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Assessment of the Miller cycle operation in a spark ignition engine via 1D numerical simulation

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Assessment of the Miller cycle operation in a spark ignition engine via 1D numerical simulation

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Abstract. The article presents the results of a 1D numerical simulation of a spark ignition engine developed to operate in Miller cycle. Miller cycle offers better thermal efficiency compared to Otto cycle due to higher volumetric expansion than compression, which in the current context is of paramount importance. In an engine with fixed geometric compression ratio, Miller cycle operation could be realized by means of either early intake valve closing (EIVC) or late intake valve closing (LIVC). Both cases lead however to a lower volumetric efficiency, thus reducing the indicating mean effective pressure, which in its turn results to a lower power output. The simulation's aim is not only to assess the impact of implementing the Miller cycle but also to obtain the necessary results for imposing the boundary conditions in a 3D CFD simulation whose purpose is to analyse the influence of the Miller cycle on the internal aerodynamics of the engine.

1. Introduction

In order to achieve the global target of 55% reduction of greenhouse gases (GHG) by 2030 compared to 1990, European Union set a target of CO_2 emissions reduction from passenger cars and vans by 55% compared to 2021 and further reduction of 100% by 2035. In short term, internal combustion engines would be the main power source in transportation due to the higher vehicle autonomy, lower price and higher fuel energy density. For that reason, an improvement in thermal efficiency is important in order to respect the future CO_2 regulations. Over expanded engine operating cycle known as Miller-Atkinson cycle is a promising technology to increase engine efficiency and to reduce pollutant emissions [1,2]. The Miller cycle is based on late or early IVC, due to that it is easier to implement in both gasoline and diesel engines. The intake process could be also optimized when variable valve train (VVT) mechanism is used [3,4]. However, the optimization of gas-exchange processes and internal aerodynamics is important in order to achieve better engine performance and efficiency. The Miller operating cycle is widely studied numerically by means of 0D-1D and CFD simulations [3]. The validation of CFD modeling is usually done on a single cylinder optical engines operating at motored mode. However, in these experimental engines the blow-by could be higher as well as clearance volume due to crevices. Moreover, the blow-by increases at higher compression ration typical for Miller cycle. The study conducted on a motored engine in [5] revealed blow-by mass losses of nearly 15% at compression ratio of 12.4 with the maximum blow-by mass loss within the range of 6-9°CAD after top dead center. Another study [6] reported blow-by mass ratio within the range of 9% to 18% depending on compression ratio and engine speed. The results are also evaluated on a motored engine. The blow-by not only has significant impact on cylinder pressure but also it affects the peak position of the pressure so called loss angle. This effect was studied in [7].

The aim of this paper is to study the effect of blow-by on in-cylinder pressure of a single cylinder motored engine with high compression ratio developed to operate in Miller cycle. The blow-by mass flow will be further used as boundary condition in a 3D CFD model of the same engine in order to study the internal aerodynamics.

2. Engine simulating model

A 1D engine model on a single cylinder motored engine was developed in the advanced simulation software AVL Boost (Fig. 1). The pressure at the boundary sections of the intake and exhaust system were experimentally obtained, thus only the cylinder and valve ports were modeled with their geometric parameters. The main elements in the engine model are: general engine element (E1), cylinder (C1), plenums (PL1 and PL2), pipes (1 to 6) and system boundaries (SB1 and SB2).

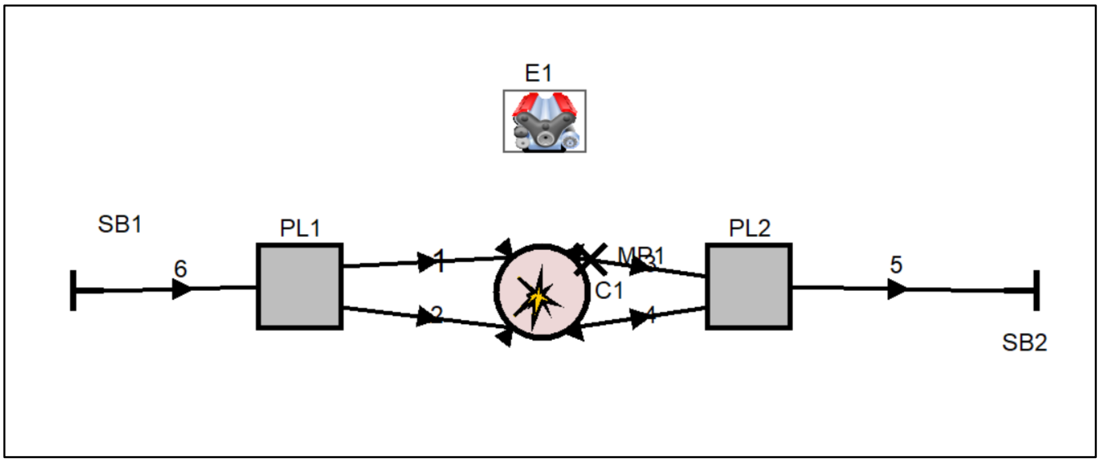

Figure 1. Engine simulation model in AVL Boost simulation software.

The main parameters such as engine speed and simulating cycle were defined in the element E1. Due to the fact that the simulation was done at motored mode (unfired operation), the mechanical losses in the engine were neglected. The main engine data are listed in Table 1.

Table 1. Engine data.

Parameter	Value
Type	Mono-cylinder
Cylinder bore	72.2 mm
Piston stroke	70 mm
Volume	286.4 cm^3
Compression ratio	12.3
Valves per cylinder	4
Operating mode	Motored
Engine speed	1200 rpm

The element called “Cylinder” was used to define the geometrical parameters of the cylinder, the blow-by gap, valve ports and lift curves as well as heat transfer to the cylinder wall. The blow-by model is based on the following correlations:

$$Blow - by\,flow\,rate = \frac{dm_{bb}}{dt} = A_{eff} \cdot p_o \cdot \sqrt{\frac{2}{R_0 . T_0}} . \psi \tag{1}$$

where: A_{eff} $[m^2]$ is the blow-by effective flow area; p_0 $[Pa]$ and $T_0[K]$ are the in-cylinder pressure and temperature; R_0 $[J/Kg.K]$ is the air constant and ψ is the flow function. The latter is estimated as:

$$\psi = \sqrt{\frac{k}{k-1} . \left[\left(\frac{p_2}{p_0}\right)^{\frac{2}{k}} - \left(\frac{p_2}{p_0}\right)^{\frac{k+1}{k}} \right]}, \tag{2}$$

where p_2 is the crankcase pressure, which was assumed to be a constant and equal to the atmospheric pressure and k is the ratio between the specific heats.

The blow-by effective flow area is estimated as follows:

$$A_{eff} = D \cdot \pi \cdot \delta, \tag{3}$$

where $D[mm]$ is the piston diameter and $\delta[mm]$ is the gap between piston and cylinder.

Heat transfer was estimated with the commonly used Woschni model, presented in [8]. The model estimates wall heat transfer coefficient as a function of the cylinder bore, cylinder pressure and temperature as well as mean piston speed. Due to the motoring simulation, the temperature of the cylinder wall, piston top and cylinder head were assumed to be equal. The plenums “P1” and “P2” represents the junctions at cylinder head intake and exhaust ports. System boundaries were used in order to determine experimentally measured pressure at that points as a function of the crank angle.

The valve lift curves are developed for engine operating in Miller cycle. For that reason, the exhaust valve opening duration is higher while the intake valve maximum lift is lower. The relative valve lifts are presented in Figure 2.

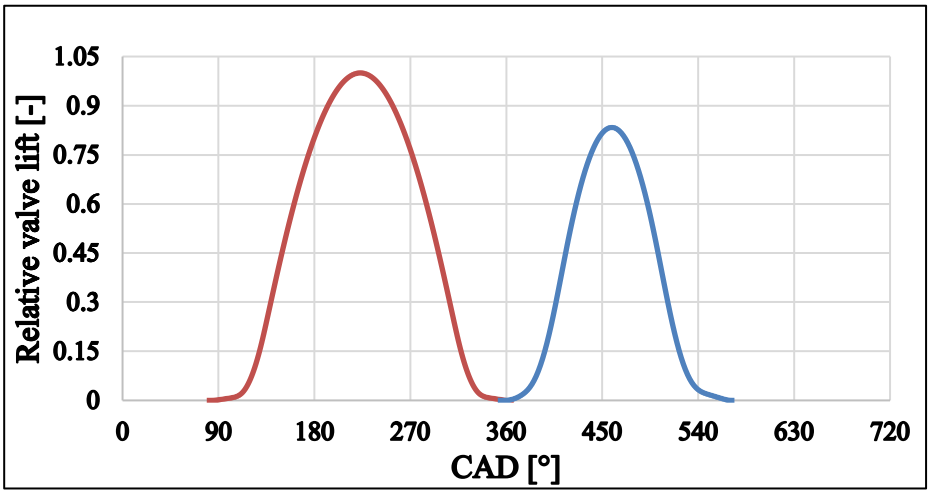

Figure 2. Valve lifts for operating in Miller cycle

3. Experimental data from engine motoring operation

In order to validate 1D simulation model as well as further validation of 3D CFD model an experimental study was conducted on a single cylinder optical engine. This engine was equipped with pressure transducers at the intake and exhaust cylinder head ports. Thus, further modelling of the whole intake and exhaust system geometry is not needed. The instantaneous pressure data was recorded for the studied operating points as a function of crank angle degrees (CAD) by increment of 1°CA. The experimental set-up can be seen in Figure 3.

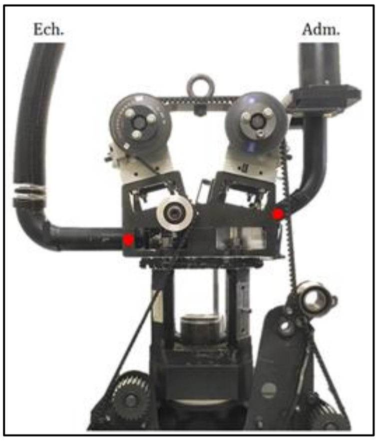

Figure 3. Single cylinder optical engine

The experiments were carried out on engine operating in motoring mode at 1200 rpm. Four cases were studied as intake valve opening point was varied by 10 degrees at each study. Thus, in the first run the intake valve timing was set to the basic value as shown in Figure 2, then intake valve opening was advanced by 10, 20 and 30 degrees compared to basic valve timing. In general, earlier intake valve opening lead to lower in-cylinder pressure at the beginning of the intake stroke. The intake and exhaust pressure measured at cylinder head intake and exhaust ports for the last run (IVO advanced by 30 degrees) are presented in Figure 4. The TDC at the end of compression stroke corresponds to 360° CAD.

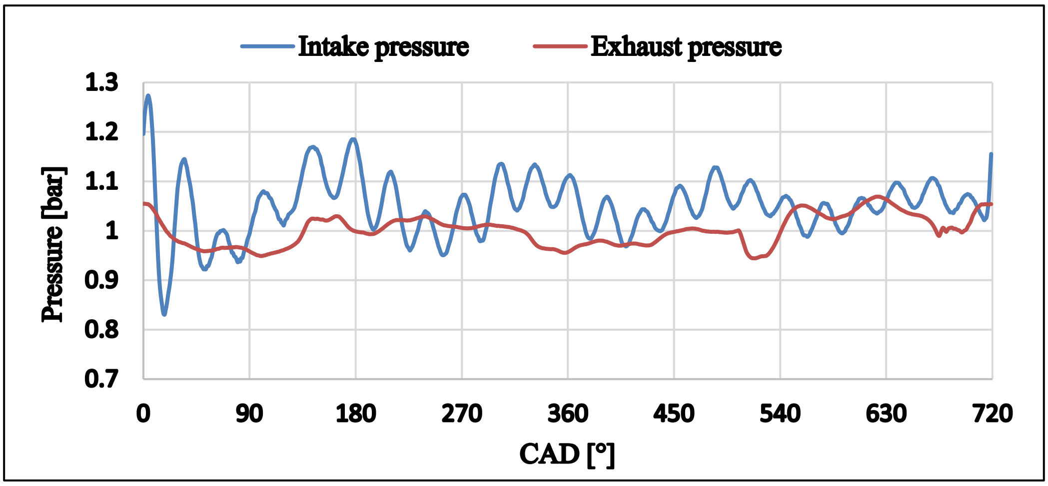

Figure 4. Experimentally measured intake and exhaust pressure

4. Simulating study

A 1D model was used in order to study the impact of both heat transfer and blow-by effect on in-cylinder pressure and loss angle. The blow-by effect was simulated by means of variation of blow-by gap in the model while the heat transfer effect was assessed by variation of heat transfer multiplier coefficient in the model.

4.1. Study the impact of in-cylinder heat transfer

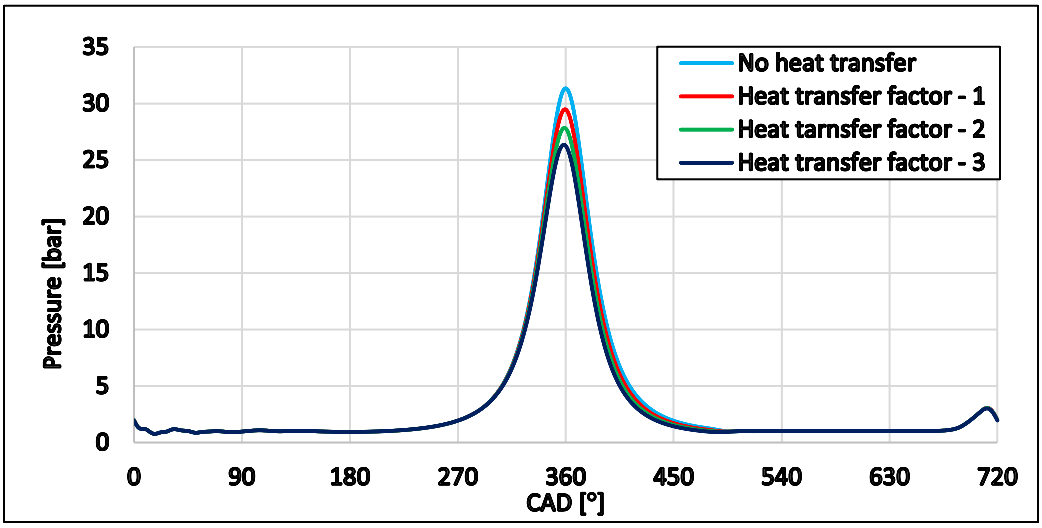

Figure 5. In-cylinder motoring pressure with different heat transfer rate

In order to modify in the model, the heat flux through the cylinder wall, piston top and cylinder head, the estimated heat transfer coefficient was multiplied by a coefficient for each case. It means that the instantaneous heat transfer value is multiplied by a coefficient. In this study the blow-by gap was set to 0 and the heat transfer multiplier coefficient varied from 0 to 3. The results concerning in-cylinder pressure are presented in Figure 5.

As the pressure and temperature has significant impact on the heat transfer coefficient, the impact on the in-cylinder pressure is important due to compression and expansion stroke. If no heat transfer is considered, the maximum in-cylinder pressure was estimated to be 31.3 bar as no loss angle was observed. Applying the heat transfer coefficient with factor 1, which means values estimated by original Woschni model were considered, reduced the maximum in-cylinder pressure to 29.5 bar as well as a loss angle of 0.6 CAD was observed. Increasing the heat transfer by multiplying factor 3 consistently reduced maximum in-cylinder pressure as well as the pressure during the expansion stroke. The loss angle was increased to 1.6 CAD. Due to the fact that the heat transfer in our case has no significant

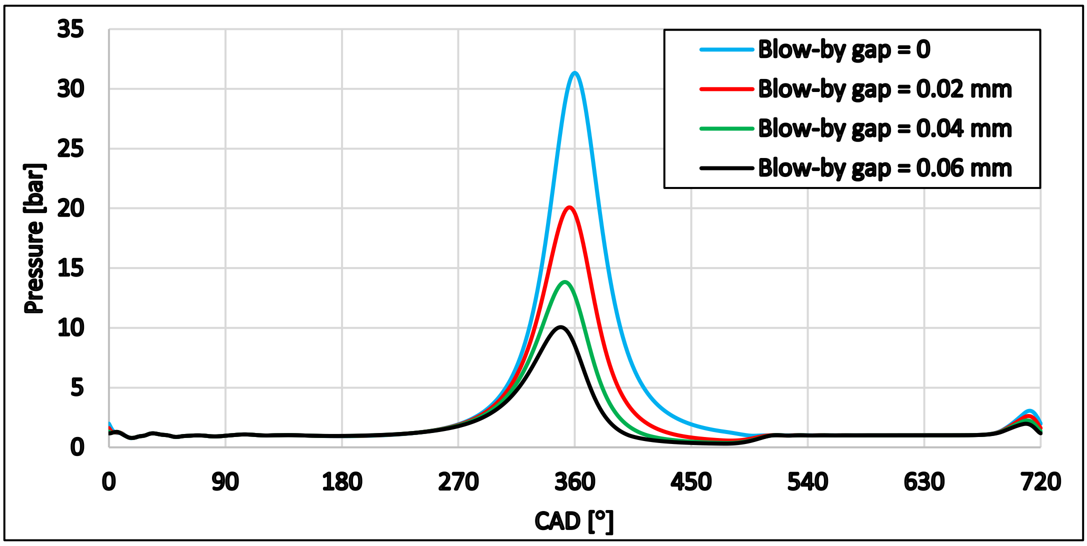

Figure 6. In-cylinder motoring pressure with different blow-by gap

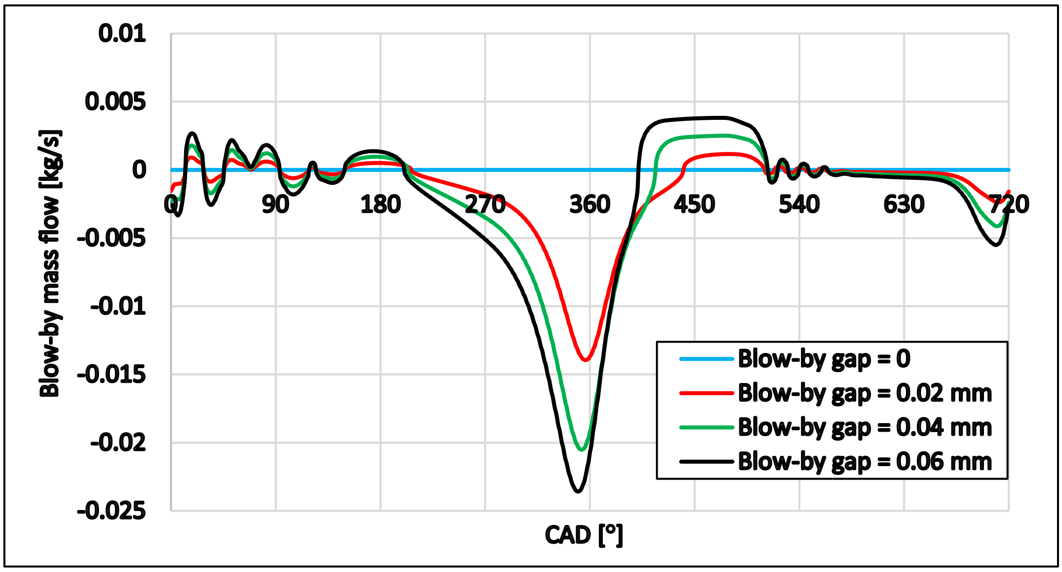

Figure 7. Blow-by mass flow with different blow-by gap

impact on gas exchange processes, we choose to use further original Woschni model with heat transfer factor equal to 1.

4.2. Study the impact of blow-by gap

The blow-by mass flow and the impact on in-cylinder pressure was studied as the blow-by gap in the model was set to the value within the range of 0 mm to 0.06 mm. The crankcase pressure was set to constant value equal to the atmospheric pressure. By means of the simulating model, in-cylinder pressure was evaluated as well as the instantaneous blow-by mass flow as well as the intake and exhaust mass flow. Figure 6 shows the effect of blow-by on in-cylinder pressure. Increasing the blow-by to 0.02 mm reduced the maximum in-cylinder pressure to 20.1 bar compared to no blow-by case and loss angle was observed to be 4 deg. Further increasing of blow-by gap led to even lower maximum pressure. At blow-by gap of 0.06 mm the maximum in-cylinder pressure was reduced to 10.1 bar with loss angle of 8.5 deg. In order to better understand the impact of blow-by gap, the blow-by mass flow need to be analyzed. The results at different blow-by gaps are shown in Figure 7. The main cylinder discharge through blow-

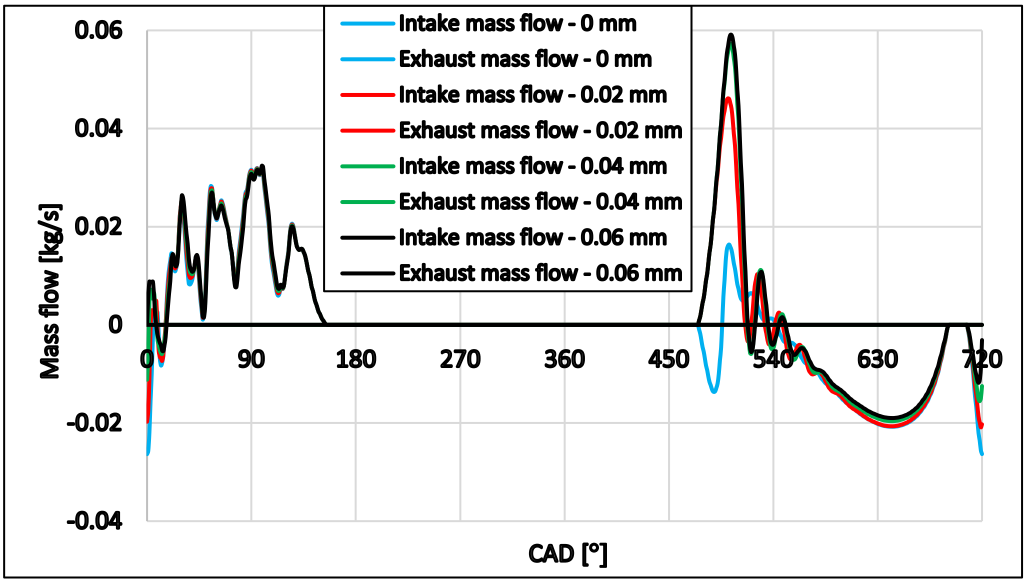

Figure 8. Intake and exhaust mass flow with different blow-by gap

by gap was observed during the compression stroke which led to reduction of in-cylinder mass. As a result, it lowers the in-cylinder pressure below atmospheric pressure during the expansion (Figure 6). Due to lowering the in-cylinder pressure below atmospheric pressure a significant cylinder charge through blow-by gap was observed. It became important at blow-by gaps higher than 0.02 mm. The blow-by gap has no important impact on blow-by mass flow during intake and exhaust stroke. However, at the end of exhaust higher blow-by gap led to higher blow-by mass flow when valve lift curves are developed for engine operating in Miller cycle.

Due to the significant impact on in-cylinder pressure during expansion stroke blow-by gap led to important cylinder backflow at the beginning of exhaust process. The mass flow through the valves is shown in Figure 8. It was observed that even small blow-by gap of 0.02 mm led to high backflow peak. However, increasing the blow-by gap to 0.06 mm did not lead to further increasing in the backflow compared to blow-by gap of 0.04 mm. Higher blow-by gap also led to higher cylinder discharged around TDC at the end of the exhaust stroke.

5. Conclusion

A numerical study of blow-by mass flow on a single cylinder motored engine operating under Miller cycle was presented. The model of the engine was developed by means of 1D simulation software AVL BOOST. Intake and exhaust pressure at the cylinder head ports were experimentally measured on a single cylinder optical engine at 1200 rpm. Firstly, the effect of heat transfer on in-cylinder pressure was discussed when using original Woschni heat transfer model and multiplying factor. Without heat transfer and blow-by the maximum in-cylinder pressure was estimated to be 31.3 bar. Increasing the heat transfer coefficient factor led to lower maximum pressure as well as to higher loos angle. Secondly, the impact of blow-by gap on in-cylinder processes was studied. For that study a heat transfer coefficient factor of 1 was chosen. Blow-by gap was varied from 0 to 0.06 mm. The results revealed that increasing the blow-by gap led to significant reduction of maximum in-cylinder pressure due to cylinder discharge during the compression stroke. It also lowering the in-cylinder pressure below atmospheric pressure during the expansion stroke. In case of 0.06 mm gap, the maximum pressure was observed to be 10.1 bar and the loss angle was estimated to be 8.5 deg. This higher blow-by mass flow led also to important cylinder charge at the beginning of exhaust valve opening. No significant impact was observed on an intake valve mass flow at different blow-by gaps.

The blow-by mass flow will be further used at 3D CFD model of the same single cylinder engine operating in the same set-up in Miller cycle.

Acknowledgment

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