Estimation of cycle-to-cycle variations for in-cylinder tumbling flows

Yujun Cao\textsuperscript{1,2} *, Lionel Thomas\textsuperscript{2}, Jacques Borée\textsuperscript{2}, Stéphane Guilain\textsuperscript{1}

\textsuperscript{1}: Powertrain engineering and technologies Department, RENAULT s.a.s., 91510 Lardy, France
\textsuperscript{2}: Institut Pprime Department of FTC, CNRS, ENSMA, Université de Poitiers, 86961 Futuroscope Chasseneuil, France
* corresponding author: cao.yujun86@gmail.com

Abstract Two engine configurations having significantly different intake port designs are studied in this work. PIV data in the symmetry plane are compared during intake and compression stroke, showing the effect of intake jet arrangement. Using a RANS decomposition, the most significant differences are observed in the pent roof chamber at the end of the compression stroke where one engine configuration is observed to promote an increase of mean flow kinetic energy before breakdown, probably due to piston work. Moreover, a significantly lower mean flow kinetic energy at TDC for one design of the intake ports can be interpreted as a much more efficient tumble breakdown. In order to extend the classical RANS analysis, we propose to use a resemblance coefficient between each instantaneous velocity field in the pent roof chamber and the mean flow field at 300CAD as a global criterion of similarity of the instantaneous spatial structure and the mean spatial structure before tumble breakdown. 300CAD is chosen because we observe a maximum of kinetic energy of mean flow and a minimum of fluctuating kinetic energy at this CAD. Moreover, the confined large-scale flow is believed to be intrinsically unstable near the end of the compression. Therefore, ignition occurs during this transition from an organized flow state to small scale turbulence and projection on the flow before breakdown seems relevant to obtain a “progress variable”, distinguishing different engine cycles. A triple decomposition of the velocity field is deduced and the corresponding statistics are analyzed.

1. Introduction

The way in which coherent motions induced in the cylinder of gasoline engine during the intake stroke evolve during the compression stroke has a great influence on the flame propagation after ignition (Heywood 1988; Lumley 1999). Tumble, the rotating flow which axis is perpendicular to the cylinder axis, is widely used in most of gasoline engines.

Tumble is interesting for a number of fluid-mechanical reasons. This large-scale coherent motion is introduced inside the combustion chamber during the intake stroke. The compression transfers energy to the tumbling motion, strengthens it during the compression stroke. At the end, tumbling motion becomes unstable as the piston rises toward top dead center (TDC) (Hill and Zhang 1994; Lumley 1999; Borée et al. 2002). Thus its kinetic energy is transferred to turbulence before the flame ignition: that is the key point for high speed combustion and good efficiency of the engine. A key challenge with present engine developments is the modeling, understanding and control of cycle-to-cycle variations (CCV). This cyclic variability is understood here as a large-scale cycle to cycle variation that ultimately controls the gas motion and turbulence in the vicinity of the spark plug gap and at the time of ignition.

The need to control this process has been reinforced, in the recent years, due to the application of the downsizing concept. Indeed, it is one of the most applied concepts to achieve high performances and low fuel consumption: by reducing engine displacement and, as a consequence, by raising the engine loads, the need of turbulence is also increased to avoid knock issue and non-efficient combustion (late ignition). Furthermore, at part load, the maximization of fuel economy, thanks to IGR (Internal Gas Residual) promotion for instance, requires to approach the stability limit of combustion. Controlling CCV is also extremely important as the high CCV levels can lead to high
pollutants formation and serious drivability issues (Heywood et al. 1988).

A literature survey of (Ozdor et al. 1994) indicates a number of factors influencing CCV, the main factors related to the aerodynamic are following: (i) Spark plug orientation with respect to mean flow velocity vector; (ii) Mean flow vector at the spark gap vicinity; (iii) Overall in-cylinder flow pattern; (iv) Turbulence scales in the spark gap vicinity; (v) Turbulence intensity. After sparking and flame ignition, these factors influence mainly the second and third combustion stage: the initial flame kernel and turbulent flame propagation.

Numerical modeling is one efficient way to study engine flow fields. Classical Reynolds Averaged Navier-Stokes (RANS) approach, a usual tool to most automobile industries, solves the modeled phase-averaged Navier-Stokes equation. So mean flow and turbulence levels can be estimated. Only Large Eddy Simulation (LES) approach gives CCV information (Enaux et al. 2011, Granet et al. 2012): this numerical method is still under development for engine application. On the experimental side, PIV measurement, pioneered by (Reuss et al. 1989), is mature technique for engine research.

A classic 2D-2C PIV was used to study the formation of engine flow by (Voisine et al. 2011) and they analyzed high-speed PIV data when focusing on the breakdown phase. A phase invariant POD enables these authors to perform triple decomposition and to distinguish cycles according to their structure near TDC: to distinguish a contribution of the in-cycle coherence from the turbulence carried by these flow states.

Our goal in this work is to compare two engine configurations with different tumble ratios due to different intake port designs (Table 1). We first analyze their main difference during intake and compression stroke in section 3 and 4. In order to extend the classical RANS analysis, we then propose to use a resemblance coefficient as a global criterion of similarity of the instantaneous spatial structure and the mean spatial structure before tumble breakdown. A “progress variable”, distinguishing different engine cycles is obtained. A triple decomposition of the velocity field is deduced and the corresponding statistics are analyzed.

2. Experimental Set-up

Two engine configurations are used in this work. Both configurations are based on a spark ignition engine with a pent-roof chamber and four valves per cylinder. The major differences are the engine displacement and the intake port design. The other differences of their characteristics are summarized in Table 1. The intake port angle is the angle between the intake port and horizontal axis (x-axis from Fig. 1b). The experimental tumble ratio is obtained by using a steady flow test bench. This standard measuring method is based on the determination of the angular momentum using a flow straightener suits. If tumble ratio is considered as 1 for first engine configuration, then it’s 2.5 for the new engine configuration of case 2.

<table>
<thead>
<tr>
<th>Engine Configuration Case</th>
<th>1</th>
<th>2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore (b)</td>
<td>78</td>
<td>72.2</td>
</tr>
<tr>
<td>Stroke</td>
<td>73.1</td>
<td>73.1</td>
</tr>
<tr>
<td>Maximum pressure at TDC [bar]</td>
<td>18</td>
<td>20</td>
</tr>
<tr>
<td>Pent-roof angle [degree]</td>
<td>28°</td>
<td>28°</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>11</td>
<td>10</td>
</tr>
<tr>
<td>Intake port angle [degree]</td>
<td>≈ 40°</td>
<td>≈ 30°</td>
</tr>
<tr>
<td>Intake valve diameter [mm]</td>
<td>28</td>
<td>27</td>
</tr>
<tr>
<td>Experimental Tumble Ratio</td>
<td>1</td>
<td>2.5</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Case</th>
<th>1</th>
<th>2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Symmetry Plane in pent-roof chamber</td>
<td>Height</td>
<td>11</td>
</tr>
<tr>
<td></td>
<td>Length</td>
<td>78</td>
</tr>
<tr>
<td>Visualized domain</td>
<td>Height</td>
<td>7.5</td>
</tr>
<tr>
<td></td>
<td>Length</td>
<td>39</td>
</tr>
</tbody>
</table>

Table 1: a. characteristics of two engine configurations; b. pent-roof visualization dimensions
The cylinder is fully transparent. A flat window in the pent-roof chamber is present to allow visualizations at the end of the compression and near the position of the spark plug gap. The dimensions of the windows in two cases of study are shown in Table 1. A flat crown piston is used in both cases instead of a real shape piston to avoid strong optical distortions. Indeed, the laser sheet is driven through the piston for the pent-roof measurements and a flat piston guarantees a precise position of the laser sheet.

Both engines run at 1200 rpm with atmospheric conditions. The PIV set-up of (Voisine et al. 2011) is not detailed in here. The database of case 2 is obtained using a standard PIV set-up from LAVISION. The frequency of camera is limited to 5 Hz. PIV system consists in a double pulsed Nd:Yag (50 mJ per pulse). The thickness of the light sheet is 1 mm. The laser frequency can be fixed depending on engine rotation speed. Images are recorded by a 1280x1024 CCD camera. Adaptive cross-correlations are computed on 64x64 pixels-size final interrogation windows with 50% overlap. Silicone is used to seed the flow (Dow Corning® 510 Fluid 50 Cst). Thus we obtain one instantaneous velocity field every two engine cycles for about a large number of realizations at fixed crank angle degree (CAD). Two types of database are presented in the vertical symmetry plane (see Fig. 1): one database is obtained in the cylinder part and the other in the pent-roof chamber. As one engine cycle is from 0 CAD to 720 CAD and the TDC is 360 CAD, Table 2 shows all the realizations analyzed in following sections, the measurements in the cylinder symmetry plane concern 90 CAD, 120 CAD, 180 CAD, 220 CAD and 270 CAD; the pent-roof measurements are 10 phases from 270 CAD to 360 CAD, every 10 CAD. The number of realizations is about 500 to guarantee the statistical convergence of first and second order moments.

<table>
<thead>
<tr>
<th>Phases in engine cycle [CAD]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cylinder Symmetry Plane</td>
</tr>
<tr>
<td>Pent-roof Symmetry Plane</td>
</tr>
</tbody>
</table>

**Table 2: all realizations for both engine configurations**

The flow will be described using a Cartesian coordinate system \((x, y, z)\): \(z\) is the vertical axis and \(x\) is the axis between the two intake ports. The origin is chosen in the middle of bore and on the joined plane between cylinder part and combustion chamber.

- 3 -
transfer them into turbulence during breakdown.

Tumble ratio characterizes the integral angular momentum level inside the cylinder. The tumble ratio in a two-dimensional domain $\Omega$ is defined by:

$$R_t = \frac{\int (W(x-x_c)-U(z-z_c))d\Omega}{2\pi R \int \left((x-x_c)^2 +(z-z_c)^2\right)d\Omega}$$

Eq 1

Where $U$ and $W$ correspond to horizontal and vertical velocity components, respectively, and $x-x_c$ and $z-z_c$ to the horizontal and vertical distances of the mesh point from the geometric center $(x_c; z_c)$ of domain $\Omega$. $R$ is the engine rotation speed. The calculation is done for each instantaneous velocity field and for the mean flow as well, noted respectively $R_{t_i}$ and $R_{t_m}$. Fig. 2a gives the values of average level for instantaneous tumble ratio, $<R_{t_i}>$ and mean flow’s tumble ratio, $R_{t_m}$ at each available phases of cycle. It shows that the temporal evolutions of the tumble ratio are very similar for both configurations. $R_{t_m2}$, a value deduced from PIV in the symmetry plane is however only 20% higher than $R_{t_m1}$. This is a striking observation because measurements obtained on a steady test bench provide a much larger difference (a 2.5 factor due to intake port arrangement). More analysis is presently devoted to this point. In both situations, we have found that the rms of instantaneous tumble ratio is about 10% of the mean value.

![Fig. 2: a. Average level evolution of tumble ratio: $R_{t_m}$: tumble ratio of the mean flow; $<R_{t_i}>$: average of tumble ratio for each instantaneous field; b. Total Energy evolution: “$K_m$”: kinetic energy of the mean flow; “$K_f$”: fluctuating kinetic energy.](image-url)

We consider $U(X, \theta, n) = U_n(\theta) = (U_n(\theta), W_n(\theta))$ the set of realizations of the velocity field in the two-dimensional domain $\Omega$, in any point $X = (x, z) \in \Omega$, at phase $\theta \in [0;720]^{CAD}$ and in cycle $n \in [1;N]$. $U_n$ and $W_n$ are two velocity components along horizontal axis and vertical axis (see Fig. 1). For the following equations, we skip $(X, \theta)$ for the sake of brevity. The Reynolds decomposition is defined with phase average of velocity field $\langle U \rangle = \frac{1}{N} \sum_{n=1}^{N} U_n$.

$$U_n = \langle U \rangle + u'_n$$

Eq 2

The mean kinetic energy of all realizations ($\langle k \rangle$), the kinetic energy of the mean flow ($k_m$) and the fluctuating kinetic energy ($k_f$) are defined as:
\[ \langle k \rangle = k_m^1 + k_f^1 \quad \text{with} \quad \begin{align*}
    k_m^1 &= \frac{1}{2} \langle (U)^2 \rangle = \frac{1}{2} \left( \langle (u)^2 \rangle + \langle (w)^2 \rangle \right) \\
    k_f^1 &= \frac{1}{2} \langle (u_n^2) \rangle = \frac{1}{2} \left( \langle (u_n^2) \rangle + \langle (w_n^2) \rangle \right)
\end{align*} \]  
\text{Eq 3}

We only take into account components in the symmetry plane.

For comparison we normalize the energies in the domain \( \Omega \) as \( \langle K \rangle = \frac{1}{\Omega} \int \langle k \rangle d\Omega \), and it leads to:

\[ \langle K \rangle = K_m^1 + K_f^1 \]  
\text{Eq 4}

These quantities are important indicators of mean flow structure’s development.

Fig. 2b shows evolutions of normalized kinetic energy \( K_m^1 \) and \( K_f^1 \). \( K_m^1 \) decreases in both cases, not only because kinetic energy of the mean flow is transferred to the fluctuating field but also because the spatial distribution of mean momentum in the cylinder can significantly vary (\( K_m^1 \) is only evaluated from the symmetry plane data). For the two studied cases \( K_m^1 \) and \( K_m^2 \) differ very significantly at 90 CAD with \( K_m^2 \approx 3 * K_m^1 \). Looking at the mean velocity field of Fig. 3, we see immediately that the trace of intake valve jets are very distinct for the two engine set-up. But later on, \( K_m^1 \) and \( K_m^2 \) evolve to similar values at BDC and mid-compression.

A detailed comparison of mean flow field structures, including data in transverse or near piston planes, is not the scope of this paper. Fig. 3 however shows that the differences induced by both intake ports are very clear at BDC. Indeed, the roll up of the intake valve jet seems to be far more complete at BDC for case 2. Mastering valve jet momentum flux distribution and direction in the valve gap is a critical issue that impacts their interaction in the cylinder and the impact of the resulting jet on the piston. Tumbling flow gets organized in each individual cycle after valve closure. Both mean flows are relatively similar at 270 CAD (Fig. 3).

Fig. 3: comparison of mean flow for case 1 and 2 in cylinder symmetry plane: black arrows represent velocity vectors and color map in background represents velocity magnitude level, [0; 15] m/s.
At the end of compression phase, the goal of engine designers is that most kinetic energy is transferred from the large-scale tumbling motion to small-scale turbulence. The pent-roof chamber is chosen for the PIV measurements over the cylinder part because we want to study the engine flow near the spark plug gap at TDC. The measurements are from 270 CAD to 360 CAD, every 10 CAD. As the two databases are acquired in two slightly different domains (see Fig. 1b), we choose the same domain for comparison, of 0.13b (b stands for bore) in height and 0.6b in length. The mean velocity fields measured in the pent-roof at 300 CAD and 360 CAD are compared in Fig. 4. The x and z axes are normalized with the bores of each engine configurations: $x \in [-0.3;+0.3]b$, $z \in [0.1;0.23]b$. The footprint of a mean tumbling motion is very clear on these graphs but the difference of mean velocity magnitude is very large. This observation appears more clearly when considering the CAD evolution of $K_m$ in the pentroof symmetry plane from 270CAD to 360 CAD (Fig. 5). We see very clearly that $K_m1$ displays no increase during compression. This means that the kinetic energy brought by compression is transferred to turbulence. On the contrary, $K_m2$ increases to a maximum at about 300 CAD, just before tumble breakdown. For a classical engine design, it is important to remark that the magnitude of the divergence of the mean flow field $\nabla \cdot \langle \mathbf{U} \rangle = -\dot{\rho}/\rho$, where $\dot{\rho}$ is the rate of variation of the density, typically peaks near 30 CAD before TDC. Therefore, we conjecture that a more organized tumbling motion, tailored using adapted intake ports, can benefit from piston work in order to increase the mean flow kinetic energy during compression and before breakdown. A 30% increase is also measured by (Müller et al. 2011). In their case, however, the mean flow kinetic energy peaks at approximately 65° before TDC. To our understanding, such sensitivity of the evolution of mean kinetic energy to the geometry is not unexpected. Indeed, the real mean flow structure is complex and is geometry dependent. In each geometry, secondary separations along the cylinder walls, for example upstream corners, lead to shear flow regions and significant transfer of mean kinetic energy to turbulence. The decrease of $K_m$ is quasi exponential beyond 300 CAD. An increase of the fluctuating kinetic energy is clearly associated with the tumble breakdown. A maximum of $K_f$ is approximately reached at 340CAD in both configurations while dissipation of $K_f$ into heat dominates at the end of the compression. One striking difference between both configurations is that $K_m2$ is much smaller that $K_m1$ at TDC. This can be interpreted as a much more efficient tumble breakdown in situation 2. The ratio $K_f/K_m$ (Fig. 5b) emphasizes the differences between both engine designs near TDC. Indeed, this “turbulence intensity” $K_f/K_m$ grows quasi exponentially during tumble breakdown and this ratio is a factor 1.5 larger for the second port configuration at TDC. One must however be careful when analyzing such results because the fluctuating field includes both in-cycle coherence structure and turbulence contributions. A more advanced decomposition is proposed in next section.

![Fig. 4: comparison of mean flow at 300 CAD and 360 CAD of both cases in pent-roof symmetry plane: x-axis range is [-0.35; 0.35]b, and z-axis range is [0.05; 0.25]b; black arrows represent velocity vectors; color map in background represents velocity magnitude level: [0;10] m/s for 330 CAD and [0;3] m/s for 360 CAD](image)
4. Triple decomposition of the flows fields during tumble breakdown

A statistical analysis was proposed by Voisine et al. 2011 using phase invariant POD based on High-Speed PIV (HSPIV) data. It consists in using statistics of the squares of first random coefficients at TDC to separate, among all realizations, cycles keeping their coherence from cycle experiencing a partial or a full tumble breakdown. We propose here a similar statistical analysis using standard PIV measurements. Indeed, HS-PIV is more difficult to tune because of seeding issues and is more expensive to perform on a daily basis, which also reduces the accuracy of statistics due to the limited number of cycles that can be acquired. In this context we first define the inner product $(U, V)$ and the norm $\|U\| = (U, U)^{1/2}$ by:

$$
(U, V) = \int_{\Omega} UV dX
$$

We define the resemblance coefficient (Hao et al. 2010) in same domain $\Omega$ between any two realizations $(U_i, U_j)$ where $U_i = U(X, \theta, n)$ as following:

$$
\forall X \in \Omega, R^{ij}(X) = \frac{(U_i, U_j)}{\sqrt{(U_i, U_i)(U_j, U_j)}} = \frac{(U_i, U_j)}{\|U_i\|\|U_j\|}
$$

Using 2D-2C PIV fields in the pentroof, our goal is to separate different types of flow state near TDC. We propose to use the resemblance coefficient (spatial correlation coefficient) between each instantaneous velocity field and the mean flow at 300CAD as a global criterion of similarity of the instantaneous spatial structure and the mean spatial structure before tumble breakdown. 300CAD was chosen because we observe a maximum of kinetic energy of mean flow and a minimum of fluctuating kinetic energy at this CAD. Moreover, the confined large-scale flow is believed to be intrinsically unstable near the end of the compression. Therefore, ignition occurs during this transition from an organized flow state to small scale turbulence and projection on the flow before breakdown seems relevant to obtain a “progress variable”, distinguishing different engine cycles.

Noting that the first POD phase invariant mode in Voisine et al (2011) is very similar to a mean flow prior to tumble breakdown and that the spatial correlation coefficient between the mean flow at 300 CAD and the mode 1 of phase invariant POD is 99.54%, there are a lot of similarities between the analysis presented here and the phase invariant POD strategy. One important difference is however the loss of cycle resolved information.

Eq 6 can be rewritten:

$$
R_n(\theta) = \frac{U_n(\theta)_{ref} \cdot \langle U \rangle_{ref} \cdot \langle U \rangle_{ref}}{\sqrt{(U_n, U_n)(\langle U \rangle_{ref}, \langle U \rangle_{ref})}}
$$

Fig. 5: a. Evolution of kinetic energy in mean flow ($K_m$) and fluctuating kinetic energy ($K_f$); b. $K_f/K_m$ ratio
$R_n(\theta)$ is the spatial correlation between the reference field $\langle U \rangle_{ref}$ and the field $U_n(\theta)$. At each phase $\theta$, about 500 projection coefficients $R_n(\theta)$ are then obtained in the pent-roof symmetry plane and statistics of $R_n^2(\theta)$ are presented in Fig. 6 and 7. $R_n^2(\theta)$ is chosen here as the progress variable of tumble breakdown because, in relation with a POD methodology, the average of $R_n^2(\theta)$ provides the “energy” of the projection on the mean flow structure at 300CAD. More precisely, Figure 6 and 7 are obtained by splitting the [0;1] interval in 5 groups of width 0.2 and by building the corresponding histograms. The group 1, representing [0;0.2] interval, is poorly related to reference mean structure at 300 CAD while group 5, representing ]0.8;1] interval, has higher correlation. A very clear destruction of in-cylinder flow is observed as the populations of higher $R^2$ groups decrease and those of lower $R^2$ groups increase as the CAD increases.

**Case 1**

**Case 2**

![Fig. 6: histogram of decomposition in 5 groups from 270 CAD to 360 CAD in pent-roof symmetry plane in both cases](image)

![Fig. 7: Comparison of histogram in 2 cases for each phase from 270 CAD to 360 CAD: abscissa is the group number and ordinate is population from [0; 1].](image)

Mean velocity fields in groups 1 and 5 at TDC are compared for cases 2 in Fig. 8. We see very clearly a difference in the mean velocity magnitude for these two extreme classes.
Thanks to this classification, a triple decomposition can be proposed and the phase-averaged Reynolds stresses can be decomposed into a contribution of the in-cycle coherence, noted “coh”, and the turbulence carried by the flow states, noted “turb”. The reasoning to define the triple decomposition is the following: for each instantaneous velocity field, a criteria $C$ ($C$ is here the value of the progress variable $R$) is defined using integral information (projection on the mean flow field prior breakdown). The value of $C$ can then be used to perform a conditional analysis of the set of velocity fields. The triple decomposition reads:

$$U_n = \langle U \rangle + u'_n \text{ with } u'_n = u'_{coh} + u''_n \text{ and } \begin{cases} u'_{coh} = \langle U|C = \beta \rangle - \langle U \rangle \\ u''_n = U_n - \langle U|C = \beta \rangle \end{cases}$$  Eq 8

The fluctuating kinetic energy is splitted in two terms with:

$$k_f = k_{turb} + k_{coh}$$  Eq 9

For a statistical analysis, $U$ and $C$ are considered as random variables at phase $\theta$. The probability density function (pdf) is $f(\alpha, \beta)$ for $\alpha \in [-\infty, +\infty]$ and $\beta \in [-1;1]$ and $f_C(\beta) \cdot d\beta$ is the probability to find $C \in [\beta, \beta + d\beta]$ for all $U$ where $f_C(\beta) = \int_{-\infty}^{\infty} f(\alpha, \beta) \cdot d\alpha$.

The probability to find the couple $(U, C) \in [\alpha + \Delta\alpha] \times [\beta + \Delta\beta]$ is:

$$f(\alpha, \beta) \cdot d\alpha \cdot d\beta = f_{U|C}(\alpha|\beta) \cdot d\alpha \cdot f_C(\beta) \cdot d\beta$$  Eq 10

The fluctuating kinetic expression in Eq 3 can be rewritten:

$$2k_f = \langle u_n'^2 \rangle = \int_{\alpha, \beta} \left( \alpha - \langle \alpha \rangle \right)^2 f(\alpha, \beta) \cdot d\alpha \cdot d\beta$$  Eq 11

Finally the decomposition of $k_f$ is (we skip the demonstration details to keep brief):

$$k_f = k_{turb} + k_{coh} \text{ where } \begin{cases} k_{turb} = \int_{\beta} \left( \langle u''_n \rangle^2 \right) \cdot f_C(\beta) \cdot d\beta \\ k_{coh} = \int_{\beta} \left( \langle U \rangle - \langle U|C = \beta \rangle \right)^2 f_C(\beta) \cdot d\beta \end{cases}$$  Eq 12

Integral values can also be calculated (see section 3) with $K_f = K_{turb} + K_{coh}$. The ratio $K_{coh}/K_f$ is drawn in Fig. 9 from 270CAD to 360CAD. The relative contribution of $K_{coh}$ is a minimum at 300CAD because the mean velocity field at 300CAD is used for the projection. It is interesting to note that the relative contribution peaks approximately from 330CAD to 340CAD, during tumble breakdown, and decrease significantly at TDC. The tumble breakdown, occurring at the time scale of the large scale rotating flow, is therefore a source of flow structure fluctuations that are likely to interact significantly with the ignition process. At these phases, note that 90% of $K_{coh}$ is due to
fluctuations in \( u \) (longitudinal direction along the roof of cylinder) while both longitudinal and vertical velocity fluctuations contribute to \( K_{\text{turb}} \) at respectively 60 and 40% approximately. Even if our measurement region has a limited size, this result shows that it is the transition of the tumbling flow that results in the coherent fluctuations.

Considering again the RANS analysis presented in section 3, we may conjecture that a more relevant indicator of the turbulence intensity is \( K_{\text{turb}}/(K_m + K_{\text{coh}}) \), instead of \( K_f/K_m = (K_{\text{coh}} + K_{\text{turb}})/K_m \) plotted in Fig. 5b. Cycle to cycle variations of large scale coherence and smaller scale turbulence have indeed different roles at and after ignition. These ratios are plotted in Fig. 8b and compared for both engine configurations. One striking and unexpected result is that the turbulence to large scale kinetic energy ratio is the same for both engine configurations at TDC. This result is presently under study.

5. Conclusion and perspectives

Two engine configurations having significantly different intake port designs were studied in this work. PIV data in the symmetry plane were compared during intake and compression stroke, showing the effect of intake jet arrangement. Using a RANS analysis, the most significant differences are observed in the pent roof chamber at the end of the compression stroke where one engine configuration is observed to promote an increase of mean flow kinetic energy before breakdown, probably due to piston work. Moreover, a significantly lower mean flow kinetic energy at TDC for one design of the intake ports can be interpreted as a much more efficient tumble breakdown.

In order to extend the classical RANS analysis, we have proposed to use a resemblance coefficient between each instantaneous velocity field in the pent roof chamber and the mean flow field at 300CAD as a global criterion of similarity of the instantaneous spatial structure and the mean spatial structure before tumble breakdown. 300CAD was chosen because we observe a maximum of kinetic energy of mean flow and a minimum of fluctuating kinetic energy at this CAD. Moreover, the confined large-scale flow is believed to be intrinsically unstable near the end of the compression. Therefore, ignition occurs during this transition from an organized flow state to small scale turbulence and projection on the flow before breakdown seems relevant to obtain a “progress variable”, distinguishing different engine cycles. A triple decomposition of the velocity field was
deduced and the main result is that the relative contribution of large scale coherence to the fluctuating kinetic energy peaks during tumble breakdown, 90% of this contribution being due to longitudinal velocity fluctuations. Such decomposition is believed to be useful to propose refined analyzing strategy for engine flow field near TDC and is presently used to further analyze different engine configurations.

Acknowledgements

The PhD grant of Y. CAO is financed by Renault. The technical support of F. Texier and P. Braud is warmly acknowledged.

6. References


Voisine M, Thomas L, Borée J and Rey P (2011) Spatio-temporal structure and cycle to cycle variations of an in-cylinder tumbling flow. Exp. in Fluids 50 n°5: 1393-1407