

CFD Analyses on 2-Stroke High Speed Diesel Engines

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ABSTRACT

In recent years, interest has been growing in the 2-Stroke Diesel cycle, coupled to high speed engines. One of the most promising applications is on light aircraft piston engines, typically designed to provide a top brake power of 100-200 HP with a relatively low weight. The main advantage yielded by the 2-Stroke cycle is the possibility to achieve high power density at low crankshaft speed, allowing the propeller to be directly coupled to the engine, without a reduction drive. Furthermore, Diesel combustion is a good match for supercharging and it is expected to provide a superior fuel efficiency, in comparison to S.I. engines. However, the coupling of 2-Stroke cycle and Diesel combustion on small bore, high speed engines is quite complex, requiring a suitable support from CFD simulation. In this paper, a customized version of the KIVA-3v code (a CFD program for multidimensional analyses) has been used to address ports and combustion chamber design of a new project (a 3-cylinder 1.8L engine, with a power rating up to 150 HP). Multidimensional calculations have been supported by 1D engine cycle analyses, using GT-Power.

Two types of combustion-scavenging system have been considered, both of them featuring direct injection: a configuration with exhaust poppet valves and another one with piston controlled ports. A development of both projects has been performed through a coupled 1d-3d computational approach.

A first set of KIVA calculations has been performed, in order to characterize the scavenging and the port flow patterns of

both configurations, considering three different operating conditions, representative an aircraft engine.

Then, several combustion simulations have been run, for defining two chambers able to match the project goals (high fuel efficiency, limited in-cylinder peak-pressure). For the two best configurations, the most interesting calculation results are presented in the paper.

INTRODUCTION

The most difficult challenge for modern high speed Diesel engines is the limitation of pollutant emissions without penalizing performance, overall dimensions and production costs, the latter being already higher than those of the correspondent S.I. engines. An interesting concept in order to meet the opposite requirements mentioned above is the 2-Stroke cycle combined with Compression Ignition. Such a concept is widely applied to large bore engines, on stationary or marine power-plants, where the advantages in terms of power density and fuel conversion efficiency are well known.

In the automotive field, since the beginning of the '90's, some promising prototypes have been proposed by Toyota [1], AVL[2], Yamaha[3], Daihatsu [4], but, at the moment of writing this paper, none of them has reached the market. Conversely, in the field of light aircraft with piston engines, the 2-Stroke Diesel power-plants are beginning to compete with conventional SI naturally aspirated units. On the other hand, the application of the 2-Stroke Diesel concept to aircraft engines is everything but a novelty: as just an example, Junkers built a very successful series of these engines in the late '30's, named "JUMO". The main advantage offered by such a type of engine is fuel efficiency:

even in 1938, the JUMO engine was capable of a Brake Specific Fuel Consumption of 213 g/kWh [5], an impressive figure also for modern standards. It should be noticed that fuel consumption is very important for aircraft performance, since a relevant portion of the aircraft total weight (sometimes up to 50%) is due to fuel storage. The main reason of the outstanding fuel economy of 2-Stroke Diesel engines is the high mechanical efficiency ensuing the 2-Stroke cycle: besides the possibility of having no poppet valves and the relative driving system, mechanical friction losses over the cycle are about halved, in comparison to a four-stroke engine having the same crank and piston and crankcase design, thanks to the double cycle frequency. Furthermore, the two-stroke cycle is a good match for aircraft engines, since it is possible to achieve high power density at low crankshaft speed, allowing direct coupling to a propeller without the need for a reduction drive (which is heavy and expensive, besides adsorbing energy).

Supercharging further improves power density and fuel efficiency, as well as enhancing altitude performance. Diesel combustion allows a higher boosting level, in comparison to Spark Ignited engines, limited by knocking. In addition, high octane aviation gasoline will shortly be subject to strong limitations, due to its exhaust emissions of lead, while a Diesel engine can burn a variety of fuels: besides automotive Diesel, also turbine fuels such as JP4, JP5, Jet A and kerosene. Further advantages in comparison to gasoline power-plants are: reduced fire and explosion hazard, better in-flight reliability (no mixture control problems), no carburetor icing problems and safe cabin heating from exhaust stacks (less danger of Carbon Monoxide intoxication).

The recent development of Diesel technology, along with the above mentioned series of advantages, has made the 2-stroke, compression ignition engine an interesting option for light aircraft manufacturers, seeking power units of 100-300 HP, preferably not heavier than existing SI powerplants. Conversion of automotive four stroke units is generally not attractive, since they are relatively heavy: thus, design must be carried out from scratch, in order to achieve an acceptable power to weight target. The mission is not impossible: as an example, already in the second half of the '90's, AVL developed a 1-liter 2-Stroke turbocharged Diesel engine, with uni-flow scavenging, achieving a brake power of 50 kW, and a weight less than 80 kg [2], and Wilksch Airmotive brought to the experimental aircraft market a 90kW three cylinder two-stroke units using IDI combustion and weighing only 100kg [6].

The aircraft Diesel engine market is in its infancy: several prototypes have been built but none have achieved type certification at the time of writing [6, 7, 8, 9, 10]. Selected cylinder configurations have included loop and uniflow scavenging as well as opposed piston uniflow. For the sake of

brevity, this paper is focused on the two most widespread scavenging designs: uniflow with exhaust poppet valves (referred to as UNIFLOW, in the following), and loop scavenging with piston controlled ports (referred to as LOOP, in the following). The adoption of loop scavenging instead of uniflow offers lower mechanical losses and a more compact and lighter design (no valves and camshaft), as well as a better cylinder permeability. The drawbacks are the necessity of a brand new design for the combustion system, in order to comply with a completely different in-cylinder flow field (no Swirl), and a much more difficult control of the scavenging features.

The paper is focused on design of the combustion and the scavenging system of both LOOP and UNIFLOW DI Diesel engines. Specific reference is made to the project of a 3-cylinder, turbocharged engine, whose general features are listed in [table 1](#). This engine has the same cylinder and crank dimensions of a commercial IDI power-plant (WAM 100/120, see reference [6]), and it has been developed by means of engine cycle simulations, reported in a parallel paper [11].

Table 1. Reference engine main features

Engine type	2-Stroke, 3-cylinder in-line
Combustion	Diesel, Direct Injection
Scavenging type	Uniflow / Loop
Air Metering	Turbocharger + Roots
Fuel Metering	Common Rail
Injector nozzle type	Multi-hole
Displaced volume	1832 cc
Stroke	90.5 mm
Bore	95.0 mm
Connecting Rod	167.0 mm
Compression ratio	17.5:1
Maximum Brake Power	150 HP @ 2600 rpm

All the calculations presented in the paper have been carried out by using a customized version of the KIVA-3V code, that will be described in the following section. These simulations are of two different types, as described below.

- Scavenging analyses: the computational domain is made up of cylinder, piston and ports; calculations start just before exhaust port/valve opening and stop at 30° BTDC; initial and boundary conditions are provided by previous GT-Power calculations. The boundary conditions are entered as pressure profiles and constant temperatures at the inlet and outlet sections. Pure Oxygen and pure Nitrogen are used to represent fresh charge and residuals, respectively. When GT-Power and KIVA results in terms of exchanged mass do not match, the GT-Power model is modified accordingly and the KIVA simulation is repeated with new initial/boundary

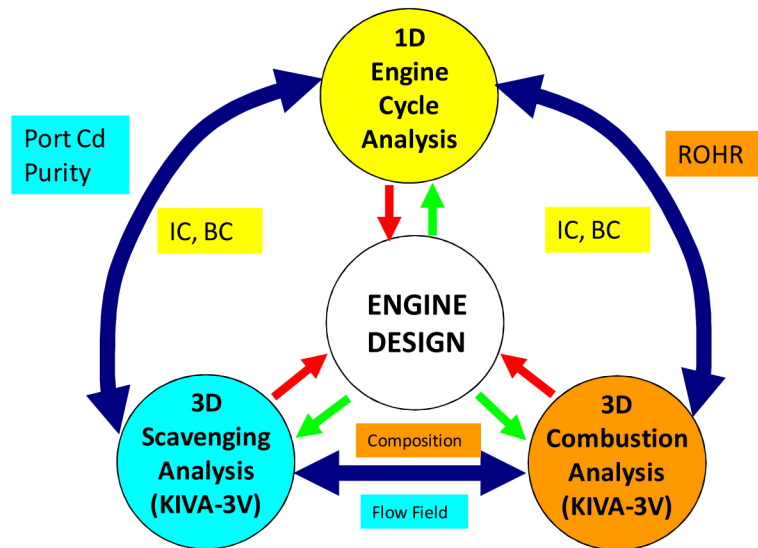


Figure 1. schematic of the engine design process

conditions. Three iterations are generally sufficient to guarantee a good agreement between 1D and 3D results.

- Combustion analyses: the computational domain is made up only of the combustion chamber; calculations start at 30° BTDC and stop at Exhaust valve/port opening; the initial velocity field has been imported from the previous KIVA scavenging simulation, while initial pressure, temperature, gas composition and the amount of fuel is defined through a correspondent GT-Power calibrated model. The injection rate is obtained from a specific model, using LMS Imagine.Lab AMESim, Rev. 8B [12].

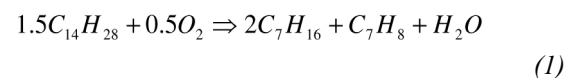
For the sake of brevity, the scavenging and combustion analyses are presented only for the final configurations, which have been defined through a computer aided design process, as summarized in the schematic of figure 1. Brake engine performance is calculated by means of 1D engine cycle analysis, with information provided by 3D-CFD detailed simulations. In particular, GT-Power requires, for each operating conditions, the burn rates, the ports/valves discharge coefficients and a definition of scavenge process (exhaust port purity as a function of the amount of residuals concentration within the cylinder). On the other hand, KIVA needs the initial/boundary conditions from GT-Power, hence an accurate analysis requires several iterations.

KIVA-3V

In this study, a customized version of the KIVA-3V code has been used [13]. The code solves the conservation equations for evaporating fuel sprays coupled with the 3D turbulent fluid dynamics of compressible, multi-component, reactive gases in engine cylinder with arbitrary shaped piston geometry. Customization of spray and combustion modeling includes detailed combustion chemistry approach, coupled with a comprehensive oxidation mechanism of diesel oil surrogate [14], turbulence / chemistry interaction model in

terms of PaSR, Partially Stirred Reactor, model [15] and a set of spray models [16] with reduced grid dependency. The fuel library has been updated by integrating thermal properties of real diesel oil. The SENKIN code of the Chemkin-2 package [17] has been used to construct and validate the chemical mechanism of combustion. Last, but not the least, the customized KIVA code has been already applied to the analysis of some current production Diesel engines and the results of calculations was found in very good agreement with experiments, as reported in [18], [19] and [20].

In order to represent real diesel oil properties, a Diesel Oil Surrogate (DOS) model was implemented, simulating the multi-component nature of the fuel vapor, regardless that the fuel is assumed to be the one-component substance. Since the properties of real diesel oil are compiled in the DI model of the KIVA-3V fuel library, this information was used in the modeling, and only the specified chemical formula of the fuel was selected as $C_{14}H_{28}$. However, since the oxidation paths for real diesel oil are not known, fuel is assumed to be decomposed into the constituent components; n-heptane (C_7H_{16}) and toluene (C_7H_8), through a global pyrolysis reaction:



Then, the detailed chemical oxidation sub-mechanisms for the constituent components were constructed. The complete chemical mechanism consists of 70 species participating in 310 elementary, global and surface kinetic stages. The aliphatic component, n-heptane, was chosen to be in excess, since its Cetane number is 56, which is similar to the Cetane number of real diesel oil. Experimental auto-ignition data for toluene at Diesel relevant conditions were reported [21].

As illustrated in Figures 2 and 3, the modified model predicts the ignition properties of the constituent components in a reasonable agreement with experimental data. The kinetic simulations were performed using the SENKIN code for a constant volume, adiabatic case. At first, Figure 2 illustrates the results of comparison for low pressures, while Figure 3 for higher pressure. In addition, it may be observed that at low and intermediate temperatures, toluene is much less reactive than n-heptane and that the aromatic component does not show the negative temperature coefficient dependence, NTC.

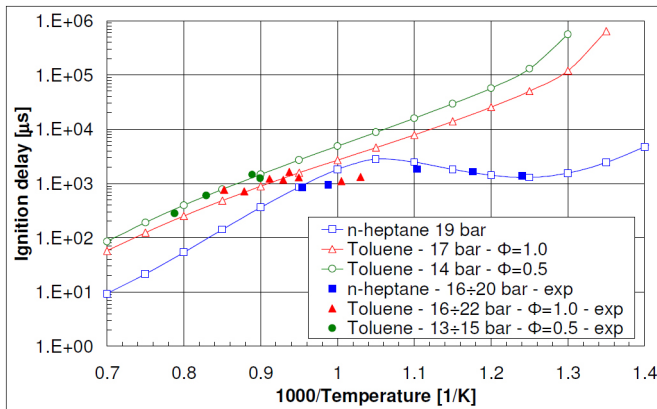


Figure 2. Calculated ignition delays for stoichiometric and lean mixtures for constituent components of DOS compared with the shock-tube data for low pressures

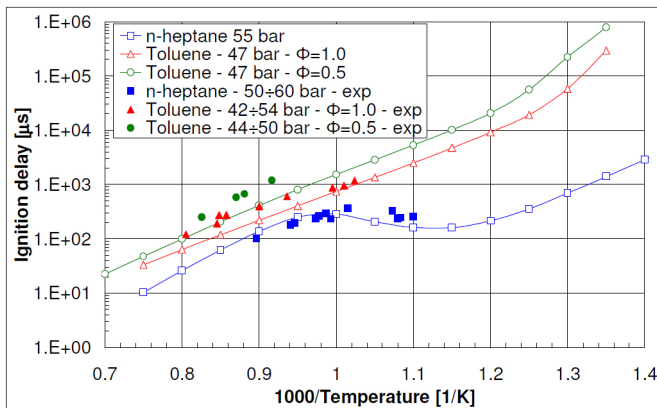


Figure 3. Calculated ignition delays for stoichiometric and lean mixtures for constituent components of DOS compared with the shock-tube data for high pressures

The auto-ignition quality of diesel oil surrogate was predicted close to n-heptane as can be seen in Figure 4. For DOS model the NTC features are predicted much weaker than for n-heptane; DOS ignition delay times for high pressure and low temperatures were predicted shorter than those for n-heptane due to the introduction of the global reaction of oxidative pyrolysis (1). For low pressures and temperatures DOS ignition delays became longer than those for n-heptane. This

phenomenon is in compliance with experimental data on diesel and n-heptane spray auto-ignition in a constant volume vessel [21]. The presence of the slowly oxidizing toluene in the surrogate fuel does not considerably affect the auto-ignition time. The reason is that the radicals formed in the oxidation of n-heptane, accelerate the co-oxidation process of toluene.

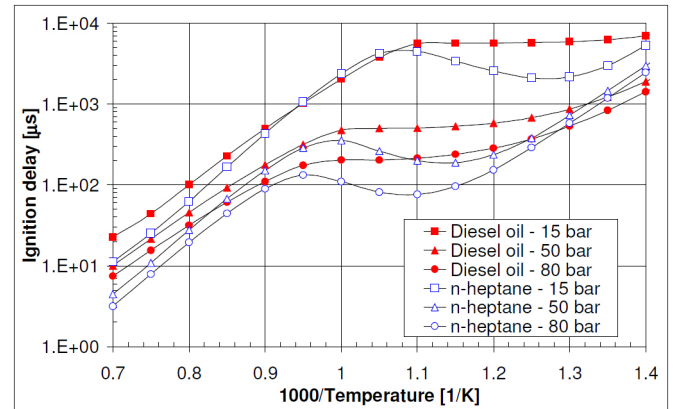


Figure 4. Calculated ignition delays for stoichiometric mixtures for DOS and n-heptane at different pressures

SCAVENGING SIMULATIONS

Particular care has been devoted to design the ports in the loop configuration, this layout being more challenging than uniflow, at least from a fluid-dynamic point of view.

The main goal of the scavenging system is to minimize mixing between fresh charge and exhaust gas. This can be achieved by generating within the cylinder a loop-shaped stream, so that the fresh air pushes out the spent charge like a fluid piston. The key is the geometry of the ports. In the optimized configuration, represented in figure 5, the intake system is made up of 2 manifolds wrapped around the cylinder and 8 intake ports. The shape of the manifolds is dictated by the need to reduce the cylinders inter-axle. It should be observed that all the inlet ports are oriented toward one focal point within the cylinder, at the opposite side of the exhaust ports, as suggested also by Blair [22]. The ports are attached to the manifold through short ducts, which have the task of driving the flow towards the cylinder head, minimizing short-circuiting to the exhaust port. These ducts have the shape sketched in figure 6: the most important parameter was found to be the upsweep angle. As this angle increases, more fresh charge remains trapped within the cylinder, but port permeability worsens. The best trade-off depends on a number of specific design issues, so that no general rule can be given. In this project, good results have been obtained with an angle of 45° for all the ports. As far as the exhaust ports are concerned, the bottom wall is inclined at an angle of 30° (see figure 7): it was found that this shape helps to reduce flow losses across the exhaust port, especially

Table 2. Operating conditions for the scavenging analysis of the LOOP engine

Engine Speed	rpm	3000	2500	2000
Volumetric Efficiency, Air	fraction	1.45	1.43	1.36
Intake Port Pressure, Average	bar	1.34	1.25	1.24
Intake Port Temperature, Average	K	343	333	333
Exhaust Port Pressure, Average	bar	1.21	1.16	1.09
Exhaust Port Temperature, Average	K	764	748	710
In-cylinder pressure at EPO	bar	7.08	6.66	5.71
In-cylinder temperature at EPO	K	1495	1471	1435

Table 3. Operating conditions for the scavenging analysis of the UNIFLOW engine

Engine Speed	RPM	3000	2500	2000
Volumetric Efficiency, Air	fraction	1.56	1.55	1.52
Intake Port Pressure, Average	bar	1.62	1.41	1.29
Intake Port Temperature, Average	K	374	349	338
Exhaust Port Pressure, Average	bar	1.58	1.40	1.26
Exhaust Port Temperature, Average	K	885	840	787
In-cylinder pressure at EVO	bar	11.93	11.25	10.46
In-cylinder temperature at EVO	K	1619	1601	1577

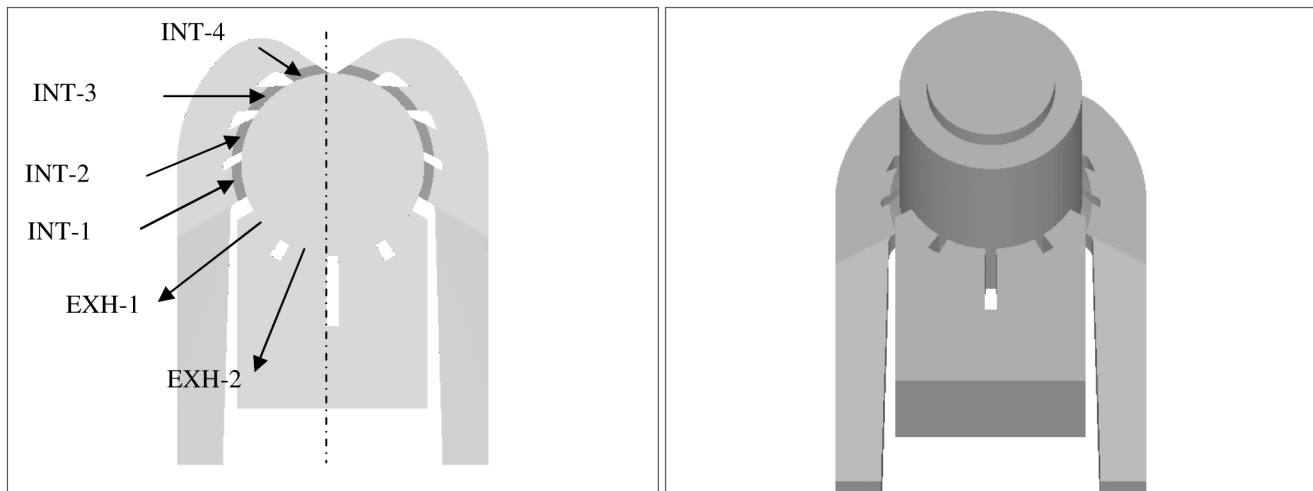


Figure 5. Schematic of intake and exhaust ports in the Loop configuration

around BDC (within the cylinder the streamlines are almost tangential to the exhaust ports, so that an inclination reduces the angle at which the flow must turn to exit)

As far as the Uniflow configuration is concerned, reference was made to the above mentioned WAM engine. No further optimization has been attempted, being the performance of the system already satisfactory.

Scavenging simulations have been carried out considering real engine operating conditions, calculated by GT-Power, as reviewed in [table 2](#) and [3](#)

RESULTS

One of the most important output of the 3D-CFD calculations is the assessment of the scavenging process quality, i.e. the dependence of Trapping, Charging and Scavenging Efficiency and Exhaust Gas Purity on Delivery Ratio. The Trapping Efficiency (TE) is defined as the ratio of the mass of fresh air retained to the mass of fresh air delivered; the

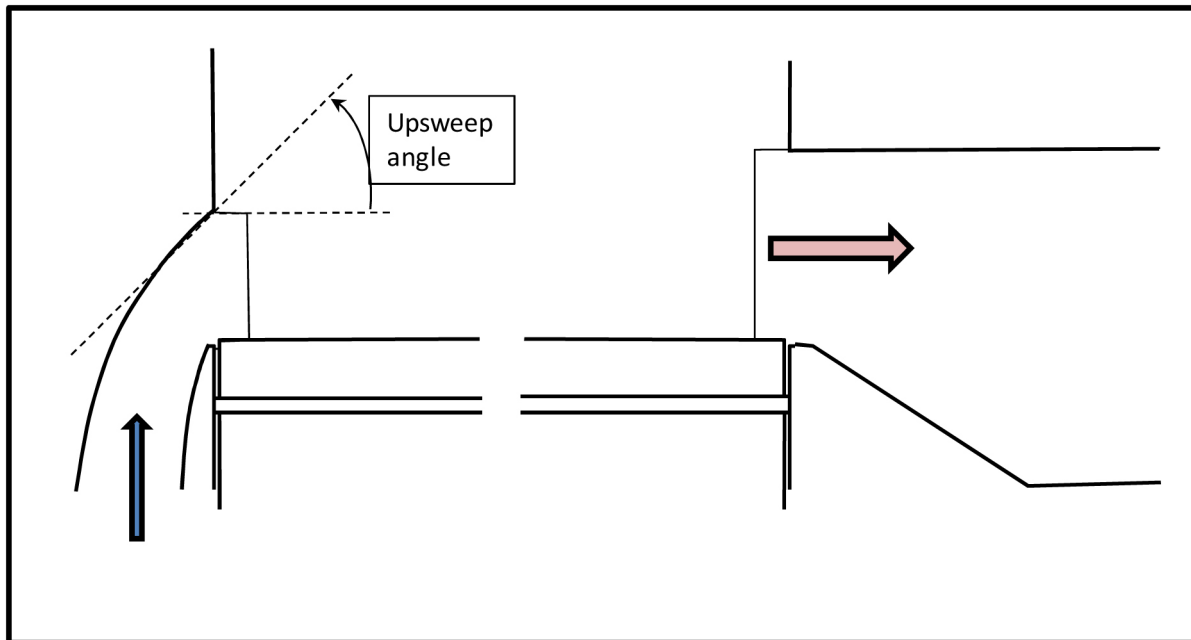


Figure 6. sketch of intake and exhaust ports in the Loop configuration

Charging Efficiency (CE) is the ratio of the mass of fresh charge retained to the reference mass; the Scavenging Efficiency (SE) is the ratio of the mass of fresh charge retained to the mass of cylinder charge; Exhaust Gas Purity is the mass fraction of fresh charge in the exhaust flow leaving the cylinder; the Delivery Ratio (DR) is the ratio of the mass of fresh charge delivered to the reference mass; finally, the reference mass is calculated considering the average delivery density and the total displaced volume.

All the parameters defined above are reported in [figures 7 and 8](#), including a comparison between Uniflow and Loop at the operating conditions defined in [tables 2 and 3](#). [Figures 9 and 10](#) show a pictorial view of the fresh charge concentration and of the gas velocities, at a several crank angles: 10° and 20° after Inlet port opening, Bottom Dead Center, 10° before transfer port closing and 5° before exhaust port/valve closing. It must be noted that the geometry of the cylinders presented in [Figures 9 and 10](#) is slightly simplified, since the project details are confidential. However, from a qualitative point of view, results are almost coincident with the real ones.

Analyzing [figures 7 and 8](#), it is observed that operating conditions affect Uniflow scavenging very slightly, while the influence is more evident on Loop. It should be considered that these conditions are defined not only by speed, but also by the pressure traces forced at both inlet and outlet, which are obviously different from case to case for representing real engine operations. The lower data scattering of the Uniflow configuration may be mainly explained by the smoother pressure traces.

The relevance of the unsteady boundary conditions can be observed in [figure 10](#), looking at the pictures at the bottom of the page: it is evident that a back-flow occurs before exhaust port/valve closing, in both Uniflow and Loop. This is a typical effect of the three-cylinder lay-out. The pressure trace in the exhaust manifold is made up of a sequence of three pulses, one for each combustion, distributed at a distance of 120°: therefore, before exhaust port/valve closing, the cylinder outflow is blocked by the pulse generated by a neighboring cylinder. This is very helpful to increase trapping efficiency, particularly in the Loop engine, when there is a long delay between exhaust and inlet port closing.

As expected, scavenging is more efficient on Uniflow: here, the process can be approximated to a perfect displacement for a DR up to 0.6; after that, some fresh charge leaves the cylinder mixed with exhaust (see the purity graph, [figure 8](#)). This mixing occurs when the stream of fresh charge climbing along the liner wall reaches the cylinder top, as typical for uniflow engines: [figure 9](#) shows this process clearly. For values of DR from 0.6 to 0.8, even if some air is lost through the valves, TE remains very high because of the relatively low exhaust mass flow rate, at least in comparison to the blow-down phase. For the maximum values of DR (1.1), TE is well over 80%, with a corresponding CE of 90%. The SE graph of [figure 8](#) indicates that, even at the maximum DR of 1.1, about a 20% of residuals remains within the cylinder. The presence of swirl (see [figure 10](#)) affects SE, increasing the mixing between fresh charge and residuals in the cylinder bulk volume. This negative effect can be balanced by a higher degree of boost, which reduces the amount of burned gas by increasing DR.