rated engine speed (rpm) | 1500 |
| Maximum valve lift (mm) | 7.6 |
| Intake/exhaust port diameter (mm) | 28.5 |
| Intake valve opening (CAD bTDC) | 4.5 |
| Intake valve closing (CAD aBDC) | 35 |
| Exhaust valve opening (CAD bBDC) | 35 |
| Exhaust valve closing (CAD aTDC) | 4.5 |

The PIV system consisting of a double pulsed ND-YAG laser with 200 mJ/pulse energy at 532 nm wave length, a CCD camera of resolution 2048x2048 pixels with a frame rate of 14 per second and a set of laser and camera controllers, and a data acquisition system and a software (Fig.1) is used for the in-cylinder flow measurements. The triggering signals for the laser and camera are generated by a crank angle encoder mounted on the engine crankshaft with a resolution of one CAD. These signals are supplied to the controllers through a signal modulator. A master signal of the crank angle encoder is set to occur at the suction top dead center (TDC) of the engine (considered as a zero CAD). The timing of the triggering signals for the laser and cameras at the required CAD is set within the software.




1. Engine, 2. Motor, 3. Encoder, 4. Test bench, 5. Speed controller, 6. Intake plenum, 7. Air compressor, 8. Seeder, 9. Camera, 10. ND-YAG Laser, 11. Laser sheet, 12. Signal modulator, 13. Data acquisition system

Fig.1. Schematic diagram of experimental setup

A seeding unit is used to generate the fine particles of one micron size with Di-Ethyl-Hexyl-Sebacat ($C_{26}H_{50}O_4$) as a seeding material. The seeding density is controlled accurately by varying the amount of pressurized air supplied to the seeding unit. The laser sheet of 0.5 mm thickness is aligned on a vertical plane passing through the axis of the cylinder. The camera is set to view the laser sheet. In this study, in-cylinder flow measurements are done during suction (30 to 180 CADs) and compression (210 to 330 CADs) strokes in a step of 30 CAD. At every measuring point, 500 image pairs are recorded and stored. The time interval ($\Delta t$) between the two images of a image pair is set (6 µs for suction and 8 µs for compression at this speed) based on the pixel shift (5 pixels), FOV, maximum expected velocity of fluid flow in FOV and the resolution of the camera [9-11]. To minimize the light reflections, a band-pass filter with central wavelength of 532 nm is mounted on the CCD camera. LaVision DAVIS (data acquisition and visualization software) is used for image acquisition and data post-processing. During post-processing, interrogation window size of 32x32 pixels with multi-pass cross-correlation algorithm is used [9-11].

| Pentroof piston with central cavity ($P_c$) | Pentroof piston with offset cavity ($P_o$) |
|---|---|

Fig.2. Pictorial view of the pentroof pistons

## III. RESULTS AND DISCUSSION

The ensemble average velocity vector fields are obtained from 500 instantaneous velocity vector fields. The velocity vector fields are quantified by tumble ratio. The tumble ratio must be evaluated under transient conditions due to the significant effect of the piston shape and its motion unlike the swirl ratio which can be estimated from the steady flow experiments [3]. Figs.4 to 13 shows the ensemble average velocity vectors of in-cylinder tumble flows with superimposed streamline patterns under various engine conditions for the two pistons considered. In these plots, constant length of velocity vector is used with a colour scale to represent their magnitudes. Also, in these plots intake valve and piston positions are shown over the vector fields in order to just show their relative position with CADs.

### A. *In-cylinder Flow during Suction Stroke*

Fig.3 shows the ensemble average velocity vectors with superimposed streamline patterns during suction stroke for a pentroof-offset-cavity piston. At 30 CAD (Fig.3 (a)), it is observe that, at the left and right sides of the intake valve, there exists a small counter clock wise (CCW) and a large clock wise (CW) vortices. At the right side of the valve, air enters the cylinder in the form of a jet. Air jet move towards the right cylinder wall, diverted downwards after striking the wall towards piston top surface in a CW manner. At 60 CAD (Fig.3(b)), the formation of the CCW and CW vortices can be seen which may be occurring due to downward movement of the piston causing the cylinder space to increase. At 90 CAD (Fig.3(c)), the intake valve has opened about 90% (full opening at 110 CAD) and the air jet at the right side of the intake valve splits into two parts. One part, after striking the cylinder wall and piston top surface forms a CW vortex in the left cylinder space. Another part after striking the cylinder wall and piston top surface moves in a CCW manner at the right cylinder space. Also, this type of air flow may be assisted by the intake valve which acts as a bluff body. At 120 and 150 CADs (Fig.3 (d) and (e)), the air flow patterns are almost similar in nature to that of 90 CAD. However, the size of the vortices increase which may due to larger cylinder space above the piston top surface and also due to higher air flow rate. At 180 CAD (Fig.3 (f)), intake valve opening is very less, fresh air entry is also very less, however, the air which is already present during the early part of the cycle is under going the changes in flow pattern. From Fig.3(f), it can be observed that, there is a large CW vortex almost dominating the entire cylinder space with center of it falling just below the intake valve. Also, it can be observed that the formation of a small vortex near the top left corner of the FOV, which may be due to the low pressure region at that point because of large vortex formation at the cylinder center space.

Fig.4 shows that the ensemble average velocity vectors with superimposed streamline pattern during suction stroke at various CADs for a pentroof-central-cavity piston. The in-cylinder tumble flow patterns with this piston at all the CADs look almost similar to those of pentroof-offset-cavity piston at the corresponding CADs by nature. However, the magnitudes of the velocity vectors and also the size of the vortices vary. Therefore, the comparison of the tumble flow patterns can't be done just by the help of velocity vector fields alone. The better way of comparison is discussed later in this paper.

Fig.5 shows the in-cylinder tumble flow patterns for a pentroof-offset-cavity piston during the compression stroke. From Fig.5(a), at 210 CAD, it is observed that a large single CW vortex created at the end of suction stroke is shifting towards left side. It may be due to the squeezing of the entire flow pattern by the ascending piston. Also, it is observed that the air flow from the other parts of the cylinder after striking the cylinder walls forms a CCW vortex near the right cylinder wall. Above vortices are striking each other forming a bifurcation zone at the right cylinder space. At 240 CAD (Fig.5 (b)), due to the vertical movement of piston, entire air flows upward.

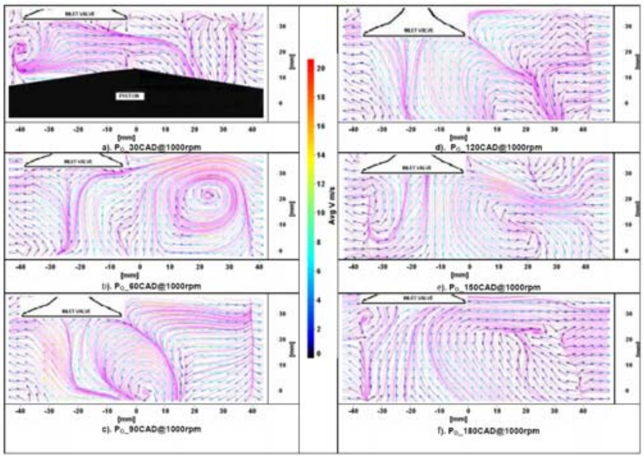


Fig.3. Ensemble average velocity vectors of pentroof with offset cavity piston during suction stroke

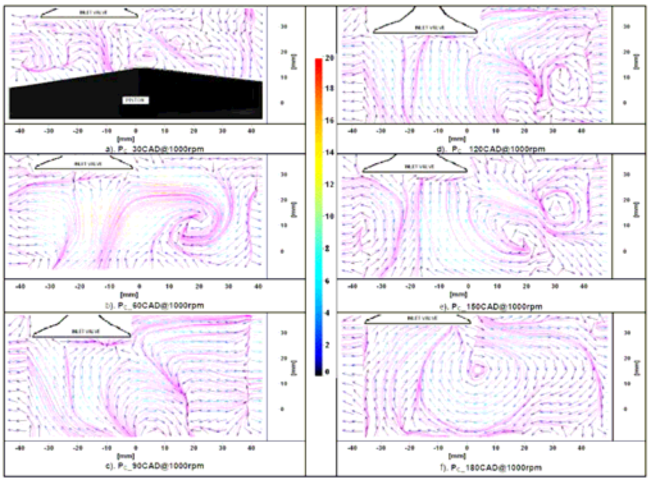


Fig.4. Ensemble average velocity vectors of pentroof with central cavity piston during compression stroke

### B. *In-cylinder Flow motion during Compression*

Air flow pattern at 270 CAD (Fig.5(c)) is similar to that of 240 CAD. At 300 CAD (Fig.5 (d)), the piston has moved further up causing further squeezing of the flow pattern with the formation of a CW vortex. From Fig.5 (e), it is observed that the formation of a CW vortex at 330 CAD. Generally, in the stratified charged and direct injection SI engines, once the ignition starts, the turbulence which depends on air movement inside the combustion space will aid the flame propagation. Therefore, it is very much required that irrespective of the formation of vortex and air flow pattern at the early stages, a favorable air flow pattern must occur at 330 CAD in the above engines.

Fig.6 shows the in-cylinder tumble flow pattern for a pentroof-central-cavity piston during the compression stroke. In this case, the in-cylinder tumble flow patterns at all the CADs are similar to that of pentroof-offset-cavity piston at the corresponding CADs. Fig.6 (e) shows the tumble flow pattern at 330 CAD. In this case also, the tumble flow is in the CW direction as that of pentroof-offset-cavity piston which is very much required.

With the two types of pistons considered here, it is observed that, a large vortex forms during the suction stroke and it changes its direction of rotation throughout the cylinder during suction and compression strokes. It is also observed that generally the vortex center moves towards exhaust valve side during compression.

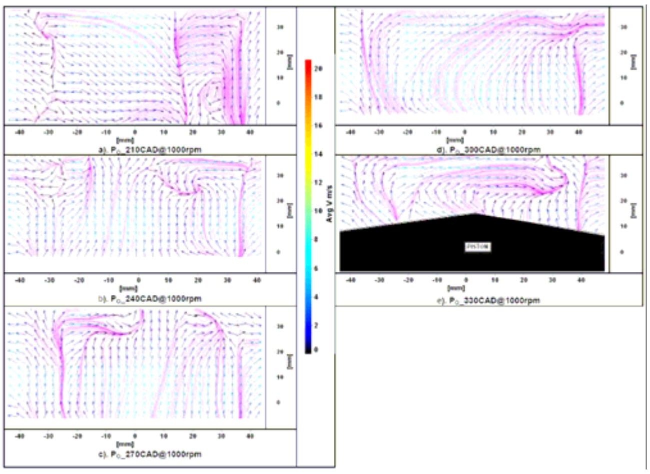


Fig.5. Ensemble average velocity vectors of pentroof with offset cavity piston during suction stroke

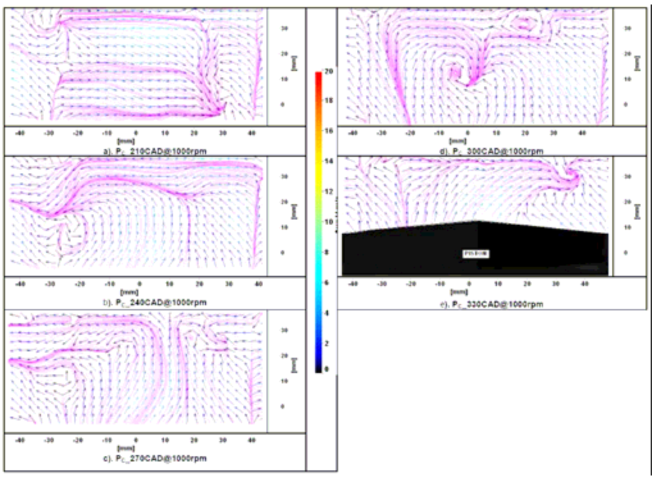


Fig.6. Ensemble average velocity vectors of pentroof with central cavity piston during compression stroke

*C. Variation of Tumble Ratio (TR)*

The quantitative analysis of the tumble motion is evaluated in terms of tumble ratio (TR) by an equation from Reference [12]. Here, the TR is defined as the ratio of the mean angular velocity of the vortices on a target plane to the average angular velocity of the crank. The negative or positive magnitudes of TR indicate the direction of the overall in-cylinder tumble flow on a given target plane as CW or CCW respectively. The temporal variation of TR during suction and compression is shown in Fig.7 at engine speed of 1000 rev/min., for the two pistons considered.

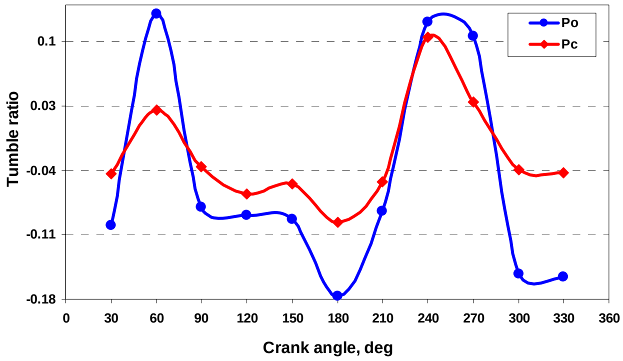



Fig.7. Variation of tumble ratio with crank angle positions

From Fig.7, it is observed that the TR is changing from negative to positive or vice versa indicating the overall air movement changing from CCW to CW directions or vice versa during suction and compression strokes. This may be attributed to: (i) the low pressure and bifurcation zones in the cylinder space (ii) change in piston speed during suction and compression strokes. Even though, in the present study, only the tumble air flow pattern is considered which is likely to happen only in direct injection engines upto compression stroke. However, it almost depicts the mixture flow pattern as well due to very lean mixtures in modern SI engines. In general, a pure tumble flow generates good charge stratification which is crucial for direct injection stratified charge engines [7 and 13]. The TR at 330 CAD for pentroof-offset-cavity piston is close to 0.16, which is 3 times higher compared to pentroof-central-bowl piston with TR of 0.04. Therefore, in order to achieve good charge stratification, it is suggested to use the pentroof-offset-cavity piston.

*D. Variation of Average Turbulent Kinetic Energy (ATKE)*

Generally, stronger tumbling air motion exhibits more turbulent kinetic energy and later it will be dissipated as turbulence during the vortex breakdown. ATKE of the flow indicates its strength as a whole and higher value of it means higher strength of the flow. Fig.8 shows the variation of the ATKE with various CADs during suction and compression strokes for different piston shapes considered. From Fig.8, it is observed that, for the two pistons considered, ATKE is gradually increasing upto 60 CAD (at about 60% opening of the intake valve) and reaches the peak. Then it drops gradually upto 180 CAD becoming almost constant afterwards till the end of the compression stroke. This may be attributed to the fact that, upto 60 CAD, the piston accelerating downwards with increase in inlet valve opening allowing more mass of air to enter into the cylinder space thereby increasing ATKE. However, after 60 CAD, the valve opening continues to increase, but piston starts decelerating; additionally cylinder is already filled with sufficient air causing air flow to reduce resulting in reduced ATKE. It is evident from Fig.8 that ATKE is higher for pentroof-offset-cavity piston (7.6 $m^2/s^2$) and is 1.7 times higher compared to pentroof-central-cavity piston (2.8 $m^2/s^2$). Therefore, from ATKE also, it is evident that pentroof-offset-cavity piston is better compared to pentroof-central-cavity piston as far as in-cylinder tumble flows is concerned.

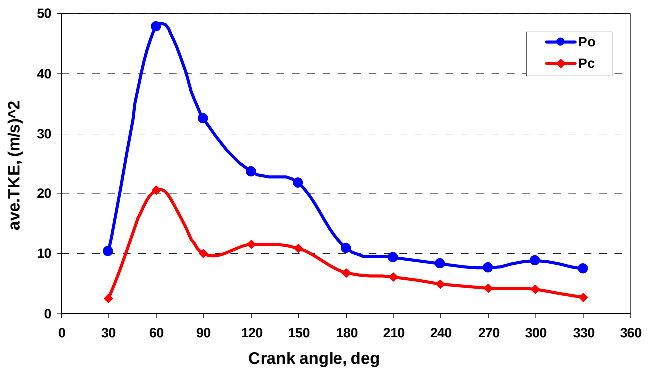



Fig.8. Variation of average TKE with crank angle positions

## IV. Conclusions

From the PIV investigations, the following conclusions are drawn:

The average velocity vectors fields demonstrate that a large tumble vortex formed during suction stroke initially located beneath the intake valve and moves towards the exhaust valve side at end of compression stroke for both the pistons considered.

At 330 CAD, the tumble ratio for pentroof-offset-cavity piston is about 3 times higher compared to pentroof-central-cavity piston.

At 330 CAD, the average turbulent kinetic energy for pentroof-offset-cavity piston is about 1.7 times higher compared to pentroof-central-cavity piston.

This study will be useful for understanding the effect of piston crown shape on the nature of the in-cylinder fluid tumble flow pattern under real engine conditions.

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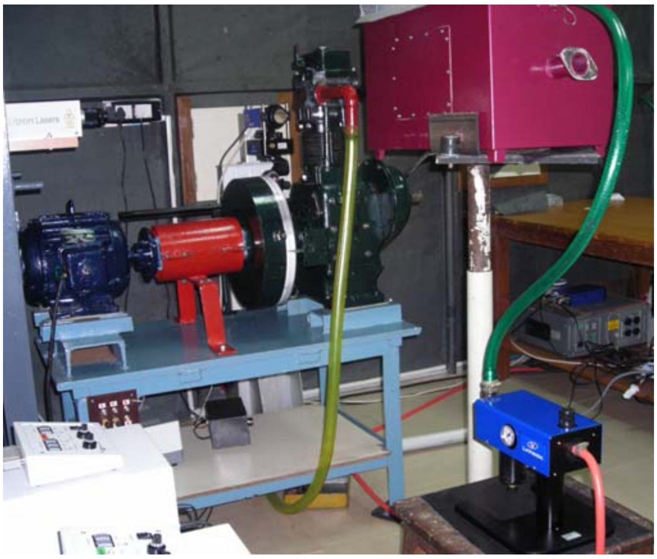
## Appendix

### Photographic view of Experimental setup



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