CFD 3D Analysis of Charge Motion and Combustion in a Spark-Ignition Internal Combustion Engine under Close-to-Idle Condition

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Abstract. The increasingly stringent limitations on noxious missions of transport sector highly affect the development of new engines. The operating conditions of the engine at low-load and idle play a relevant role along the regulatory homologation cycles, contributing to overall emissions. In this work, the effectiveness of some solutions to improve the behaviour under close-to-idle operation of a Spark-Ignition motorcycle engine are compared by 3D CFD analyses. Specifically, the effects of two designs of the intake port and of the opening direction of the throttle valve, either clockwise or counterclockwise, are investigated. Multi-cycle simulations are carried out, under motored and fired conditions, for a single close-to-idle operating point. The various designs are compared in terms of capability to generate a stable tumble vortex during the intake phase and to produce an adequate turbulence level at the beginning of the combustion process. The analyses revealed that a clockwise throttle opening can produce enhanced turbulence levels at the end of the compression stroke, especially in a close-to-spark region (increase of about 5% and 27 % at the TDC at a global and local level, respectively, compared to the base configuration). Additional limited improvements are obtained with the high tumbling design, where, however, a penalty on the maximum power output could emerge. The flow and turbulence motion differences among the tested geometries reflect on combustion development in its early stage, and on its degree of completeness at the exhaust valve opening. A clockwise opening of the throttle valve leads to an increase of the mass fraction burned of 5 percent points, compared to the base configuration.

1 Introduction

The emission limits entered in force in 2020 in Europe for the motorbike sector pose challenging tasks in the development of new engines. An idea about the difficulties to comply with legislation emerges observing the comparison between the current Euro 5 standards and the previous Euro 4 ones. Recent limits require a further reduction of the common pollutants,

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namely HC, CO and NO_x, between 12% and 41%, as shown in **Fig. 1**. As known, the current legislation considers that pollutant emissions are measured at the test bench on the World Harmonized Motorcycle Test Cycle (WMTC) [1]. In the framework of the legislation beyond Euro 5, the European Commission is considering the possibility to evaluate the environmental impact of motorcycles under more realistic conditions, introducing a motorcycle-specific Real Driving Emissions (RDE) regulation [2].

Fig. 1. Comparison of HC, CO and NO_x emission limits between Euro 4 and Euro 5.

Beyond the regulatory aspects, public opinion shows an increasing interest on environmental issues, particularly with reference to CO₂ emissions. In this context, for instance, a limitation of fuel consumption for two-wheelers, depending on engine displacement [3], is already in force in China. The adoption of a regulation on CO₂ emission of motorcycles appears a realistic possibility in many other countries. In the light of the above observations, the industry is focusing, not only on the maximization of the full-load performance, but also on the improvement of engine efficiency and pollutant emissions in a wide range of operating conditions.

With reference to high-performance engines, the manufacturer efforts are addressed to the full load operation, with the aim of combining the maximum rated power, with a well-designed torque curve. Unfortunately, the design criteria to meet emission goals and the performance are in most cases in contrast. With the aim of improving the torque performance, a longer valve timing is usually chosen, coupled to a relevant valve overlap. In this way, the exhaust gas flushing of the cylinder and the volumetric efficiency at higher speeds are enhanced. Conversely, a considerable residual gas content under throttled operation may occur, inducing a certain combustion instability [4]. Another typical feature of high-performance engines is a port design which minimizes the flow losses to improve the cylinder filling. Following this approach, even if the trapped mass can increase, the mixture formation and combustion speed could be penalized, especially at lower loads, by poor intensities of charge motion and turbulence. Another aspect to be optimized is the exhaust manifold layout to maximize the cylinder scavenging. The length of the exhaust runner has to be chosen as a compromise between the maximization of torque and power output, by influencing the exhaust pipe gas dynamics [5], and the need for a fast reaching of the light off temperature for the catalytic converter [6], which is influenced by the distance of the catalyst from the exhaust valves. While the automobile industry offers a number of solutions for the above issues [7] (for instance, a variable valve timing or actuation), these solutions are not yet widespread in the motorcycle sector. Some applications offer a variable valve timing, but this last mainly focuses on the optimization of full load behaviour [8, 9, 10]. However, these devices may not be effective for the optimization of mixture preparation and of efficiency at low load. On the other hand, as stated above, the development of standardized procedures for emission measurement makes relevant the optimization of the engine behaviour even at very

low load and at idle. To give an idea, along the WMTC, the time percentage spent by the engine at idle conditions is about 8%. The optimization of the engine behaviour at idle becomes even more relevant considering that the powertrain hybridization, including Start&Stop devices, is not yet widespread in motorbike applications. On the contrary, it substantially helps to reduce or nullify fuel consumption and pollutant emissions of light- and heavy- duty vehicles under stationary conditions.

Based on the above considerations, in this work, the re-design of the intake port of a high-performance motorbike engine is investigated by 3D CFD simulations with the aim of improving the in-cylinder charge motion and the combustion process under close-to-idle operation. In this framework, the effect of the opening direction of the throttle valve is also studied.

The paper is organized as follows: after the introduction, the main engine characteristics are presented, then, the setup and validation of the 3D model are described, and finally, the results of the numerical analyses are discussed, in terms of in-cylinder charge motion formation, turbulence production and combustion development.

2. Engine Description and Tested Operating Condition.

The engine under study is a 4-cylinder high-performance spark-ignition engine for motorbike application, manufactured by Ducati Motor Holding SpA. The combustion chamber presents a typical pentroof design, with a centered spark-plug and four valves per cylinder. Four sequential PFI injectors supply the gasoline in the intake runner of each cylinder. Some geometrical data of the tested engine are listed in **Table 1**. The base design of the intake system (labelled as “GEO1” in the following, **Fig. 2a**) is conceived to maximize the cylinder filling, in order to improve the maximum rated power. A second design is also proposed, shown in **Fig. 2b**, which is conceived to promote the tumble motion (“GEO2”). This is obtained decreasing the angle between the intake port and the cylinder head plate (angle α in the figure), reducing the bending at injector location (which means increasing the angle β in the figure), and introducing a sharp edge at intake valve intrados. The pictures on the bottom of **Fig. 2** depict the throttle valve in a position close to the minimum opening. The picture on the left (right) shows the throttle valve slightly turned in a CounterClockWise - CCW (ClockWise - CW) direction, compared to the fully closed position. The first option is here considered as the reference solution. As stated above, the 3D analysis concerns an operating point close to idle condition, representative of the phase 1 of WMTC. For all the intake configurations, the position of the throttle valve is hence almost fully closed.

Table 2. Engine main features.

Stroke / Bore	0.64
Displacement, cm ³	>1000
Valves per cylinder	4

Fig. 2. 3D view of the combustion chamber, intake and exhaust ducts. (a) Base intake design (GEO1) and (b) tumble-promoting design (GEO2). (c) Counterclockwise and (d) clockwise opening of the throttle valve.

3. CFD 3D Model Setup and Preliminary Validation

AVL FIRE version 2017, licensed by AVL, is used to perform the 3D simulations. Four different meshes are realized, corresponding to the geometrical configurations represented in **Fig. 2**. The 3D domain includes one half of the combustion chamber, which is cut along the symmetry plane between the intake and exhaust valves. It contains the in-head intake and exhaust ports, together with a portion of the related runner (**Fig. 3**). A hexahedral mesh is built, characterized by a maximum cell size of 4.0 mm, with increasing refinement levels in the areas where the greatest velocity gradients are expected. Particular care is devoted to control the mesh refinement around the intake valve and the throttle (**Fig. 4**). It is worth noting that **Fig. 3** refers to a fully closed throttle condition, while **Fig. 4** refers to a wide-open throttle condition.

Fig. 3. 3D domain, including cylinder, intake and exhaust ducts (throttle fully closed).

Fig. 4. Grid refinements near to intake valve and throttle (throttle fully opened).

The simulations are carried out under motored and fired conditions. Instantaneous pressure and temperature at the intake and exhaust ducts are extracted from a 1D model of the same engine, built in the Wave-Ricardo software modelling environment, and imposed as time-varying boundary conditions in the 3D simulations. In this way, the wave propagation in the whole intake and exhaust system is taken into account, in order to well reproduce the actual cylinder filling conditions. The 1D model is also used to extract the wall temperatures, which are assigned in the 3D simulation as time-constant boundary conditions. A uniform value is specified for the different computational domain portions, namely cylinder liner/head, piston, exhaust port/pipe and intake pipe, while a linear temperature gradient is assigned in the intake port. Such simplifying assumptions, even if may affect the consistency of the heat transfer estimation at a local level, should not drastically influence the accuracy of the prediction the in-cylinder flow and turbulence fields, which are the main focuses of the presented work. A similar approach was followed in other researches oriented to the flow description, such as, for instance, [11, 12].

The aspirated fluid at the intake boundary is assumed as a perfectly homogenous mixture of air and fuel vapor, with an imposed air/fuel ratio. Hence, the liquid fuel injection and evaporation processes in the intake port are not simulated. This choice arises from the intention of isolating the burning rate improvements due to the modification of flow and turbulence development inside the cylinder induced by the intake geometry rearrangement, without considering additional effects of air-fuel mixture preparation and inhomogeneity.

A RANS approach is adopted for the flow equations solution, while the turbulence is described by a two-equation $k-\epsilon$ RNG model [13], fitted for compressible flows [14]. A high-Re approach is chosen for the wall treatment, based on proper wall functions to define the near-wall velocity and temperature distributions. The turbulence model is used without any tuning of its constants, which are assumed equal to values indicated in the software manual.

A linearly increasing velocity field, from the cylinder head to the piston, is assigned as initial flow condition. A multi-cycle simulation is performed, until the periodic convergence is reached. The maximum time-step is set in the order of $1e-5$ s, to maintain the average Courant number below the unity, and properly shortened at run-time, depending on flow velocity and cell size.

As an indicator of the in-cylinder turbulence level, the normalized *turbulence intensity* I is computed at each crank angle, as a function of the mean *turbulent kinetic energy* (TKE) k and the mean piston speed, $S_{p,m}$:

$$I = \frac{\sqrt{\frac{2}{3}k}}{s_{p,m}} \quad (1)$$

The turbulence intensity is calculated both as an average over the whole cylinder volume and locally near the spark plug. In this latter case, a local I level is computed over a spherical region, with a radius of 10 mm, centered on the spark-plug electrodes.

As an indicator of the tumble motion intensity, the Tumble Ratio (TR) is computed at each crank angle as the ratio between the angular velocity of the tumble flow field (ω_T) and the engine angular velocity (being n its rotational speed):

$$TR = \frac{\omega_T}{2\pi n/60} \quad (2)$$

The velocity ω_T in turn is defined as the angular velocity of a solid body rotating around the tumble axis, having the same angular momentum and moment of inertia as the actual flow field. The reference frame used in this work is shown in **Fig. 5**. Following this reference, the axis around which is expected to develop the tumble vortex is the y-axis, considering the geometrical symmetry of the combustion chamber. The intake port is designed to promote a counterclockwise tumble vortex around the y-axis, which would determine a negative tumble ratio according to the definition in Eq. (2). To make easier the readability of the results proposed in the following, the opposite of the tumble ratio from Eq. (2) will be reported, which corresponds to a positive value when the tumble vortex has a counterclockwise direction.

Fig. 5. Reference frame.

The reliability of the 3D flow model and its settings is verified in preliminary simulations under steady-state condition. More specifically, those calculations aim to estimate the intake valve flow coefficient for the GEO1 case at different valve lifts and at wide open throttle. The comparison of the numerical results and the experiments are shown in **Fig. 6**. The agreement is satisfactory, having a maximum error of about 3% for the lowest considered lift. For the normalized valve lift of 0.24, a sensitivity analysis of the grid density is carried out, investigating a larger (5 mm) and a smaller (2.5 mm) maximum mesh sizing. The different grids are compared in **Fig. 7**, in terms of flow coefficient and computational time. The selected mesh, having a maximum grid dimension of 4 mm, gives a result comparable with the finest mesh, but with a much more reduced computational effort.

Fig. 6. Comparison of computed and experimental intake valve flow coefficient normalized by the computed level at $h/D_v = 0.24$.

Fig. 7. Comparison of numerical estimations of the intake valve flow coefficient (a) and the related computational time (b) for different grid densities at $h/D_v = 0.24$.

4. Combustion Model Setup and Validation

The ECFM (Extended Coherent Flamelet Model) is used to model the combustion process, originally introduced in [15] and extended in [16]. A transport equation of the flame surface density Σ is solved:

$$\frac{\partial \Sigma}{\partial t} + \frac{\partial}{\partial x_j} (\bar{u}_j \Sigma) - \frac{\partial}{\partial x_j} \left(\frac{\nu_T}{\sigma_\Sigma} \frac{\partial \Sigma}{\partial x_j} \right) = S_g - S_a + S_L \quad (3)$$

where Σ is the flame surface density of the turbulent flame, σ_Σ is the turbulent Schmidt number, ν_T is the turbulent kinematic viscosity. The equation presents, on the right-hand side, the production term related to the turbulent field, S_g , the destruction term due to the consumption of reactants, S_a , and the laminar speed contribution, S_L . This last considers three contributions, related to the propagation, curvature and deformation of the flame.

The model is tuned with reference to the configuration GEO1-CCW, which is the only geometry for which experimental data are available. The adoption of the model constant values advised in the Fire Users' Manual [17] does not realize a good matching with the

experimental pressure cycle. The initial flame surface density (Σ_{init}), the flame kernel dimension (d_{kern}), and the turbulence enhancement multiplier (acting on S_g , and labelled as k_{sg}) are hence substantially modified with reference to their default values (300 m⁻¹, 2 mm and 1.6, respectively). After a trial and error process, a set of tuning constants is identified with the aim of minimizing the difference between the computed pressure cycle with the experimental counterpart ($\Sigma_{init} = 75$ m⁻¹, $d_{kern} = 3$ mm and $k_{sg} = 0.4$). The comparison between the numerical pressure cycle and the experimental counterpart is shown in **Fig. 8**. The figure highlights that the combustion progress is extremely slow in the close-to-idle operation under investigation. The combustion mainly evolves during the expansion stroke, and this is primarily due to the selection of a late spark timing. For the same reasons, the pressure cycles resemble the motored counterparts around the TDC. The choice of a delayed spark timing arises from the need to limit the engine load and to maintain the temperature of the exhaust gas entering the catalytic converter at an adequately high level.

Fig. 8. Comparison between the ensemble averaged measured pressure cycle and the computed pressure trace for the configuration GEO1-CCW.

Some numerical/experimental differences emerge at the TDC and after about 60 CAD aTDC. The disagreement at the TDC may be justified by not modelling the crevices volumes, which would have led to a reduced actual compression ratio, and by neglecting the blow-by process. The misalignment during the expansion phase is mainly due to the difficulties in reproducing, in the adopted simulation approach, the consequence of the cyclic variability, which characterizes the selected operating condition. It is not worthless to underline that the experimental pressure cycle is obtained by the ensemble averaging process of a sequence of pressure cycles, and hence the reference cycle may not correspond to none of the actual individual pressure traces. Despite the above-mentioned inaccuracies, the experimental / numerical agreement appears satisfactory, especially considering the quite unusual operating point under investigation, characterized by a very low intake pressure and high residual content. To confirm the consistency of the selected set of tuning constants, a sensitivity analysis on Σ_{init} , d_{kern} , and k_{sg} , is performed, whose results is shown in **Fig. 9**. The analysis entails the modification of one parameter at a time, starting from the identified optimal values. As expected, **Fig. 9** puts into evidence that higher values of Σ_{init} and d_{kern} speed up the initial phase of the combustion process, while k_{sg} mainly enhances the combustion core.

Fig. 9. Combustion model sensitivity analysis for the configuration GEO1-CCW : (a) initial flame surface density (Σ_{init}), (b) flame kernel dimension (d_{kern}), and (c) turbulence enhancement multiplier (k_{sg}).

5. Results Discussion

A first comparison between the analysed configurations is shown in **Fig. 10**, which depicts the tumble ratio trend during the intake and the compression strokes. All trends are normalized by the tumble level at IVC of the base geometry (GEO1-CCW). The figure puts into evidence that, during the first part of the intake phase, a reverse tumble vortex is generated. The different geometries behave in a similar way during this first stage. Moving towards the ending part of the intake phase, a positive tumble vortex is established in all cases, having the same sign, but different intensities. The base configuration (GEO1-CCW) presents the lowest tumble level at the BDC, but it recovers during the first part of the compression stroke. Later, the well-known phenomenon of tumble collapse occurs. In all the other cases, higher tumble levels are reached at BDC, followed by a continuous decrease from BDC to TDC. A clockwise throttle opening induces an advanced tumble collapse. The GEO2-CCW, combining a high tumble ratio at BDC with a delayed collapse, produces the highest tumble levels at IVC and during the compression stroke.

Fig. 10. Comparison of the tumble ratio during intake and compression strokes among the considered geometrical configurations.

Fig. 11. Comparison of the velocity flow field over a cross plane parallel to the symmetry plane among the considered geometrical configurations at -300 CAD aTDC.

Fig. 12. Comparison of the velocity flow field over a cross plane parallel to the symmetry plane among the considered geometrical configurations at -220 CAD aTDC.

The differences among the geometrical configurations in the capability to generate a stable tumble vortex at the end of the intake phase are clearly underlined by the velocity fields shown in **Fig. 11-13**, which refer to -300, -220 and -180 CAD aTDC, respectively. The figures depict, in the same colour scale, the velocity field in a cross plane parallel to the symmetry plane and passing through the intake valve stem. The pictures (especially see **Fig. 12** and **Fig. 13**) underline that, in all cases, a main positive vortex arises in the cylinder, centered just below the intake valve disc. This is generated by the flow coming out from the intake port extrados. In addition to the main tumble vortex, an ordered flow emerges from the intake valve intrados, which moves along the cylinder liner near to the intake side (right side in the figures). This secondary flow is more intense for the configurations with the throttle valve opened in a CCW direction (GEO1-CCW and GEO2-CCW). For the GEO1-CCW configuration, it is so intense to persist towards the end of the intake phase. This secondary flow causes, in all cases, the onset of secondary vortices, which most likely have a direction opposite to the main tumble vortex.

Fig. 13. Comparison of the velocity flow field over a cross plane parallel to the symmetry plane among the considered geometrical configurations at -180 CAD aTDC.

An overview about the flow distribution around the intake valve is given in **Fig. 14**. It depicts the normalized mass flow through eight sectors of the curtain area at the same crank angles as considered before, namely at -300, -220 and -180 CAD aTDC. **Fig. 14** confirms that GEO1-CCW drives a large portion of the flow towards the valve intrados, especially at end of the intake phase (**Fig. 14b-c**). GEO2-CCW, which determines the highest tumble intensity at the BDC, is the solution which is able to combine a large flow towards the valve extrados, and reduced flow towards the intrados, during the whole intake phase. The configurations with a CW opening promote an intrados flux in a first stage (**Fig. 14a**), followed by an extrados flux prevalence (**Fig. 14b-c**). The above discussed differences help

to explain the turbulence intensity trends in **Figs. 15** and **16**. The former concerns the average turbulence intensity, while the latter refers to the local level near to the spark-plug. Displayed values are normalized by the average turbulence intensity at IVC of the base configuration.

Fig. 14. Comparison of the mass flow distribution through the intake valve curtain area among the considered geometrical configurations at different crank angles: -300 (a), -220 (b) and -180 CAD (c) aTDC.

Fig. 15. Comparison of the in-cylinder averaged turbulence intensity during intake and compression strokes among the considered geometrical configurations.

Fig. 16. Comparison of the local close-to-spark turbulence intensity during the compression stroke among the considered geometrical configurations.

The geometry which leads to the highest tumble at the end of the intake phase (GEO2-CCW) also determines the highest average turbulence at the same crank angle. The differences among the configurations in terms of average turbulence intensity reduce drastically moving through the compression phase, leading to similar peak values just before the TDC. The tumble destruction, which is observed since the middle of the compression strokes, occurs with a similar rate among the tested cases, but with a different timing (see trends in **Fig. 10** after -90 CAD aTDC). According to the well-known energy cascade mechanisms from ordered flow structures to turbulence, the tumble collapse is responsible for the turbulence intensity rise before the TDC. The advanced tumble destruction in case of CW opening of the throttle valve also determines a higher rate in the turbulence increase towards the compression end. This effect is less intense in the cases of a CCW opening, which allows the tumble vortex to persist later during the compression phase. In those cases, the initial turbulence levels at the compression start are lower than the CW counterparts. The balance between increased rate and initial turbulence levels causes, as stated above, similar peaks towards the compression end. The local turbulence trends in **Fig. 16** underline greater differences among the considered geometries. A CW throttle opening leads to higher turbulence intensities at the TDC. The additional information in this case is that GEO2 gives higher turbulence levels compared to GEO1, whatever is the throttle valve opening direction. The reason of the different behaviour between local and global trends can be derived by the observation of the TKE fields depicted in **Fig. 17**, **Fig. 18**, and **Fig. 19**. These represent the turbulence fields in the symmetry plane at different times, namely -40, -20 and 0 CAD aTDC. The pictures at the earlier angle (**Fig. 17**) point out that the GEO2 determines a “high-turbulence” zone close to the spark-plug, with lower levels in other locations (for instance in the squish zone on the exhaust side). On the contrary, GEO1 leads to a more homogeneous TKE distribution, with less extended “high-turbulence” zones. For the GEO1-CCW case, the larger “high-TKE” area is close to the piston on the intake side, while, in the spark-plug proximity, the “high-TKE” zone is the smallest among all the configurations.

Fig. 17. Comparison of the TKE field over the symmetry plane among the considered geometrical configurations at -40 CAD aTDC.

The above differences enhance moving towards the TDC, as shown in **Fig. 18** and **Fig. 19**. Looking to this last figure, the cases with a CW throttle opening present a TKE peak close

to the spark-plug, over a rather extended area. On the contrary, the configurations with a CCW throttle opening do not reveal the maximum TKE level close to the spark-plug, and, in the GEO1-CCW case, the TKE peak is far away from this.

Fig. 18. Comparison of the TKE field over the symmetry plane among the considered geometrical configurations at -20 CAD aTDC.

Fig. 19. Comparison of the TKE field over the symmetry plane among the considered geometrical configurations at TDC.

The turbulence evolution, as expected, impacts on the combustion process as clearly evidenced by the comparison of in-cylinder pressure traces and of the related mass fraction burned, depicted in **Figs. 20a-b**, respectively. Whatever is the intake configuration, the

combustion process develops late during the expansion stroke. A portion of the fuel is not yet burned when the exhaust opens, which in turn affects the cylinder-out HC emission and the engine efficiency. The re-design of the intake duct (GEO2), coupled to the employment of a CW throttle opening, shows some potential advantages compared to the original solution (GEO1-CCW), in terms of both combustion speed and degree of completeness.

Fig. 20. Comparison between the in-cylinder pressure cycles (a) and the mass fraction burned evolution (b), among the considered geometrical configurations.

Fig. 21. Comparison of the mass fraction burned over the symmetry plane among the considered geometrical configurations at 100 CAD aTDC.

The differences among the geometries emerge especially at the beginning of the combustion process, highlighting, for the considered operating condition, a greater effectiveness in the burning speed enhancement of the throttle opening direction compared to the intake port geometry. This is consistent with the TKE local levels (**Fig. 16**) and distributions (**Fig. 19**)

before the TDC. Starting from the base configuration (GEO1-CCW), no significant combustion enhancement appears changing the intake geometry (GEO2-CCW), while the sole modification of the throttle valve opening leads to a certain speed up of the initial burning speed, partially compensated by a slowdown towards the combustion tail, with a global improvement of about one percent point of MFB at the EVO (GEO1-CW). The maximum improvement is reached additionally modifying the intake port geometry (GEO2-CW). In this case, the speed up of the combustion development persists throughout the whole process, leading to an MFB increase at the EVO of about 5% compared to the base configuration, reaching the 94 %. This improvement is expected to reduce the unburned HC emissions of the engine under the considered operating condition. To highlight the combustion development more deeply in the analysed solutions, the distribution of the burned mass fraction is shown in **Fig. 21** over the symmetry plane at 100 CAD aTDC. Consistently with the trends in **Figs. 20**, a CW throttle opening determines a faster flame propagation, enhancing particularly the combustion in the region under the exhaust valve sector.

6. Conclusions

An assessment of different geometries of the intake port design was presented in this work by means of a 3D CFD analysis. Specifically, the geometry of the intake port was modified, passing from the base configuration (GEO1), designed to maximize the cylinder filling, to a modified version, which was proposed to promote the tumble motion (GEO2). Moreover, the influence of throttle valve rotation direction was investigated. The study concerned a high-performance spark-ignition engine operating in a close-to-idle condition. Multi-cycle simulations were performed to guarantee the cyclic convergence.

In a first stage, the capability of the intake system in generating an ordered tumble vortex was investigated with reference to both global parameters and velocity flow fields. GEO2, coupled to a CW throttle opening, led to the highest tumble intensities at the end of the intake phase.

The analysis of the global turbulence intensity evidenced that a CW throttle opening determined an advanced collapse of the tumble vortex. In the cases with CCW throttle opening, higher turbulence levels were attained at IVC. The combination of varied turbulence levels at IVC and intensities of tumble collapse caused a certain compensation at the TDC, where similar values were attained among the tested geometries. The differences resulted more evident at a local level, near to the spark-plug. The GEO2 configuration, coupled to a CW, determined an increase of turbulence intensity at the TDC of about 5% and 27% at global and local level, respectively, compared to the base case (GEO1-CCW). The higher turbulence levels provided by a CW throttle opening reflected on the combustion process evolution. This last resulted very slow for all the tested operating condition whatever was the intake configuration, with a combustion development which occurred late during the expansion stroke and relevant unburned levels. The GEO2-CW geometry however provided a higher degree of combustion completeness, which passed from 89% of the base case up to 94%.

The study gave an insight of a practical solution to improve the engine behaviour under close-to-idle operation, which is relevant for the limitation of the HC emission and fuel consumption along current homologation cycles.

Definitions/Abbreviations

1D/3D	One/Three-Dimensional
CFD	Computational Fluid Dynamics

aTDC	after Top Dead Centre
BDC	Bottom Dead Centre
CAD	Crank Angle Degree
CCW	CounterClockWise
CW	ClockWise
ECFM	Extended Coherent Flamelet Model
EGR	Exhaust Gas Recirculation
EVC	Exhaust Valve Closure
EU	European Union
I	Turbulence Intensity
ICE	Internal Combustion Engine
IVC	Intake Valve Closing
MFB	Mass Fraction Burned
PFI	Port Fuel Injection
RANS	Reynolds-Averaged Navier-Stokes
RDE	Real Driving Emission
RNG	ReNormalization Group
SI	Spark Ignition
TDC	Top Dead Centre
TKE	Turbulent Kinetic Energy
TR	Tumble Ratio
WMTC	World Harmonized Motorcycle Test Cycle

Symbols

d_{kern}	Flame kernel dimension
D_v	Intake valve diameter
h	Intake valve lift
k	Turbulent kinetic energy
k_{sg}	Turbulence enhancement multiplier
Re	Reynolds number
S_a	Destruction term of surface flame density due to reactants consumption
S_L	Laminar Flame Speed
S_g	Production term of surface flame density due to turbulence
t	Time
V	Volume
Sp,m	Mean piston speed

Greeks

α	Angle between the intake port and the cylinder head plate
β	Angle between the throttle valve plate and the cylinder head plate
ε	Dissipation of turbulent kinetic energy
λ	Relative air-fuel ratio
ρ	Density
Σ	Flame surface density
Σ_{init}	Initial flame surface density
ν_T	Turbulent kinematic viscosity.
σ_Σ	Turbulent Schmidt number
ω_T	Angular velocity of the tumble vortex

References

1. Regulation (EU) No 168/2013 of the European Parliament and of the Council of 15 Jan. 2013 on the Approval and Market Surveillance of Two- or Three-Wheel Vehicles and Quadricycles.
2. J. Hiesmayr, S. Schmidt, S. Hausberger, R. Kirchberger et al. *Results, Assessment and Legislative Relevance of RDE and Fuel Consumption Measurements of Two-Wheeler-Applications*. SAE Tech. Paper 2017-32-0042, (2017)
3. TransportPolicy.net. Available online: <https://www.transportpolicy.net/standard/china-motorcycles-fuel-consumption/> (accessed April 2018).
4. J.B. Heywood, *Internal Combustion Engine Fundamentals*; McGraw-Hill: New York, USA, p. 424-425. (1988)
5. D.E. Winterbone, R.J. Pearson, *Design Techniques for Engine Manifolds: Wave Action Methods for IC Engines*; Professional Engineering Publishing Limited: London and Bury St. Edmunds, UK, (1999)
6. E. Otto, F. Albrecht, J. Leibl, *The Development of BMW Catalyst Concepts for LEV / ULEV and EU III / IV Legislations 6 Cylinder Engine with Close Coupled Main Catalyst*. SAE Tech. Paper 980418, (1998)
7. G. Mincione, *Ventiltriebskonzepte zur Verbrauchsreduzierung bei Motorradmotoren*. Dissertation University Stuttgart, (2013)
8. Honda Technology Close-up, Hyper V-TEC. Available online: <http://world.honda.com/motorcycle-technology/vtec/> (accessed on May 2018).
9. Ducati.com, Testastretta DVT. Available online: http://www.ducati.com/en/testastretta_DVT/index.do#start (accessed on May 2018).
10. H. Yoshitake, Y. Horiuchi, H. Arima, Y. Utsumi, *Development of Touring Motorcycle with a 1,352 cm³ Engine*. SAE Tech. Paper 2009-32-0109, (2009)
11. C. Yin, Z. Zhang, Y. Sun, T. Sun, R. Zhang, Eng. App. of Comput. Fluid Mech. **10**(1), 311-329, (2016)
12. P. Janas, I. Wlokas, B. Böhm, Flow Turb. Comb., **98**, 237–264 (2017)
13. V. Yakhot, S.A. Orszag, I. Basic Theory. J. Sci. Comput., **1**, 3-51, (1986)
14. Z. Han, R.D. Reitz, Comb. Sci. Tech, **106**(4-6), 267-295, (1995)
15. F.E. Marble, J. E. Broadwell, *The Coherent Flamelet Model for Turbulent Chemical Reactions*. Project Squid TRW-9-PU, (1977)
16. T. Poinso, D. Veynante, *Theoretical and Numerical Combustion*. 3rd ed. (2011)
17. FIRE v2009.1 - ICE Physics & Chemistry Users Guide.