ESTIMATION OF TURBULENCE LEVEL IN SPARK IGNITION ENGINE USING EXPERIMENTAL DATA OF IN-CYLINDER PRESSURE

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ABSTRACT – The objective of this study was to analyze the applicability of the proposed procedure for the estimation of in-cylinder turbulence level in SI engine from measured pressure data and to use it for the calibration of cycle simulation model constants. In the Laboratory of IC Engines and Vehicles at the Faculty of Mechanical Engineering and Naval Architecture in Zagreb, Croatia, the experimental setup of single cylinder IC engine was built and experimental results were used for the analysis in this study. An offline in-house code for the calculation of heat release rate from the measured in-cylinder pressure profiles was applied over different engine speeds at full load conditions. By means of the additionally developed code the estimation of turbulence intensity was made. The estimated values of turbulence intensity have been used for the calibration of 0-D turbulence model. The estimated levels of in-cylinder turbulence over analyzed operating conditions were compared with levels of in-cylinder turbulence intensity at top dead center calculated by means of the mean piston speed. The constants of cycle simulation with the quasi-dimensional combustion sub-model were calibrated and the simulation results of turbulence and combustion are compared with the reference and measured data. The average difference between the estimated and simulated turbulence intensity was below 15%. The prediction of in-cylinder turbulence was performed on the cycle-resolved basis for one operating point, resulting with the information about cycle-to-cycle variations of in-cylinder turbulence during the combustion. The prediction of in-cylinder turbulence based on the measured data represents efficient inverse method whose results can be used for the faster calibration of 0-D turbulence models without performing time-consuming 3-D CFD simulations.

INTRODUCTION

The development of cycle simulation models for internal combustion (IC) engines became continuous process during the last few decades. The most popular combustion models within the cycle simulations are quasi-dimensional combustion models that take into account the combustion chamber geometry and turbulence level on the combustion rate. Therefore, such combustion models integrate additional sub-models for the turbulence modeling, combustion chamber geometry description, etc. In order to calibrate particular sub-model parameters, the reference data are required. Experimental measurement of turbulence intensity in the combustion volume of IC engine is difficult due to extreme in-cylinder conditions. Hot-wire anemometers can be used for turbulence intensity measurement, but in this method it is required that the engine operates only under motored conditions [1]. Laser-based methods (e.g. Laser Doppler Anemometry - LDA) can provide both spatial and temporal data on the turbulence intensity under firing conditions, but extensive engine modifications are required [2]. On the other hand, the validation of cycle simulations can be performed by the comparisons of simulation results with 3-D CFD data for the same operating
conditions [3], but it requires additional computational power and time compared to cycle simulations. In the study presented in [4] the model for turbulence intensity prediction was developed based on the governing equations of quasi-dimensional turbulent entrainment model. The calculated turbulence intensity from 5% to 10% mass fraction burned was averaged to define a single value that represents each operating point and the difference between the cycle simulation values for the early stages of combustion and predicted values were within 10% [4].

Within the study presented in this paper the model for the estimation of in-cylinder turbulence was proposed based on the new quasi-dimensional combustion model – flame tracking model (FTM) [5]. In order to define the single value of turbulence intensity that defines each operating point the averaging of turbulence intensity from 10% to 90% of mass fraction burned when developed turbulent flame occurs was defined. The estimated turbulence intensity was used for the calibration of cycle simulation model constants related to the turbulence sub-model. It was demonstrated that the estimation model can be very useful engineering tool for the better calibration of cycle simulation model when 3-D CFD data of analyzed engine are not available.

EXPERIMENTAL SETUP

The new experimental setup of single cylinder IC engine was made in the Laboratory of IC Engines and Vehicles (Faculty of Mechanical Engineering and Naval Architecture in Zagreb). The original compression ignition engine was slightly modified in order to work in a spark ignition operation. The modifications of experimental engine include the mounting of spark plug, machining of the piston top (decrease of compression ratio) and the new design of intake system with the connection of fuel injector. The details of experimental engine and operating parameters are given in Table 1.

Table 1: Experimental engine data and operating conditions.

<table>
<thead>
<tr>
<th>Manufacturer/Engine Name</th>
<th>Hatz 1D81</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
<td>100 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>85 mm</td>
</tr>
<tr>
<td>Connecting Rod Length</td>
<td>127 mm</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>11.95</td>
</tr>
<tr>
<td>Intake Valve Opens / Closes</td>
<td>36°CA BTDC / 60°CA ABDC</td>
</tr>
<tr>
<td>Exhaust Valve Opens / Closes</td>
<td>54°CA BBDC / 21°CA ATDC</td>
</tr>
<tr>
<td>Engine Speed</td>
<td>1000, 1500, 2000, 2500, 3000 rpm</td>
</tr>
<tr>
<td>Air excess ratio</td>
<td>1.00</td>
</tr>
<tr>
<td>Load</td>
<td>WOT – wide open throttle</td>
</tr>
<tr>
<td>Fuel Type</td>
<td>Commercial gasoline (RON95)</td>
</tr>
</tbody>
</table>

The measurement of intake pressure was performed with the AVL LP11DA sensor that is located at the intake pipe 60 mm upstream the intake valve, while the instantaneous in-cylinder pressure was recorded via AVL GH14DK pressure sensor with the recording resolution of 0.1°C. The excess air ratio (lambda value) was controlled by the wideband sensor in the exhaust line. The operating points used in this study were recorded at engine speeds 1000 rpm to 3000 rpm with wide open throttle and without the recirculation of exhaust gases. Pressure traces of 300 cycles have been stored enabling the appropriate statistical analysis of cyclic combustion variability. The offline application for the pressure data filtering (Savitzky-Golay filter) and calculation of combustion rate was applied so that the experimentally recalculated data can be further used in the estimation model of
turbulence intensity and for the validation of combustion model.

BACKGROUND OF COMBUSTION MODEL

In order to estimate the level of in-cylinder turbulence from the experimentally defined rate of heat release, the governing equations of quasi-dimensional model have to be employed. The basic equations of newly developed Flame Tracking Model - FTM [5] were applied within the developed code for the estimation of turbulence level. Before the prediction of turbulence intensity in the flame front, the turbulent flame speed has to be calculated from the instantaneous rate of heat release $dQ/d\alpha \ (\text{J/deg})$ that is previously calculated by the offline in-house code already used in previous studies [6,7]. The instantaneous turbulent flame speed $u_{t,f} \ (\text{m/s})$ can be expressed from the mass burning rate equation assuming the propagation of smooth and thin flame front across the combustion chamber:

$$\frac{dm}{d\alpha} = \frac{dQ}{d\alpha} \cdot \frac{1}{H_s} = \left( \rho_{\text{UZ}} \cdot A_f \cdot u_{t,f} \right) \cdot \frac{1}{\omega},$$

(1)

where $H_s \ (\text{J/kg})$ is the mixture lower heating value, $\rho_{\text{UZ}} \ (\text{kg/m}^3)$ is density of unburned zone, $A_f \ (\text{m}^2)$ is the flame surface area and $\omega \ (\text{deg/s})$ is angular engine speed. The unburned zone density is correlated to the total density $\rho_{\text{tot}} \ (\text{kg/m}^3)$ and mass fraction burned $x_B \ (-)$ according to the following expression:

$$\rho_{\text{UZ}} = \rho_{\text{tot}} + C_p \cdot x_B \cdot \rho_{\text{tot}} = \frac{m_{\text{tot}}}{V_c} \cdot \left( 1 + C_p \cdot x_B \right),$$

(2)

where $C_p \ (-)$ is calibration constant set to 2.0, $m_{\text{tot}} \ (\text{kg})$ is total in-cylinder mass, and $V_c \ (\text{m}^3)$ is the instantaneous volume of cylinder. The total in-cylinder mass is calculated assuming the combustion efficiency of 95% and the residual gas content of 5%. Once the unburned density is calculated, the burned volume can be calculated as a difference between the total and unburned zone volume. With the newly developed sub-model for the description of combustion volume described in [5], the free flame surface area $A_f$ and burned (entrained) volume $V_B$ are defined as a function of flame radius and relative piston position. When the instantaneous free flame surface area is defined, the turbulent flame speed can be calculated from equation (1). On the other hand, the turbulent flame speed $u_{t,f}$ is defined with the correlation presented in [8] and used in the developed FTM [5]:

$$u_{t,f} = u_N \cdot \max \left[ 1.0, \left( -4.37 + 1.13 \left( \frac{u'}{u_N} \right)^{1/2} \cdot \left( \frac{u_N \cdot L_1}{\nu} \right)^{1/4} \right) \right],$$

(3)

where $u_N \ (\text{m/s})$ is laminar flame speed, $\nu \ (\text{m}^2/\text{s})$ is unburned zone kinematic viscosity, $L_1 \ (\text{m})$ is integral length scale, while $u' \ (\text{m/s})$ is the turbulence intensity. In order to estimate the turbulence intensity from equation (3) the kinematic viscosity, laminar flame speed and integral length scale have to be predicted. The laminar flame speed of gasoline is defined using the correlation developed by Metghalchi and Keck [9] and it is function of the instantaneous in-cylinder pressure, temperature of unburned zone, excess air ratio and combustion product concentration. The approximation of kinematic viscosity is performed using the temperature of unburned zone [10], while the correlation of integral length scale is made to the instantaneous cylinder height. Within the study presented in [3] it was
demonstrated that the geometrical length scale of turbulent eddies in the cylinder volume does not change significantly during the expansion when the engine speed is changed which confirms the geometrical feature of length scale. Therefore, it is reasonable to make the correlation of integral length scale using the cylinder height (that changes with the piston movement) when the engine operating conditions are changed. In this study for the estimation of turbulence intensity, the integral length scale is approximated to 25% of instantaneous cylinder height which is function of piston position.

Evaluation of Flame Front Surface Area

In the newly developed FTM that is integrated in AVL BOOST™, the free flame surface area is evaluated considering the homogeneous icosahedron sphere and its intersection with the cylinder walls. For the given piston position and flame radius, the summation of surface area of triangles that are within the cylinder domain yields to the total area of the flame front, while the burned volume is evaluated calculating the particular tetrahedron volumes. More details of the applied procedure for the geometry description can be found in [5]. The main novelty in the combustion chamber geometry description is the possibility to consider the complex combustion chamber design where 3-D CAD model can be imported. The mentioned model feature was applied in this study since the piston has original ω-shaped bowl, as shown in Figure 1 (left). The local central axis of piston bowl is shifted by 7.7 mm at y-axis.

Due to existence of in-cylinder convective flow the flame kernel center of gravity may be shifted from the spark position in the route which corresponds to the local flow velocity vector [11]. The previous application of quasi-dimensional ignition model that calculates the shifting of kernel center indicates the shifting of 2.5 mm at 1000 rpm and 8.7 mm at 3000 rpm of engine speed. In this study, the shifting direction is assumed and it corresponds to positive y-axis, as shown in Figure 1 (right). For the determination of flame surface area during the combustion, the geometrically defined flame area and burned volume table are calculated for each operating condition. The free flame surface area relationship to the burned gas volume for different piston positions and for two engine speeds are plotted in Figure 2. Such tabulated data are used in the developed code for the estimation of in-cylinder turbulence intensity. For the given piston position the linear interpolation is applied for the calculation of burned volume location between the tabulated piston positions. Once the burned volume is calculated for the given piston position, the free flame surface area required for equation (1) can be defined from geometrical maps, as presented in Figure 2.
COMPARISONS OF ESTIMATED/EXPERIMENTAL DATA WITH SIMULATION RESULTS

The described procedure for the estimation of turbulence intensity was applied over the operating conditions of experimental engine. The estimation of turbulence intensity was performed during 80° CA starting from the defined ignition timing so that the entire combustion process is taken into account. The estimated value of turbulence intensity corresponds to the average value at the flame front that satisfies measured in-cylinder pressure and main equations of quasi-dimensional model presented by equations (1) and (3). Since the newly developed FTM considers different turbulence intensity at the flame discretization points, the averaging of turbulence intensity over the flame points was required and the average value was compared with the estimated value. The profile of estimated turbulence intensity in the flame at engine speed equals to 1500 rpm is shown in left diagram (triangle markers) of Figure 3.

It is visible from the left diagram in Figure 3 that estimated turbulence intensity during the initial and terminal combustion phase has very low values. During the initial period of the combustion process, the laminar-turbulent flame transition occurs and hence the increase of turbulence intensity can be observed. Due to effects of viscous forces and limitation for flow
at cylinder walls when the flame front is near the wall, the decrease of turbulence quantities in flame front occurs which significantly decreases the combustion rate. Therefore, during the last part of the combustion process the estimated turbulence intensity decreases to the minimum level. In order to define a single turbulence intensity value that characterize analyzed engine operating condition the averaging of estimated values between 10% and 90% mass fraction burned was performed because during this phase the developed turbulent flame occurs and turbulence effects on the flame propagation are the most pronounced. The average values of estimated turbulence intensity over the engine speeds at full load conditions are plotted in Figure 3 (right diagram). As the engine speed is increased, the estimated turbulence intensity almost linearly increases. The same behavior in the turbulence intensity increase (with engine speed increase) was presented in the prediction model published in [4], but the averaging of turbulence intensity in their work was performed between 5% and 10% mass fraction burned. The position of turbulence intensity compared to mean piston speed mainly depends on the intake port geometry and engine operating conditions and it is usually within the interval of 0.5 to 1.5 of mean piston speed [12]. The slope lines that correspond to MPS, upper (1.5·MPS) and lower boundary (0.5·MPS), are also plotted in Figure 3. The turbulence intensity level calculated by the proposed procedure is within the expected interval according to the literature.

During the analysis of experimental data at engine speed 1000 rpm the low temperature heat release was noticed and at defined ignition timing (6° CA ATDC) approximately 5% of total heat is released. Since the flame propagation is not representative for the low temperature heat release phenomenon, the estimated turbulence intensity is not fully reliable for the calibration of FTM constants.

The estimated turbulence intensity values were used for the calibration of newly developed FTM [5]. The simulation results of turbulence intensity are plotted in Figure 3 (solid red lines). The turbulence modelling was performed by the application of 0-D $k$-$\varepsilon$ turbulence model [13] and the turbulence intake production constant $S_{10}$ was set to $4.5\cdot10^{-5}$ s/m so that the average difference in turbulence intensity between the estimation and simulation is below 15%. The unique values of turbulence model constants were used for the simulation of analyzed operating points of experimental engine.

**Cycle-Averaged Results**

The comparisons of averaged pressure profiles and burned mass fraction for the analyzed operating points are given in Figure 4. The results of cycle simulation are plotted with the red solid lines. The ignition associated phenomena were simulated with the application of previously developed quasi-dimensional ignition model [6], while the heat losses are simulated with widely used Woschni correlation [14].

As already mentioned before, at engine speed of 1000 rpm the low temperature heat release occurred increasing the pressure in the cylinder before ignition timing. Since the applied quasi-dimensional combustion model cannot capture such phenomenon, the simulation results are reasonable. As the engine speed increases, the ignition timing is set earlier and induction time for the low temperature heat release is lower. Hence the low temperature heat release is suppressed at higher engine speeds. The results of cycle simulation fit the experimental results very good indicating that the FTM constants are correctly calibrated. The unique group of FTM constants was used over different engine conditions.
Cycle-Resolved Results

The experimental results presented in Figure 3 and Figure 4 are related to average cycles which are defined by averaging over 300 operating cycles. During the real combustion process in SI engines the particular heat release profile has non-repeatable feature called as cyclic combustion variability - CCV. In order to define variations of in-cylinder turbulence from one cycle to another, and to capture the most of CCV [6], the estimation of turbulence intensity was performed on each operating cycle for one operating point where the engine speed was fixed to 1500 rpm. The average values of turbulence intensity (averaging between 10% and 90% of mass fraction burned) over 300 cycles are plotted in left diagram of Figure 5.

The selection of slow and fast burning cycle was made by the definition of operating cycle that has the minimum and maximum mass fraction burned at crank angle of 25° after ignition timing. The in-cylinder pressure profiles for average, slow and fast burning cycle are plotted in right diagram in Figure 5. It can be seen from the oscillation of average turbulence intensity that slow burning cycle (# 281) has the turbulence intensity below the average value, but not the minimum one. The same conclusion can be drawn for the fast burning cycle (# 106); the turbulence intensity of fast burning cycle is above the average value, but slightly lower than the maximum one. This confirms that the turbulence intensity variation is not the only parameter that contributes to the CCV, but it is the dominant one [6]. The other parameters that contribute to the CCV is the mixture stratification
(stratification of fuel, air and residual gases) and variation of flame center shifting caused by the stochastic and chaotic nature of local flow that are not captured by the presented model. In the previous study published in [6] it was shown that variation of turbulence intensity covers 90% of total cyclic combustion variability. Therefore, the proposed estimation model for the turbulence intensity can give the insight into the variability of in-cylinder turbulence intensity.

For the simulation of CCV, the slow and fast burning cycle were simulated and simulation results are compared and shown (red solid lines) in right diagram of Figure 5. The slow burning cycle was simulated by decreasing the turbulence intake production constant $S_{10}$ to $3.3 \cdot 10^{-5}$ s/m, while the increase of this constant to $6.5 \cdot 10^{-5}$ s/m was required to correctly capture the fast burning cycle. For the simulation of cyclic combustion variability over 300 consecutive cycles the random perturbation of turbulence intake production constant $S_{10}$ was defined following the normal distribution. From the upper and lower value of this constant it is possible to approximate the relative standard deviation of $S_{10}$ and it was set to 0.1. The CCV was simulated with variability of turbulence level from one cycle to another with the unique value of constants. The cycle simulation results of indicated mean effective pressure (IMEP) and its coefficient of variation (CoV IMEP) are compared in Figure 6.

![Figure 6: Comparisons of averaged IMEP (left) and CoV IMEP as a measure of cyclic variability (right).](image)

The average difference between the measured and simulated IMEP over the analyzed operating points is equal to 4.1%. The cyclic combustion variability is slightly over predicted at first two operating points, while under predicted for engine speeds above 1500 rpm. Under prediction of CCV expressed calculated with CoV IMEP is partially caused by the over prediction of average IMEP values, as it can be concluded from Figure 6 (left diagram). Although the unique set of turbulence and combustion parameters is used, the cycle simulation results correspond well the experimental data on the cycle averaged and cycle resolved basis.

CONCLUSIONS

In the study presented in this paper the estimation model for in-cylinder turbulence intensity was proposed and applied on the experimental SI engine. The governing equations used for the estimation of turbulence intensity are taken from the previously developed FTM that is available in AVL BOOST™. The estimated profile of turbulence intensity during the combustion progress begins and ends with low values because the turbulence effect is the most dominant at freely propagating flame. Therefore, the single value of turbulence intensity that represents each operating point was defined as average value at interval from 10% to 90% of mass fraction burned. The increase of engine speed increases the estimated
turbulence intensity whose values are within the expected boundaries correlated to the mean piston speed. The estimated turbulence intensities over the different engine speeds were applied for calibration of parameters in the cycle simulation model. The average difference between the estimated and simulated turbulence intensity was below 15%. The procedure for the estimation of turbulence intensity was extended to the cycle-resolved base for one selected operating point. It was shown that cyclic variability of turbulence intensity is the most influencing parameter that contributes to CCV in SI engine. On the other hand, the estimated values of turbulence intensity can be used for the definition of perturbations of turbulence constant when the cyclic combustion variability is studied. The presented model for the estimation of turbulence intensity is valid for the operating conditions of SI engine characterized with a thin flame front propagation without the low temperature heat release or knock combustion. It can be useful engineering tool that enables faster calibration of simulation model for the average cycle simulation and CCV simulation when 3-D CFD data are not available.

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