Chapter 5

Advances in The Design of Two-Stroke, High Speed, Compression Ignition Engines

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Additional information is available at the end of the chapter

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1. Introduction

The most difficult challenge for modern 4-Stroke high speed Diesel engines is the limitation of pollutant emissions without penalizing performance, overall dimensions and production costs, the last ones being already higher than those of the correspondent S.I. engines.

An interesting concept in order to meet the conflicting requirements mentioned above is the 2-Stroke cycle combined to Compression Ignition. Such a concept is widely applied to large bore engines, on steady or naval power-plants, where the advantages versus the 4-Stroke cycle in terms of power density and fuel conversion efficiency (in some cases higher than 50% [1]) are well known. In fact, the double cycle frequency allows the designer to either downsize (i.e. reduce the displacement, for a given power target) or "down-speed" (i.e. reduce engine speed, for a given power target) the 2-stroke engine. Furthermore, mechanical efficiency can be strongly improved, for 2 reasons: i) the gas exchange process can be completed with piston controlled ports, without the losses associated to a valve-train; ii) the mechanical power lost in one cycle is about halved, in comparison to a 4-Stroke engine of same design and size, while the indicated power can be the same: as a result, the weight of mechanical losses is lower.

Unfortunately, the 2-Stroke technology used on steady or naval power-plants cannot be simply "scaled" on small bore engines, for a number of reasons. First of all, the increase of engine speed makes combustion completely different, in particular for what concerns the ignition delay; second, small Diesel engines are generally designed according to different targets and constraints (for instance, they have to be efficient and clean on a wider set of operating conditions, they must comply with specific emissions regulations, et cetera); third, most of the engine components (such as bearings, connecting rods, piston rings, et cetera) are generally different, at least from a structural point of view. As a result, a brand new engine design is mandatory to develop a successful 2-Stroke high speed CI engine.

The most interesting attempts in the automobile field started at the beginning of the '90. Besides the studies at the Queen's University of Belfast [2], one of the first relevant examples is the prototype developed by Toyota [3], converting a commercial 4-Stroke, 4-Cylinder 2500 cm^3 engine into a 2-Stroke unit. Such a result was achieved by using the poppet valves as scavenging ports, and by boosting the engine through a Roots compressor. In comparison to the contemporary Diesel engines, Toyota claimed an increase of both maximum power and torque equal to 25 and 40%, respectively, while halving Nitrogen Oxides emissions.

In the second half of the '90, AVL [4] developed a 980 cm^3, three cylinder in-line prototype following a different path. The engine features an uniflow scavenging, obtained by means of inlet ports on the cylinder wall and exhaust poppet valves on head. The combustion chamber is based on a traditional HSDI four stroke design (i.e. bowl in the piston), fuel metering is provided by a Common Rail system while air boosting is obtained by a mechanical supercharging combined with a turbocharger. Combustion is assisted by a strong swirl motion whose strength can be set up by means of a proper design of the inlet ports. In the more advanced configuration, the engine shows a power density of 50 kW/l, a minimum specific fuel consumption of 235 g/kWh, along with relatively low in-cylinder peak pressures (120 bar). AVL claims that the engine is much lighter than a four stroke unit of the same top power and with similar single cylinder displacement (the total weight is less than 80kg). As far as emissions are concerned, the behavior of this two stroke engine does not differ from a four-stroke counterpart, and additional advantages have been found in terms of noise and NOx reduction.

The 2-Stroke High Speed Diesel engine concept was investigated in 1999 also by Yamaha, who built a 1000 cm^3, 2-Cylinder engine, with crank-case loop scavenging [5]. The most peculiar issue of this prototype is the combustion system, made up of a pre-chamber, connected to the cylinder through four holes. During compression, these holes impart a swirling motion to the charge entering the pre-chamber, while, during expansion, they allow the gas to expand in the cylinder, with limited flow losses, in comparison to traditional indirect Diesel engines. Even if power output was not particularly high (33 kW@4000rpm), this engine featured compact dimensions, along with very low fuel consumption and engine-out emissions, at least in in comparison to the contemporary 4-Stroke engines.

In 2005, Daihatsu [6] announced a 2-cylinder, 1200 cm^3 of capacity automotive engine, exhibiting a maximum power of 65kW and a maximum torque of 230N.m. Daihatsu claimed that the prototype was very fuel efficient and clean, being able to comply with EURO V regulations. The scavenging and the air metering system are like the ones previously mentioned about the AVL prototype, with particular care devoted to reduce the mechanical loss of the supercharger, as well as to generate a moderate swirling motion within the chamber. The engine featured a cooled EGR device and the latest Common Rail injection system.

Still in 2005, FEV announced the development of a four cylinder supercharged 2-Stroke Diesel engine, for military ground vehicles [7]. This engine, called OPOC (opposed-piston, opposed-cylinder), features uniflow scavenging (intake and exhaust ports at opposite ends of the cylinder), asymmetric port timing (exhaust ports open and close before intake) and electrically-assisted boosting. FEV claims a very high power to weight ratio (325HP, 125kg) and low fuel consumption.

A 2-Stroke high speed engine concept has been developed also by the University of Modena and Reggio Emilia [8]. The core of the project is a brand new type of combustion system. As well known, conventional DI Diesel engines (both Two and Four Stroke) adopt a bowl in the piston, whose shape is optimized in order to generate an optimum mean and turbulent flow field around TDC, provided that a proper swirl motion is imparted to the intake flow. Conversely, in the new combustion system the combustion chamber is carved within the engine head, while the piston crown is flat. Furthermore, for the sake of compactness and cost, scavenging is obtained without poppet valves, but using piston controlled slots at the bottom of the cylinder liner. Since this scavenging is of the loop type, the combustion chamber and the injection system are designed in order to comply with a flow field characterized by a strong tumble vortex at exhaust port closing, that is going to destroy itself just before top dead center. The new combustion system is expected to yield some advantages, in comparison to the prototypes characterized by uniflow scavenging with on-head exhaust poppet valves, and bowl in the piston. First, on-head exhaust valves are not used, with ensuing advantages in terms of overall compactness, cost, reliability, weight and friction losses. Second, the piston becomes simpler and lighter, while its thermal load is dramatically reduced. Third, heat transfer during expansion is strongly reduced because of the absence of swirl: as a result, heat losses are less.

While in the automobile field the 2-Stroke Diesel engine still hasn’t found an application to industrial production, beside some exceptions (in 1999 Daihatsu proposed the “Sirion” car with a 3-cylinder 2-Stroke 1.0L engine), this concept is starting to be applied in the aeronautic field, to power light aircrafts [9-14].

The application of the 2-Stroke Diesel concept to aircraft engines is everything but a novelty: as just one example, Junkers built a very successful series of these engines in the late 19-’30’s, named “JUMO”. The main advantage offered by such an engine, in comparison to the contemporary piston engines was fuel efficiency: in 1938, the JUMO engine was capable of a Brake Specific Fuel Consumption of 213 g/kWh [15], an impressive figure even by 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.

In addition to the advantages already mentioned, 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 the 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 is expected to be subject to strong limitations, due to its polluting emissions of lead, while a Diesel engine can burn a variety of fuels: besides automotive Diesel, also turbine fuels such as JP4 and JP5, and Jet A. 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).

2. Design options

As it can be deduced by the previous section, there are several options that can be explored in the design of 2-Stroke CI high speed engines. The most typical ones are listed below.

1. Uniflow scavenging with exhaust poppet valves and piston controlled inlet ports; external blower and 4-Stroke- like oil sump; direct injection, bowl in the piston.

2. Uniflow scavenging with exhaust poppet valves and piston controlled inlet ports; external blower and 4-Stroke- like oil sump; indirect injection with a pre-chamber connected to the cylinder through one or more orifices.

3. Loop Scavenging with piston controlled transfer and exhaust ports; crankcase pump; indirect injection with a pre-chamber connected to the cylinder through one or more orifices

4. Uniflow scavenging with opposed pistons, twin crankshafts; external blower and oil sump; indirect injection with a pre-chamber connected to the cylinder through one or more orifices

5. Loop scavenging with inlet and exhaust poppet valves in the engine head; 4-Stroke-like crankcase and external blower; indirect injection with a pre-chamber connected to the cylinder through one or more orifices.

6. Loop scavenging with inlet and exhaust poppet valves in the engine head, 4-Stroke-like crankcase and external blower; direct injection, bowl in the piston.

7. Loop Scavenging with piston controlled transfer and exhaust ports; 4-Stroke-like crankcase and external blower; indirect injection with a pre-chamber connected to the cylinder through one or more orifices.

8. Loop Scavenging with piston controlled transfer and exhaust ports; 4-Stroke-like crankcase and external blower; direct injection with a chamber carved in the engine head.

A synthetic comparison among the configurations is given in Table 1 while Figure 1 shows the relative layouts.

FEATURES	CONFIGURATIONS							
	1	2	3	4	5	6	7	8
Scavenging quality	B+	A-	A	B+	D	D	B+	B+
Thermal efficiency	A-	C-	C-	C-	C-	A	C-	A
Mechanical efficiency	B	B	B	A-	B	B	A	A
Engineering cost	B	B	C	B+	A-	D	A-	D
Injection system cost	B	A	A	A	A	B	A	B
Overall dimensions	B	B	A	A	B	B	A	A
Power density	A	B	B	D	C	B-	B-	A

Table 1. Comparison among the different designs listed in the previous section. Grades: A=Excellent, B=Good, C=Average, D=Poor

From the scavenging quality point of view, uniflow scavenging is generally better than loop, even if the necessity of imparting a swirling motion to the inlet flow can spoil the advantage a little bit. Since the swirl requirement is more stringent for direct injection, DI Uniflow scavenging configurations generally yield lower trapping and scavenging efficiency than Uniflow IDI designs. Another advantage of the IDI design is the cost of the injection system, that can be of the mechanical type. The downsides are the low thermal efficiency and the limitation on power rating due to smoke emissions at high speed and load.

When scavenging is obtained only by means of piston controlled ports, the valve-train is absent. However, the advantage in terms of mechanical efficiency can be spoiled without a proper lubrication, or in the case of a double crankshaft (opposed piston design). Particular care must be devoted when using a crankcase pump, since some oil uniformly dispersed in the airflow is generally not sufficient at high load. On the other hand, the combination of loop scavenging and crankcase pump enables a very compact design when power rating is low.

Except for the opposed pistons configuration, the piston-controlled ports design implies that a tumble motion is generated within the cylinder. The same type of flow field can be found in the designs with inlet and exhaust poppet valves, referred to as 5 and 6. The optimization of a DI combustion system without swirl is far from trivial and it requires a strong support by simulation and specific experiments, with ensuing rise of the engineering costs. The same problem may be faced in the development of an opposed piston design, because of the lack of reference in recent projects.

In general, every solution presented in table 1 has its own pros and cons, so that the best choice depends on the project specifics. In the authors' opinion, the most balanced solutions are #1 and #8.

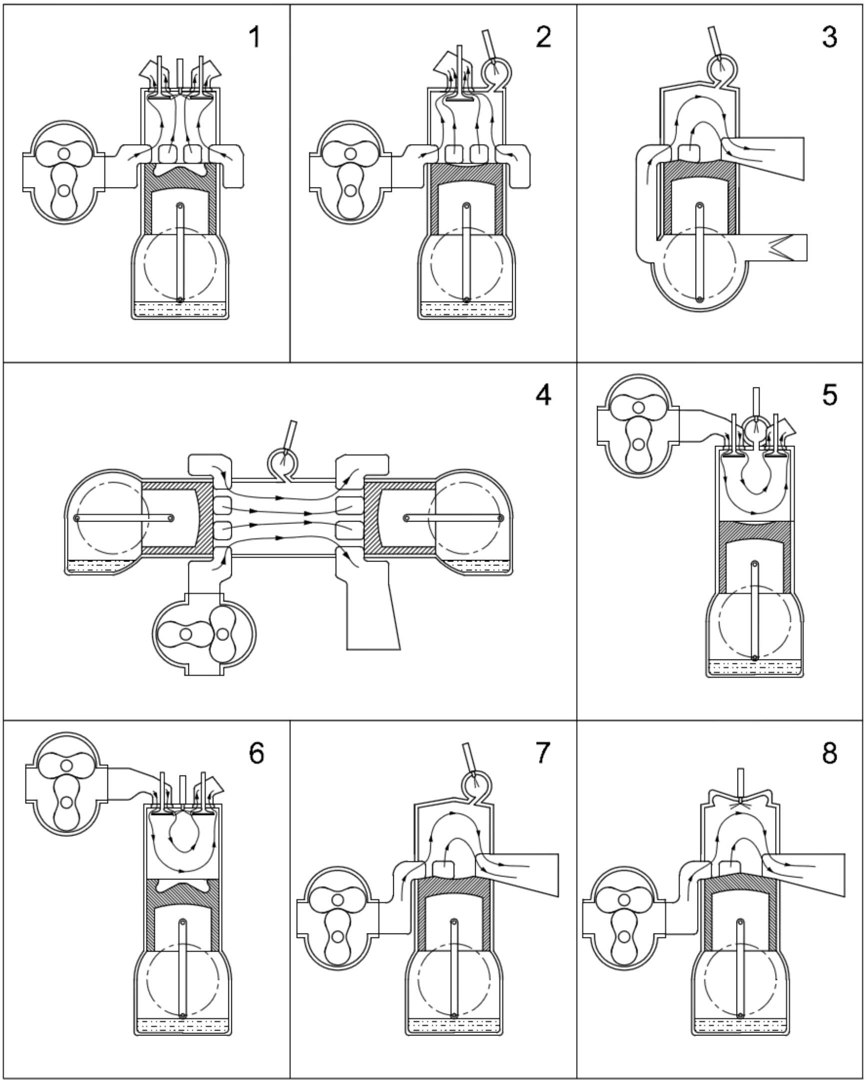

Figure 1. Typical configurations of 2-Stroke CI high speed engines

3. Scavenging systems

The optimization of the scavenging process is one of the most challenging task in the design of 2-Stroke engines. In fact, the geometry of the ports-cylinder assembly should be defined in order to guarantee a smooth path of the flow across the engine (low flow losses), while minimizing short circuiting and the mixing between fresh charge and exhaust gas. Another important issue is the conditioning of the mean in- cylinder flow field (swirl or tumble), which strongly affects both combustion and heat transfer. The optimum intensity of the swirl/tumble rates depends on the type of combustion system, as well as on the specific project targets. As an example, the swirl ratio in DI engine with a bowl in the piston should be high enough to promote the diffusion of the fuel vapor in the chamber. However, an excessive mean turbulence is detrimental to spray penetration, and heat losses increase.

The energy spent to pump the fresh charge across the cylinder is a fundamental parameter, even if not the only one, to assess the quality of the scavenging system. In order to find a simplified correlation among the average pressure drop across the cylinder (Δp) and the main engine parameters, the gas exchange process in a 2-Stroke engine can be idealized as a steady phenomenon, with the piston fixed at bottom dead center and both inlet and exhaust ports partially open, so that the geometric area of each port corresponds to the average effective area, calculated over the cycle. As a further simplification, the flow is assumed as uncompressible. According to these hypotheses, the mass flow rate across the cylinder can be expressed as:

$$A_{eff,av}\sqrt{2\rho\cdot\Delta p}=\frac{\rho\cdot DR\cdot A_p\cdot U_p}{2} \tag{1}$$

Where ϱ is the charge density, DR is the Delivery Ratio of the engine (ratio of the delivered fresh charge to the reference mass, calculated as the product of charge density to cylinder displacement), U_p is the mean piston speed. $A_{eff,av}$ is the average effective area of all the ports, that can be expressed as:

$$A_{eff,av}=\frac{1}{\sqrt{1/A_T^2+1/A_E^2}} \tag{2}$$

Being A_T the mean effective area of the transfer ports and A_E the mean effective area of the exhaust ports.

Combining equation 1 and 2, the following expression for Δp is found:

$$\Delta p\propto\rho\cdot DR^2\cdot U_P^2\cdot\left(\frac{A_p}{A_{eff,av}}\right)^2 \tag{3}$$

The following observations can be made:

1. Equation (3), despite the simplifications, is able to yield qualitative information about the engine permeability, i.e. the attitude of the ports system to throttle the flow across the cylinder.
2. The higher is the delivery ratio and the maximum mean piston speed, the more important is to have high values of effective area, in comparison to the piston area. Also the charge density plays a role, thus supercharged engines are more demanding in terms of permeability than naturally aspirated units.
3. The ports average effective area can be increased by reducing the flow losses and/or by increasing the opening area of both inlet and exhaust ports.

While permeability is related to the mean piston speed, Diesel combustion is affected by engine speed: the lower is the maximum number of revolutions per minute, the less is the need of turbulence to support air-fuel mixing.

A number of different lay-outs has been proposed in more than one century of history, and it would be quite hard to review all of them. The two most widespread designs, at least for high speed engines, are the Loop and the Uniflow configurations, the former with piston controlled ports, the latter with exhaust poppet valves, driven by a camshaft, and piston controlled inlet ports. Uniflow scavenging with opposed pistons is not considered, for the sake of brevity.

CFD simulation is the key for the design of modern scavenging systems. The numerical analyses are carried out by means of 3D tools, which are able to predict the flow field details within the cylinder and through the ports under actual engine operating conditions. Because of the computational cost, the simulation domain is limited to a single cylinder, and to the portion of cycle included between exhaust port opening and exhaust port closing. Therefore, initial and boundary conditions must be provided by another type of CFD tool, able to analyze the full engine cycle and the influence of the whole engine lay-out, even if in a simplified manner (in particular, the spatial distribution of the flow through the intake and exhaust systems is considered as one or zero dimensional). The authors have applied this methodology in a number of studies [8, 12-14, 20-25], comparing the simulation results to the experiments, whenever possible. CFD simulation was found to be a quite reliable tool, provided that the numerical models are always underpinned by some experimental evidence.

4. Loop scavenging

For loop scavenged engines, a quite successful design, applicable to both crankcase and external scavenging, is shown in figure 2 (left). As visible, there is a symmetry plane passing through the twin exhaust ports (E1, E2) and the rear transfer port (T5). The transfer ports 1-4 blow the fresh charge toward the wall opposite to the exhaust side, while the elevation angle of the rear transfer port should be higher than that of the other transfers, to prevent short circuiting. A design optimized for a SI racing engine is shown in figure 2 (right) [16]. For a Diesel engine, this design represents the limit at which to tend for achieving the maximum cylinder permeability. However, since mean piston speed is generally low, it is convenient to reduce the width of the ports (less concern for piston rings and liner durability) and avoid the overlapping between transfer and exhaust (less risk of short-circuit). When permeability is not an issue at all, a further simplification that can be done is to design just one exhaust port. The advantage is the removal of a quite critical region, from the thermal point of view, i.e. the bridge between the two exhaust ports.

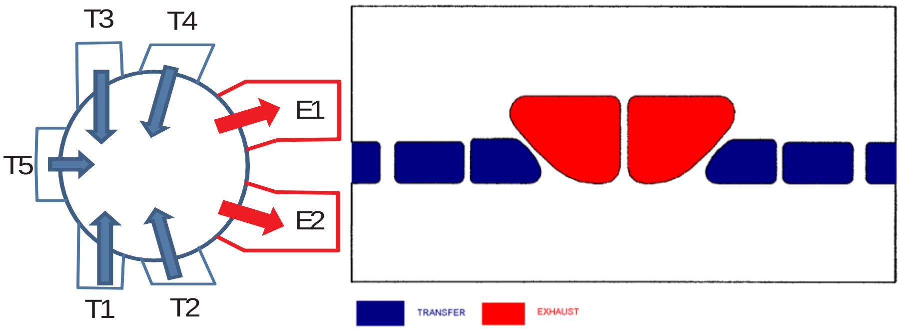

Figure 2. left) sketch of the ports in loop scavenged configuration 1 and (right) ports development of a 125 cc 2-S SI racing engine by Honda, bore x stroke: 54 x 54.5 mm, EPO/TPO: 82.0/111.6 atdc, [16]

In another loop configuration, represented in figure 3, the intake system is made up of 2 symmetric manifolds, wrapped around the cylinder, and 2 symmetric sets of 4 inlet ports. This solution is specifically designed for external scavenging. The manifolds cross section width is smaller than the height, in order to reduce the cylinders inter-axle. Furthermore, the cross section area is decreasing along the manifold axis, in order to have a more uniform distribution of the flow rate through the inlet ports. It is 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 [17]. The ports are attached to the manifold through short ducts, which have the task of driving the flow towards the cylinder head, for minimizing short-circuiting. These ducts have the shape sketched in figure 4.

In the CFD studies reported in [8] and [14], the most important design parameter for the inlet system was found to be the upsweep angle of the ports, see figure 4. As this angle increases, scavenging efficiency improves, but the port effective area is reduced. The best trade-off depends on a number of specific design issues, so that no general rule can be given. In the project described in [8], where the unit displacement of the engine was 350 cc (bore 70 mm, stroke 91 mm, maximum engine speed 4500 rpm), the best results have been obtained with an angle of 45° for all the ports.

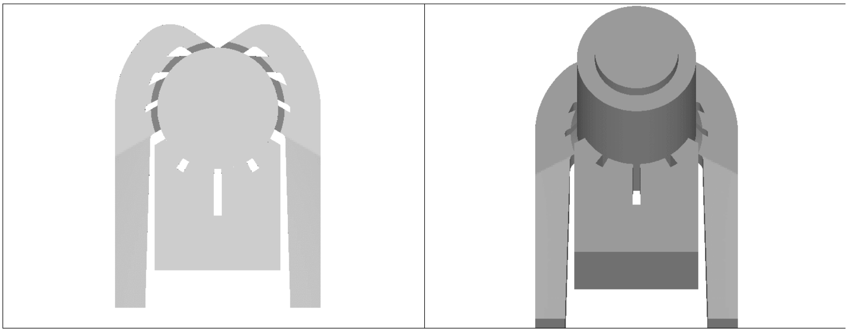

Figure 3. Schematic of intake and exhaust ports in the Loop configuration#2

As far as the exhaust ports are concerned, figure 4 shows that it is convenient to assign a downward angle to the bottom wall, in order to increase the maximum port effective area. In fact, around BDC, the streamlines within the cylinder tend to be almost tangential to the exhaust port, so that an inclination of the port bottom wall reduces the angle at which the flow must turn to exit.

It is important to notice that the permeability of a loop scavenging system is related to the choice of the bore-to-stroke ratio. In fact, since engine speed is limited by combustion constraints, the critical factor generally remains the average effective area of the ports, referred to the piston area (see equation 3). It can be easily demonstrated that a low bore-to-stroke ratio helps to have larger opening areas for both types of ports, so that it is generally convenient to have a stroke longer than bore.

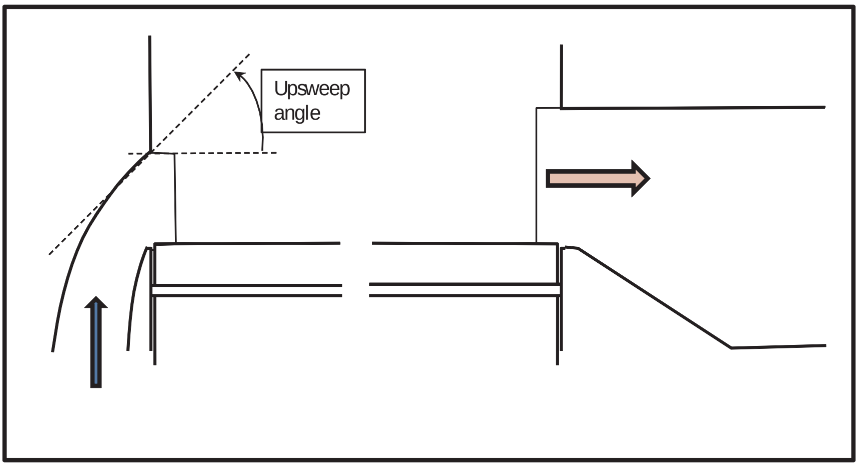

Figure 4. Sketch of intake and exhaust ports in the Loop configuration

5. Uniflow scavenging

As far as the Uniflow scavenging is concerned (piston controlled inlet ports and exhaust poppet valves on the cylinder head) some design guidelines are provided below.

EXHAUST VALVES In the Uniflow scavenging, the critical issue for permeability is the effective area of the exhaust valves. Even if the engine speed is low, strong constraints are generally placed upon the maximum lift, since the optimum opening duration is at least 30% less than a corresponding 4-Stroke engine. From this point of view, the higher is the number of valves, the better. As an example, passing from 2-valve to 4-valve, the maximum geometric flow area increases by about 30%; furthermore, the valves are smaller and lighter, so that it is possible to define in a more free manner the valve actuation law (maximum lift and duration), and provide a more effective cooling; last, but not least, the injector can be placed on the cylinder axis, without penalization on the valve dimensions. The central position of the injector is particularly important when the combustion chamber is in the piston bowl, in order to guarantee a uniform distribution of the fuel within the cylinder. Obviously, with more valves, the valvetrain is more expensive and heavy, while the flow losses may significantly increase, without a proper design of the valve ports and ducts.

INLET PORTS A comprehensive CFD study on the influence of the inlet ports geometry has been carried out by Hori [18]. A simple but effective configuration studied by this author is presented in figure 5, where a set of 12 ports uniformly distributed along the cylinder bore is shown. The ports do not need an upsweep angle, since the piston skirt is already driving the flow toward the cylinder head. At BDC, the upward direction of the flow can be imposed by leaving a small step (1-2 mm) between the piston crown and the bottom wall of the ports. It may be noticed that the axis of each port forms an angle with the radial direction. As this angle increases, the swirl ratio grows up, along with the pressure drop across the cylinder. A large angle is desirable in order to sweep the exhaust gas along the circumference of the liner, but a pocket of exhaust gas may remain in the cylinder core. Conversely, near-radial ports are less effective in the outer region, but they better sweep the cylinder bulk. In terms of scavenging efficiency (concentration of fresh charge at inlet port closing), the former solution is better, according to Hori. This outcome can be explained considering that a ring of exhaust gas trapped in the outer region of the cylinder contains more mass than a ring of similar thickness close to the cylinder axis. In order to achieve a good scavenging efficiency in combination with low swirl, Hori proposed an “alternate port” configuration, i.e. a sequence of one radial port and one swirling port, the former with an upward angle of elevation, the latter with a downward angle. Another important parameter investigated by Hori is the opening area ratio, that is the fraction of cylinder bore occupied by the ports. As this ratio increases, the flow losses goes down, along with the swirl ratio. A typical range for the opening area ratio is between 50 and 80%: the upper limit concerns problems such as durability of the piston rings and of the liner. Finally, Hori showed the importance of the chamfering radius of the ports: as this parameter increases, flow losses are diminished and the swirl ratio goes down (the portion of straight channel is lower, so that it becomes increasingly difficult to impart the direction to the flow). For a liner 10 mm thick, an optimum chamfering radius of 3 mm was suggested.

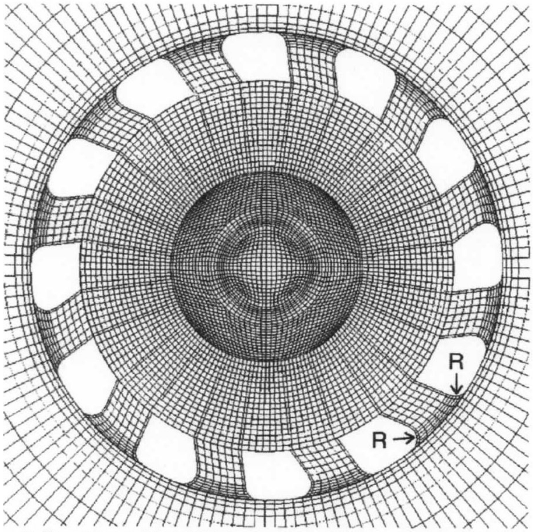

Figure 5. Computational mesh of a Uniflow design analyzed by Hori [18], showing the layout of the inlet ports

6. Uniflow vs. loop scavenging

In literature it is quite hard to find an objective comparison between a uniflow and a loop design under real operating conditions, since it is very expensive and time-consuming to build two different prototypes complying with the same constraints and targets, and developed with the same degree of technical sophistication.

As an example, in [19] a comparison was presented between a uniflow and a loop design having the same bore (80 mm) and compression ratio (19). Unfortunately, the uniflow engine featured an external blower and a stroke/bore ratio of 1.23, while the loop design was characterized by crankcase scavenging and bore/stroke ratio 0.875. In such a different conditions, the outcome of the study, i.e. the superiority of the uniflow design, is quite questionable.

In a theoretical study presented in [13] and [14], a uniflow and a loop design were developed on the same starting base, a commercial aircraft engine, named WAM 100, whose features are listed in table 2.

The new designs were developed trying to maintain as much as possible of the original engine: therefore, bore, stroke, number of cylinders, air metering system, et cetera are the same, while the rated power is set at a higher value (150 HP), thanks to the introduction of a specifically developed combustion system featuring direct injection and a Common Rail system. Since the Uniflow scavenging was already optimized in the original engine, most of the attention was paid to the loop version. Here, a ports design as the one visible in figure 3 was adopted, and optimized via CFD-3D simulations.

Engine type	2-Stroke, 3-cylinder in-line
Combustion	Diesel, Indirect Injection
Scavenging type	Uniflow
Number of Valves/Ports	2 Exh. valves/20 inlet ports
Air Metering	Turbocharger + Roots blower
Fuel Metering	In-line mechanical pump
Injector nozzle type	Single-hole (Pintle)
Displaced volume	1832 cc
Stroke	95.0 mm
Bore	90.5 mm
Connecting Rod	167.0 mm
Compression ratio	17:1
Exhaust Valves Open	83° before BDC
Exhaust Valves Close	80° after BDC
Inlet Port Open	53° before BDC
Inlet Port Close	53° after BDC
Maximum Brake Power	102 HP @ 2750 rpm

Table 2. Main features of the WAM 100 engine, assumed as a starting base for the CFD study presented in [13] and [14].

A comparison between the scavenging parameters calculated under real engine operating conditions (2000, 2500 and 3000 rpm, full load) is presented in figure 6. Figure 7 presents a pictorial view of the fresh charge concentration on a plane passing through the cylinder axis, at different crank angle. Engine speed is 2500 rpm, full load.

The scavenging parameters are defined as follows. The Trapping Efficiency (TE) is the ratio of the mass of fresh air retained within the cylinder to the mass of fresh air delivered; the Scavenging Efficiency (SE) is the ratio of the mass of fresh charge retained to the total cylinder mass (fresh+exhaust); the Exhaust Gas Purity is the mass fraction of fresh charge in the exhaust flow leaving the cylinder; finally, the reference mass is calculated considering the average delivery density and the total displaced volume.

Analyzing figures 6 and 7, 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 the inlet and the outlet boundaries, which are obviously different from case to case for representing real engine operations. The lower data scattering of the Uniflow design may be mainly explained by the more regular pressure traces.

It is also important to notice that the scavenging parameters under real operating conditions can be quite different from the ones expected when performing a steady characterization. First of all, the mass flow rates entering and leaving the cylinder are all but constant (in a properly tuned exhaust system, the flow must change its direction after transfer port closing, to reduce the loss of fresh charge); furthermore, the density of the charge entering the cylinder changes along the cycle, as well as the pressure drop across the ports. Among the dynamic effects, a very good help can be found in the 3-cylinder lay-out. In fact, 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 particularly helpful in the Loop engine, when there is a long delay between exhaust and inlet port closing.

As generally expected, scavenging is more efficient in the Uniflow design: 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). This mixing occurs when the stream of fresh charge climbing along the liner wall reaches the cylinder top, as typical for uniflow engines: figure 8 shows this process clearly. For values of DR higher than 0.6 some air is lost through the valves, but TE remains very high because of the charge stratification within the cylinder: the air concentration in the head region is always lower than in the other parts of the cylinder. As a result, at the maximum values of DR (1.1), TE is well beyond 80%. The SE graph of figure 6 indicates that, even at the maximum DR of 1.1, about a 20% of residuals remains within the cylinder. The presence of swirl 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.

Loop scavenging is reasonably good: the flow patterns remains very close to those of a perfect displacement up to a DR of 0.5, while for higher delivery rates the situation is intermediate between a perfect mixing and a perfect scavenging. TE plots are consistent with Purity trends: the drop of retaining capability corresponds to the presence of fresh charge in the exhaust outflow. Scavenging features seem to improve a little bit as engine speed decreases, but this effect may not be related only to speed, as already discussed. An advantage of Loop on Uniflow can be observed in the SE plots: Loop seems to better sweep the residuals from the cylinder, at any DR value. This effect is ascribed to the lower permeability of Uniflow, in particular of exhaust valves in comparison to ports: it is well known that a piston controlled port yields a much larger average flow area than a valve of about similar dimensions. As a consequence, in the Uniflow cylinder the residuals leaves at a slower pace, and a larger quantity remains trapped for each DR. However, the better scavenging efficiency of Loop in comparison to Uniflow is not a general result, but it strictly depends on the specific geometric details and on the valve actuation law.

From the pictorial view of figure 8, it may be observed how different is the in-cylinder flow field between Uniflow and Loop, after BDC. While in Uniflow the swirl ratio can be adjusted varying the tangential inclination of the transfer ports, a strong tumble is always observed in the Loop case. This difference is expected to have a big influence on combustion: while the swirl angular momentum decays very slowly, supporting turbulence around TDC and later, the tumble vortex is destroyed well before the start of combustion. Furthermore, the turbulent kinetic energy field at TDC depends more on the momentum transferred from fuel injection than on the in-cylinder flow patterns. Therefore, combustion in loop scavenged engines is much less sensitive to the scavenging patterns, and it must be optimized according to new concepts.

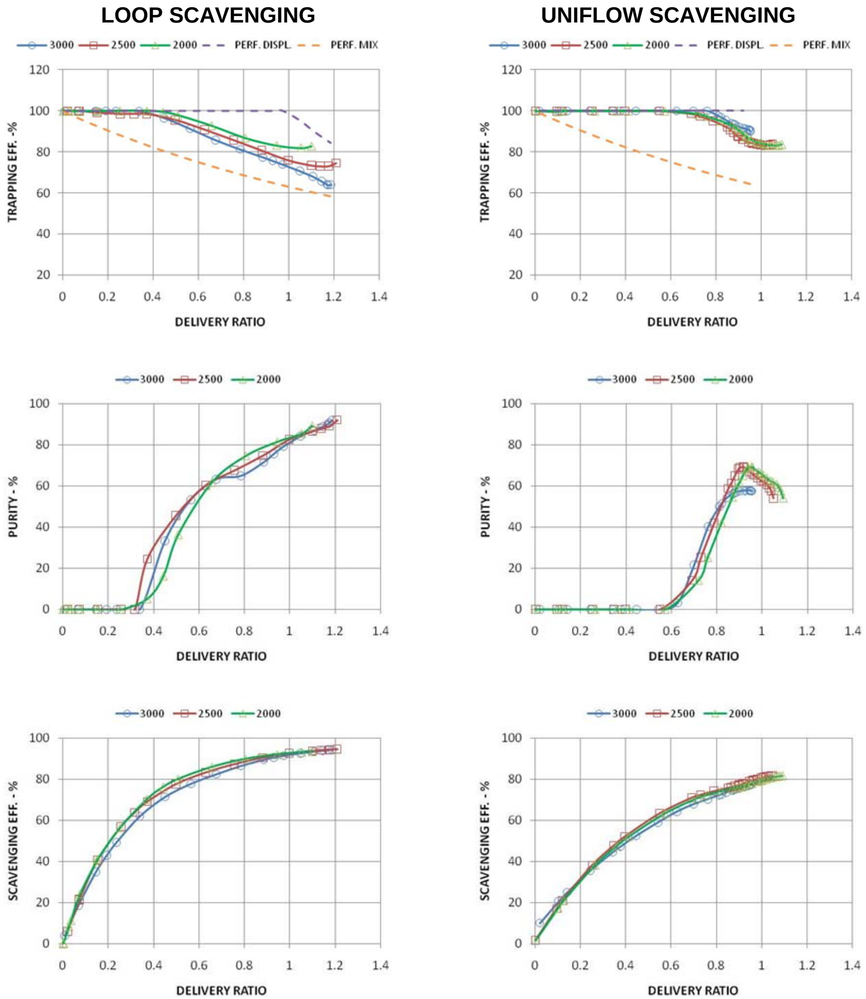

Figure 6. Trapping efficiency, Purity and Scavenging efficiency plotted as a function of Delivery Ratio. Values calculated at three different operating conditions on the optimized Loop and Uniflow configurations (engine speed: 2000, 2500 and 3000 rev/'), reference [14]

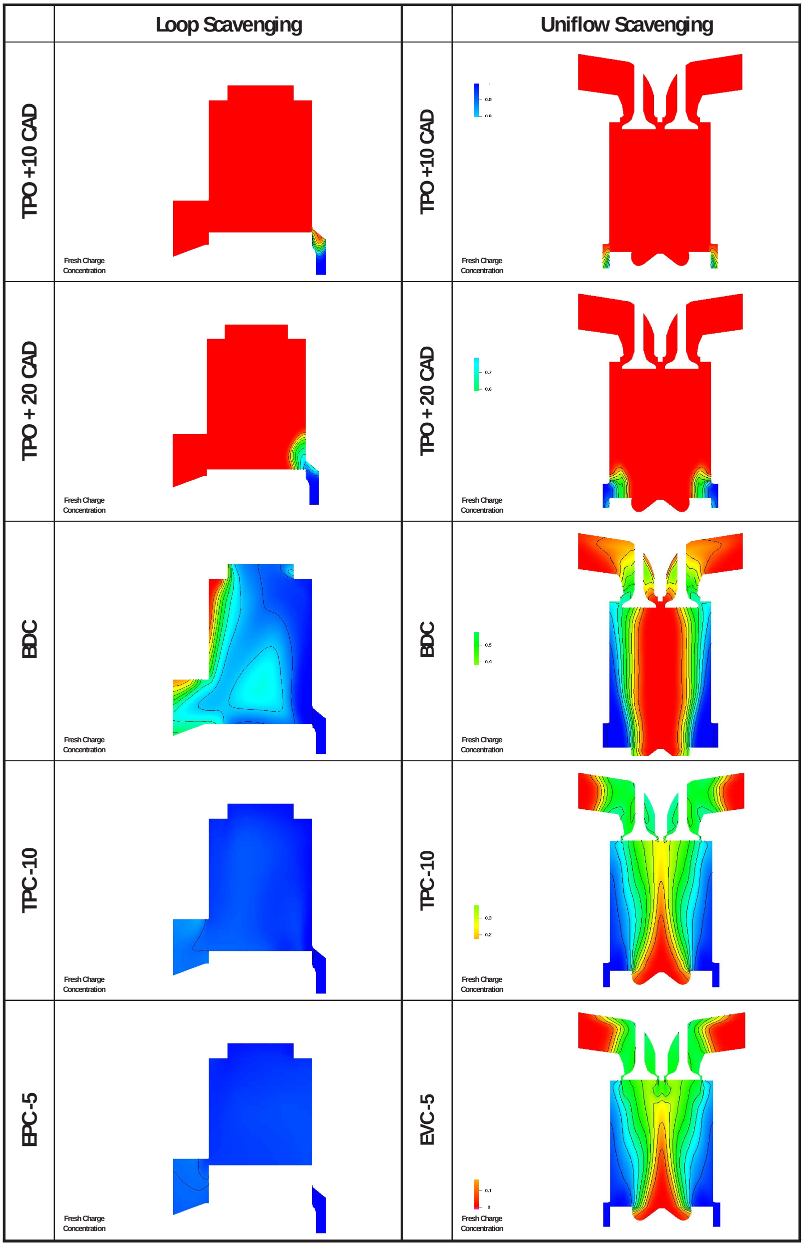

Figure 7. Fresh charge concentration plotted on a plane passing through the cylinder axis at different crank angles. Comparison between Loop and Uniflow at 2500 rpm, full load [14]

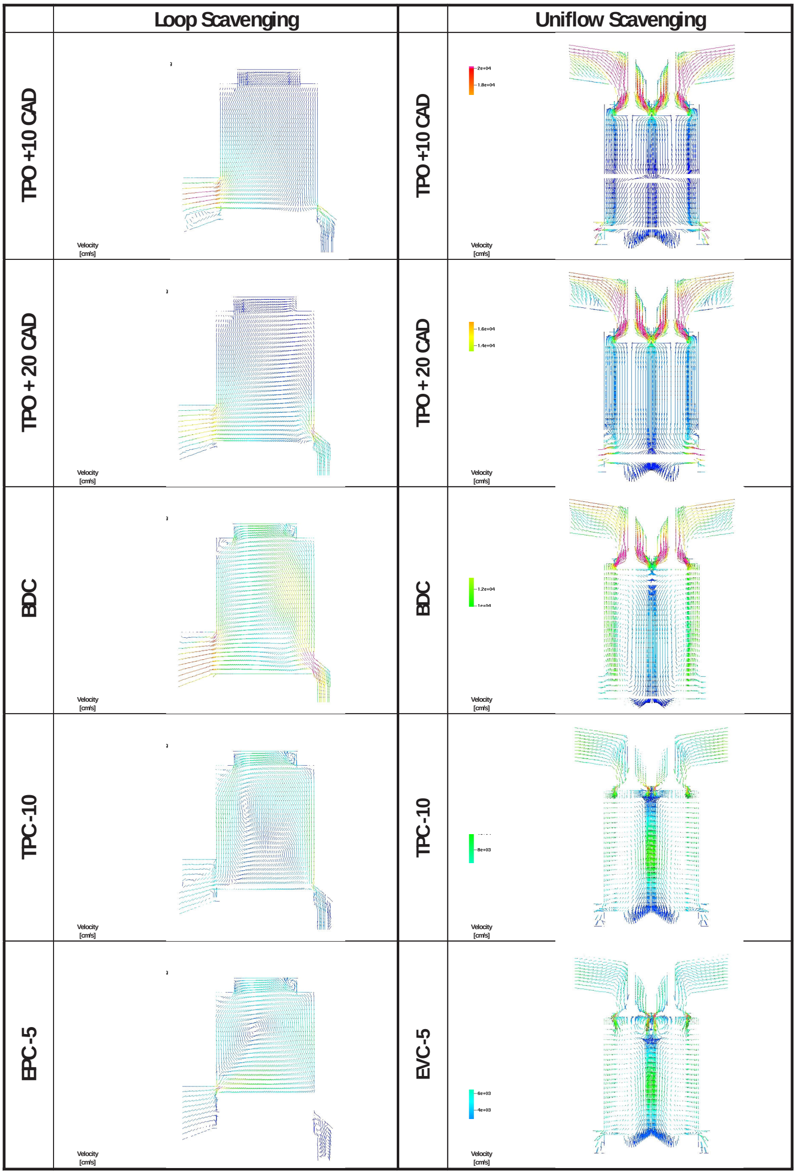

Figure 8. Velocity vectors plotted on a plane passing through the cylinder axis at different crank angles. Comparison between Loop and Uniflow designs at 2500 rpm, full load [14]

7. Combustion system design

As anticipated in the previous sections, the most difficult challenge when designing the combustion system of a 2-Stroke Diesel engine is to achieve an efficient combustion without swirl. This situation always occurs in loop scavenging, while, for the uniflow design discussed in the previous section, it is possible to import a combustion system directly from a 4-stroke project. Since a comprehensive literature already exists on the optimization of bowl in the piston chambers, this subject won't be considered, and all the attention is going to be focused on the loop scavenged engines. The lack of knowledge on this type of combustion systems for high speed Diesel engines requires an extensive work in order to optimize the wide range of design parameters.

In the two different projects reviewed in [8], [20] and [14], full theoretical investigations, supported by experimentally calibrated 1D and 3D CFD tools, have been carried out in order to address the combustion chamber design and to define the most appropriate injection system set-up. Figure 9a shows the different types of combustion chambers analyzed in the first project for an automobile engine (bore x stroke: 70 x 91 mm, compression ratio: 19.5:1, maximum engine speed: 4000 rpm; scavenging system as shown in figure 3), while figure 9b presents a design optimized for an aircraft engine (bore x stroke: 90.5 x 95 mm, compression ratio: 17:1, maximum engine speed: 2600 rpm; scavenging system as shown in figure 2)

For each configuration of figure 9a, four different speeds have been considered (1500, 2000, 3000 and 4000rpm) and calculations have been performed at full load, being this condition the more critical for the 2-Stroke engine. Combustion simulations have been carried out from EPC to EPO, while initial and boundary conditions have been set according to the results of 1D and CFD-3D scavenging simulations. Combustion simulations results are reported in Figure 10 in terms Gross Indicated Mean Effective Pressure (GIMEP). GIMEP is calculated as the integral of the pressure-volume function between Exhaust Port Closing and Exhaust Port Opening. As visible, the best solution appears to be the reverse bowl (named E) in figure 9a). This configurations has been further refined in terms of both geometrical details and injection strategy.

As visible in figure 11, the main parameters to be optimized in the development of a 2-Stroke combustion chamber are: compression ratio; squish ratio (i.e. the ratio of the squish area to the cylinder cross section); squish clearance (minimum distance from piston crown to cylinder head); bowl shape; piston crown slope; number, diameter and direction of the injector holes; injector nozzle position. For the automobile engine the injection strategy (pressure, number of pulses, timings) is fundamental, while the aircraft engine is much less demanding, so that a simple mechanical system may be even more suitable than a Common Rail.

In the automobile project, the combustion patterns calculated for the optimized 2-Stroke configuration have been then compared to the features observed in a reference 4-Stroke engine [21], at the same operating conditions (see [20] for details). Figure 12 presents this comparison in terms of Heat Release Rate / Cumulative Heat Release, in-cylinder average pressure, in-cylinder average temperature at 4 different engine speeds. Furthermore, figure 13 shows the distribution of Oxygen within the 2 chambers at different crank angles (engine speed is 3000 rpm)

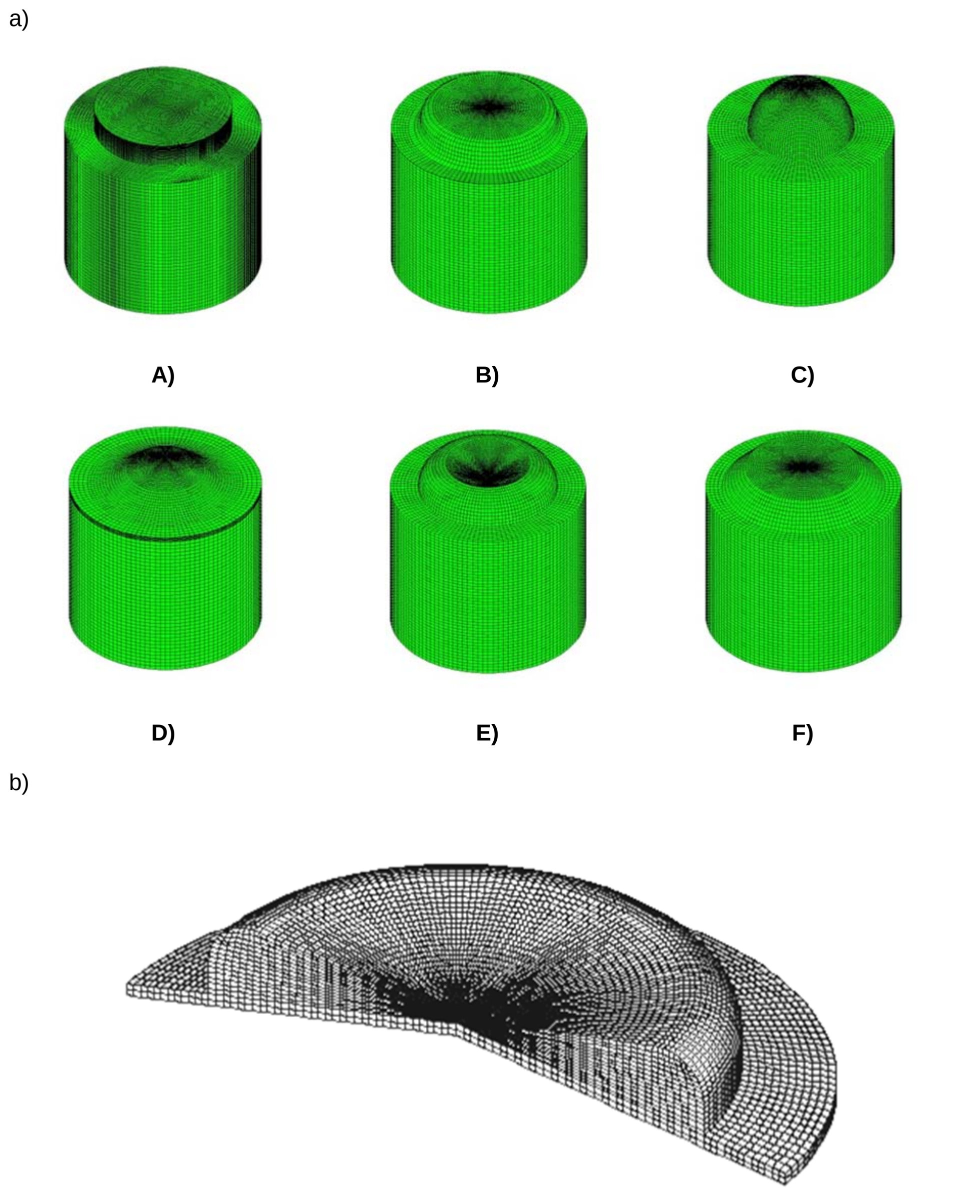

Figure 9. a): The analyzed combustion chamber grids at EPO/EPC for the automobile engine: A) Cylindrical; B) Smoothed Cylindrical; C) Spherical; D) Smoothed Spherical; E) Reverse Bowl; F) Conical. For details, see [8, 20].b): Chamber optimized for an aircraft engine [13, 14]

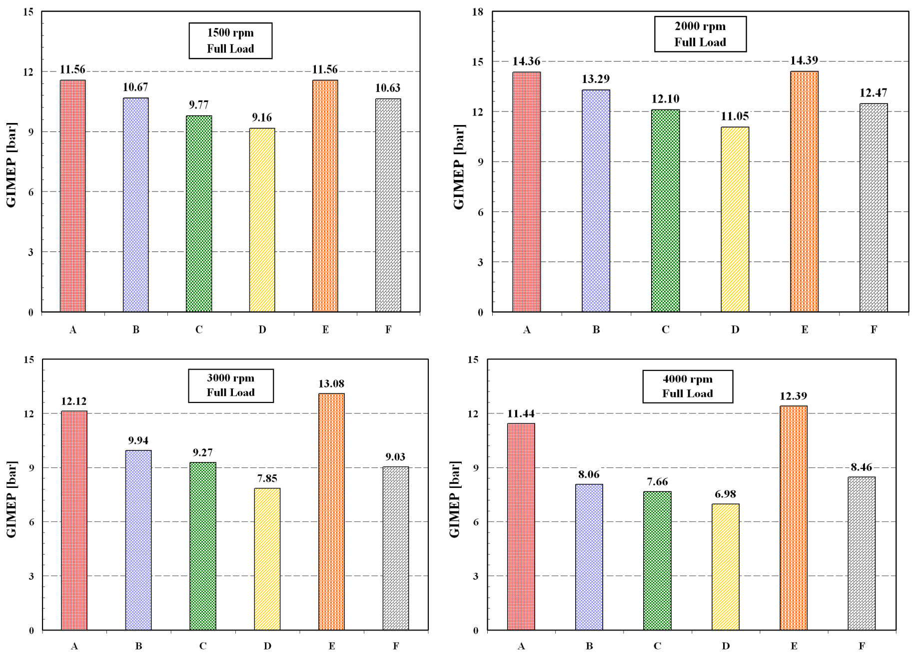

Figure 10. Calculated values of GIMEP at 1500, 2000, 3000 and 4000rpm, full load, for the different configurations presented in figure 9a. For details, see [8, 20]

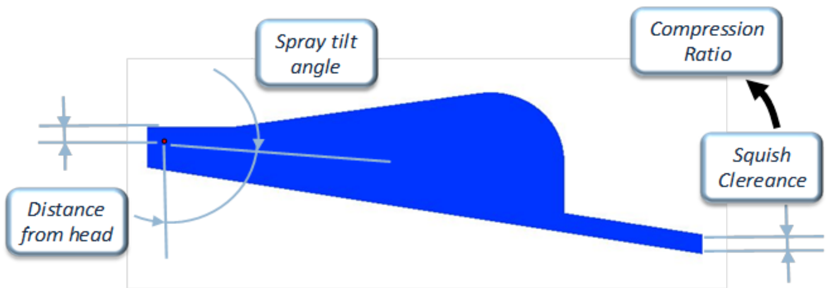

Figure 11. The most important parameters for the optimization of a reverse bowl combustion chamber

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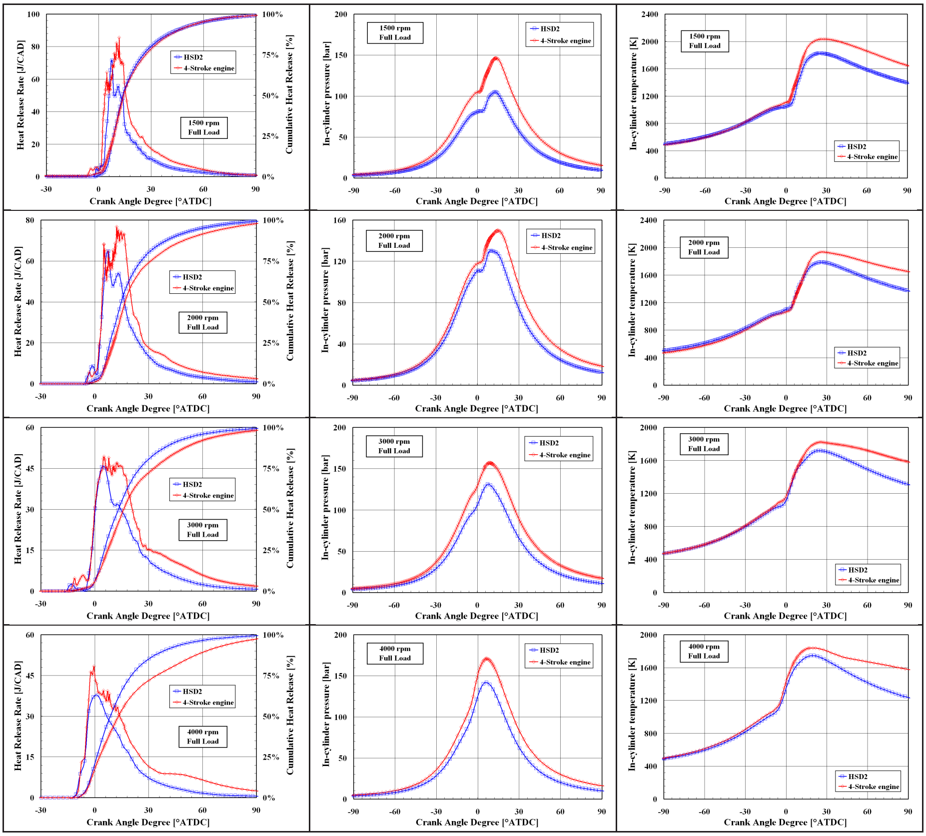


Figure 12. Curves of Heat Release Rate and of Cumulative Heat Release fraction (left), in-cylinder average pressure (middle) and in-cylinder average temperature (right) plotted for the 2-Stroke and the 4-Stroke engine. CFD-3D calculations performed at full load, engine speed: 1500, 2000, 3000 and 4000 rpm. For details, see [20].

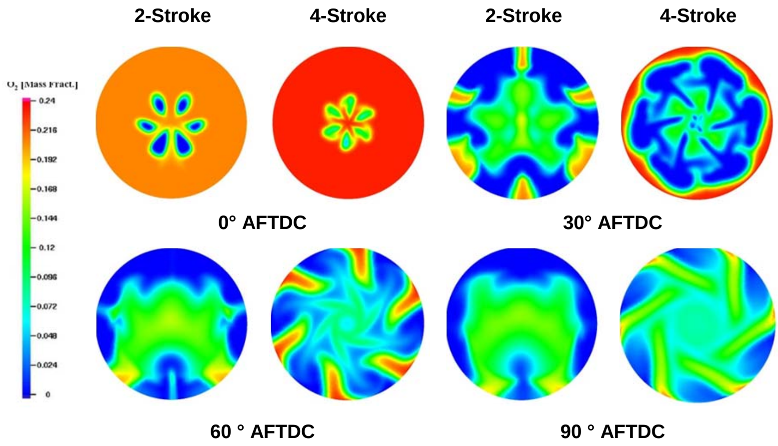


Figure 13. Oxygen concentration plotted on a plane perpendicular to the cylinder axis at 0, 30, 60 and 90° AFTDC for both the 2-Stroke and the 4-Stroke engine, at full load, 3000 rpm. For details, see [8, 20]

It is observed that, in the 2 Stroke engine, the peak of Heat Release Rate is always lower than that of the 4-stroke, mainly because of the larger amount of residuals trapped within the cylinder. This feature affects both in-cylinder pressure and temperature traces. From a point of view of thermo-mechanical stress and Nitrogen Oxides emissions, the lower temperatures and pressures are an important advantage, that can be exploited in order to increase boost pressure, thus power density, without risks of mechanical failures.

Furthermore, in the 2-Stroke engine the combustion process enters the completion phase (after the end of injection, when burnt gases, air and fuel vapor are mixing throughout the chamber) earlier than the conventional engine. Assuming the beginning of this phase when the 90% of fuel is burnt, the lead of the 2-Stroke over the 4-Stroke grows up as engine speed increases: while at 1500rpm this advance is about 5°, it becomes 30° at 4000 rpm. The explanation for this behavior is that, in the 2-stroke engine, fuel jet penetration is higher, for a number of reasons: bigger distance between injector and walls, lack of a strong charge motion interfering with sprays, lower rate of chemical reactions (because of the high concentration of residuals). Therefore, in the first part of the combustion process, fuel vapor is surrounded by more air, compared to that available in a conventional piston bowl, and it burns quickly, consuming all the oxygen at the periphery of the chamber. Later, the combustion rate decreases, since the unburned fuel have to diffuse back towards the cylinder axis, where some Oxygen may still remain. This behavior is clearly visible in the pictorial views of figure 13.

8. Engine performance

As well known, engine performance is related to the specific features of each project. For the sake of brevity, only two projects will be analyzed in this document: the former is the 1.05L automobile engine developed by the University of Modena and Reggio Emilia and described in [8], the latter is the aircraft engine of table 2, in both uniflow and loop versions [21, 22].

The automobile 2-S engine is a 3-cylinder, DI loop scavenged unit, having a total capacity of 1050 cc, bore x stroke: 70x91 mm, compression ratio: 19.5:1, supercharged and intercooled. The supercharging system is made up of a turbocharger, with variable geometry turbine, and a Roots blower, serially connected. The intercooler is between the two compression stages. Different versions of the engine have been developed, but only one will be considered here, for the sake of brevity. This version, named BASE, includes a valve, able to modify the opening/closing timing of the exhaust port. The 2-S automobile engine is compared to a reference 4-Stroke commercial unit, whose features are: 4-cylinder in line, direct injection with a Common Rail system; 4-valve; total displacement: 1251 cc; bore x stroke: 69.6x 82 mm; compression ratio: 17.6; turbocharged with a variable geometry turbine and intercooled; max. power 67 kW@4000 rpm; cooled EGR system, EURO IV compliant.

Since no prototype of the 2-S engine has been built at the moment, the comparison with 4-S is performed by means of CFD simulations, carried out at the same conditions and with models as similar as possible. In particular, at full load the injection rates are set in order to have the same value of trapped air-fuel ratio..

First, the comparison is made in terms brake torque, power and fuel specific consumption obtained at constant speed and full load. A graph of IMEP is added too, because of the importance of this parameter as an index of the engine thermo-mechanical stress. Such a comparison is shown in figure 14.

Figure 14 clearly demonstrates the superiority of the 2-Stroke engine under every point of view, except fuel economy. However, it should be considered that friction losses of the 2-stroke unit are probably over-estimated. A definitive confrontation, under this point of view, will be possible only when a 2-Stroke prototype will be physically built and tested.

For a passenger car engine, emissions at partial load are paramount. Therefore, a comparison between the 2-S and the 4-S engine is carried out at low load, corresponding to a brake torque of 60 N.m. The calculations are performed using a 3-D CFD tool (KIVA-3V) in combination with the usual GT-Power analysis.

As visible in table 3, Soot and Carbon Monoxide emissions are strongly reduced in the 2-Stroke engine (-89% and -75%, respectively), while the reduction of Nitrogen Oxides is less significant. However, it is reminded that the 2-Stroke engine does not need any EGR device to keep the NOx under control. These outcomes can be easily explained considering that the torque target in the 2-S engine is achieved at a much higher air-fuel ratio, because of the double cycle frequency and the presence of the Roots blower keeping the turbocharger speed higher. The air excess makes oxidation processes much more complete, while temperature remains low, without need of external EGR. The last issue has a positive influence also on brake specific fuel consumption, since the external EGR system introduces additional pumping losses.

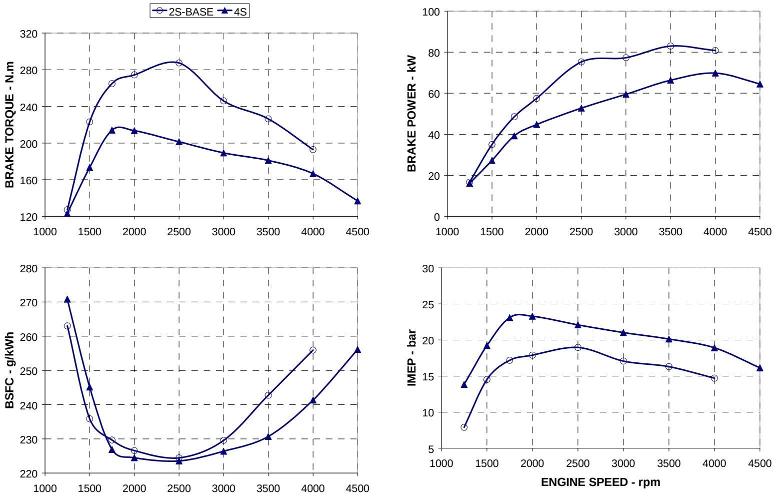


Figure 14. Comparison between the automobile 2-Stroke engine – BASE configuration and the 4-Stroke reference engine: results of CFD-1D simulation [8].

	NOx - g/kWh		CO - g/kWh		Soot - mg/kWh	
	2-S	4-S	2-S	4-S	2-S	4-S
1500	1.85	2.22	0.18	1.00	1.07	68.22
2000	1.41	1.96	0.40	1.08	28.13	56.26
2500	2.10	1.78	0.29	1.24	4.71	59.91
3000	2.02	2.10	0.23	1.15	6.18	172.42

Table 3. Comparison between the 2-Stroke and the 4-Stroke reference engine in terms of specific emissions at partial load (torque 60 N.m). See [8] for details.

A study for the development of the aircraft Diesel engine whose features are listed in table 2 is presented in [13]. On the basis of this work, a loop and a uniflow design will be compared, both of them featuring specifically optimized combustion and scavenging systems. Also the supercharging system is adapted for each configuration to the project goals.

For both engines, figure 15 shows the pressure traces and mass flow rates at inlet/exhaust ports, at 2600 rpm, full load. For the aircraft application, this is by far the most important operating condition, since it corresponds to the maximum speed at which the propeller can safely rotate. Typically, the engine is set at this speed and full load throughout the take-off. Furthermore, standard cruise conditions are generally very close to the top speed (no less than 80%)

The 3-cylinder lay-out is particularly suitable for the optimization of the exhaust dynamics, since at any engine speed there are three pulses, spaced at about 120°, corresponding to the blow-down from each cylinder. The phase of the pulses is almost perfect: after the scavenging ports close, the flow of fresh charge leaving the cylinder is blocked by the compression wave traveling from the manifold to the exhaust port/valves, that is generated by the cylinder next in the firing order. The manifold volume must be as small as possible, in order to minimize flow losses: in this way, pulses are strong, resulting in a better capability of retaining the fresh charge, as well as of transferring energy to the turbine, enhancing boosting. It is also interesting to observe that, in the Uniflow design, the advance of EVO is almost identical to the retard of EVC. This is an evidence of the fact that with a triple there is no need to reduce the retard of EVC, since the manifold dynamics alone are able to produce a good scavenging quality.

The differences of in-cylinder gas-dynamics between Loop and Uniflow are mainly related to the different exhaust design (piston controlled ports versus poppet valves). On the one hand, the poppet valves leave complete freedom in the timing choice, while the port advance coincides with the retard. On the other hand, with a cam-controlled lift profile it is more difficult to yield high flow areas. As just one example, AVL needed 4 poppet valves in the prototype described in [4], albeit for a higher engine speed than is required for a direct drive aircraft engine. In a 3-cylinder engine the manifold gas-dynamics, if properly tuned, can overcome the limitation inherent to the piston-controlled ports of the Loop engine, while it cannot help with the lack of flow area of Uniflow.

Finally, a comparison between the best Loop and Uniflow engine is shown in figure 16. Ambient conditions are at sea level (pressure 1.013 bar, temperature 293 K); a trapped air-to-fuel ratio of 20 is imposed at any speed, except at 2400/2600 rpm, where fueling is set in order to meet the power target of 150 HP. The positive displacement compressor is always active: however, its displacement and speed are set in order to have a pressure ratio close to unity at maximum engine speed, so that the blower has a very small influence on scavenging and fuel efficiency.

The results have been presented against speed at full load (subject to a trapped AFR of 20) to allow comparison with other CI engines. The reduction in fueling at 2400/2600rpm to respect the target rating has the effect of increasing the trapped AFR above 20 thereby ensuring the smoke limit is also respected. This has distorted the results away from a pure engine characteristic and inflected some of the curves, most notably the % burned mass at cycle start.

In a real aircraft application the results at full load and lower speeds are of little interest, due to the propeller loading curve. Furthermore, it can be seen that there is little torque “back-up” as it is expected for engines without any form of turbocharger control. This is also acceptable for the aircraft application since this class of aircraft will typically be fitted with a “constant speed” propeller where the blade pitch is varied by a suitable controller or “governor”. Indeed, a variable pitch propeller will be necessary to exploit the favorable altitude performance without engine over-speed.

It can be seen that the Loop engine consumes more air and requires a bigger compressor, not a major disadvantage. The Uniflow engine has much better trapping ratio at low speed and hence low air delivery to the cylinder but this is of no advantage to the aircraft application. At the top end of the speed range, the difference in trapping ratio is smaller. While the Loop engine needs more air mass flow, it gains from faster blowdown due to the fast opening of the exhaust ports compared to the cam operated valves in the Uniflow engine. Further, it is also more effective in trapping the exhaust pressure pulse that arrives just before exhaust port closing.

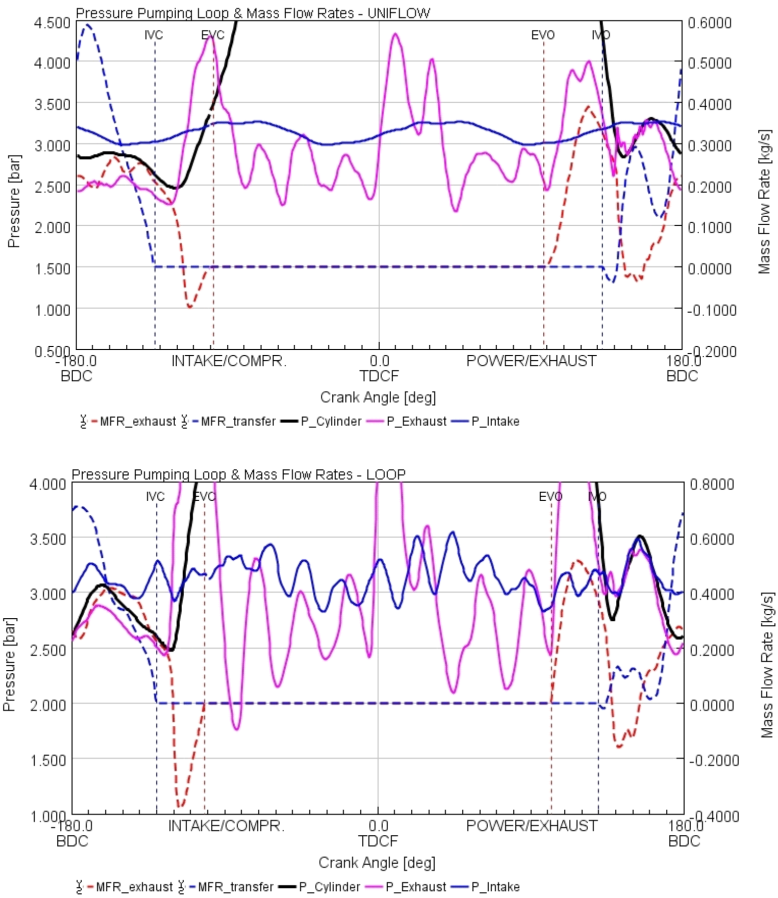


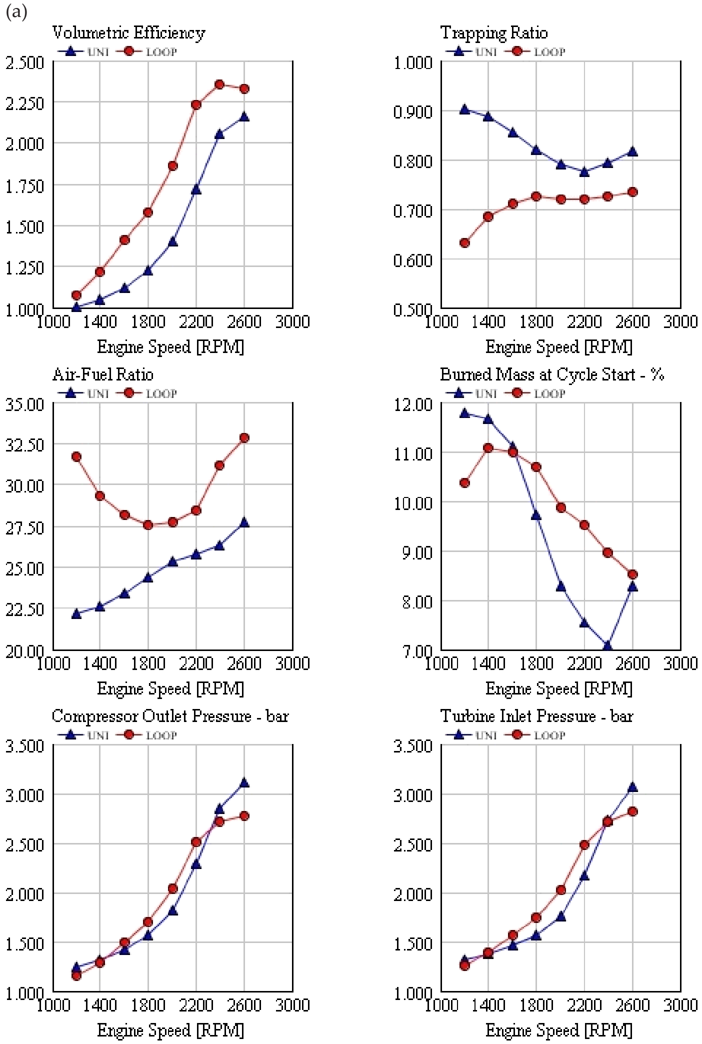
Figure 15. Pumping loop and mass flow rates at 2600 rpm, full load, calculated for the two types of aircraft engines described in [13]

The Loop engine has no greater cylinder pressure at rating, as it gains a benefit from lower friction and better cycle efficiency. The lower friction is due to the lack of a valvetrain. The cycle efficiency benefit can be explained by lower heat loss and the more advantageous combustion chamber geometry. Absence of swirl in the Loop engine further assists with reducing heat loss. The Woschni correlation incorporated in the GT-Power code and calibrated by comparison with KIVA, predicts cylinder heat loss of 1/3 less for the Loop engine, which will translate into reduced cooling pack size and cooling drag on the aircraft.

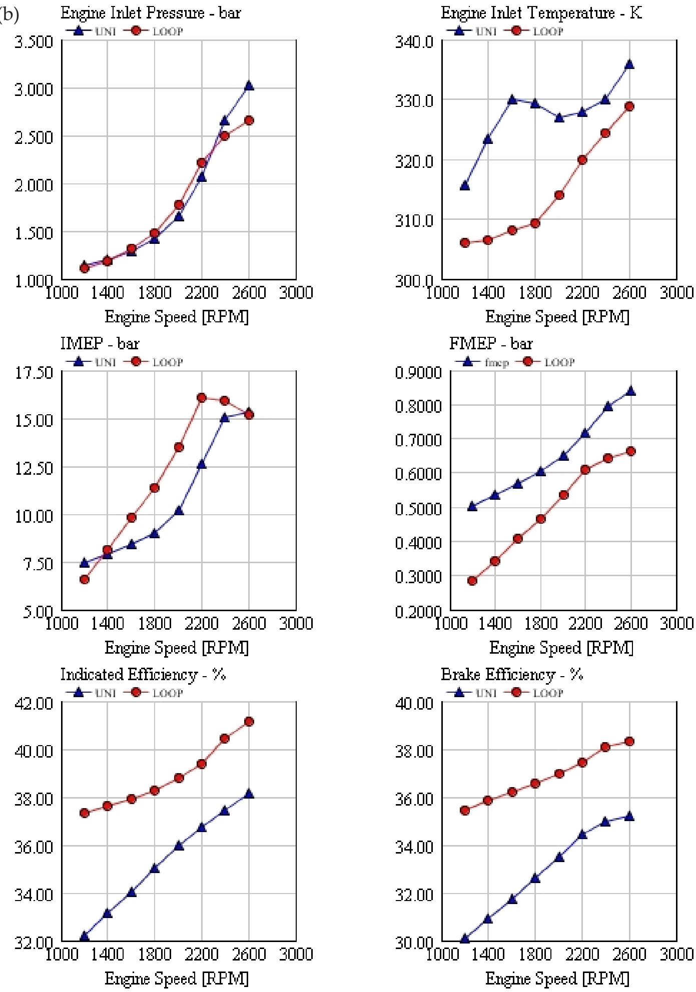
The net result of these efficiency gains put the Loop engine about 8% ahead in SFC at rating.

The higher AFR and better cycle efficiency of the Loop engine will result in lower exhaust gas temperatures thereby reducing its possible durability disadvantage against the Uniflow engine.

(a)



(b)



Engine Inlet Pressure - bar
UNI
LOOP
3.500
3.000
2.500
2.000
1.500
1.000
1000 1400 1800 2200 2600 3000
Engine Speed [RPM]
Engine Inlet Temperature - K
UNI
LOOP
340.0
330.0
320.0
310.0
300.0
1000 1400 1800 2200 2600 3000
Engine Speed [RPM]
IMEP - bar
UNI
LOOP
17.50
15.00
12.50
10.00
7.50
5.00
1000 1400 1800 2200 2600 3000
Engine Speed [RPM]
FMEP - bar
fmep
LOOP
0.9000
0.8000
0.7000
0.6000
0.5000
0.4000
0.3000
0.2000
1000 1400 1800 2200 2600 3000
Engine Speed [RPM]
Indicated Efficiency - %
UNI
LOOP
42.00
40.00
38.00
36.00
34.00
32.00
1000 1400 1800 2200 2600 3000
Engine Speed [RPM]
Brake Efficiency - %
UNI
LOOP
40.00
38.00
36.00
34.00
32.00
30.00
1000 1400 1800 2200 2600 3000
Engine Speed [RPM]

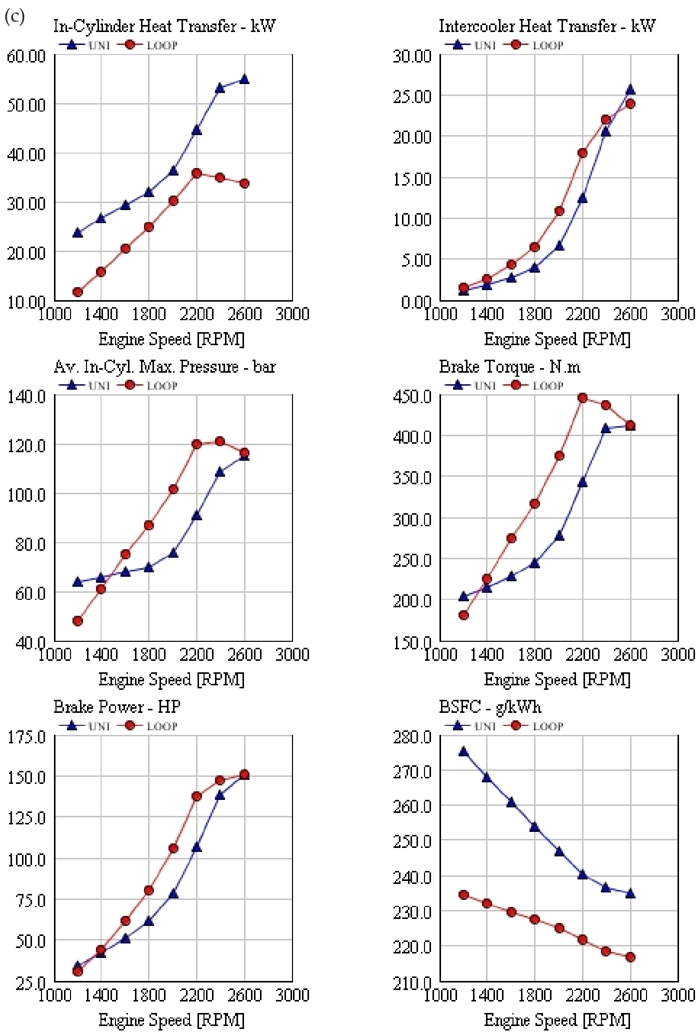


Figure 16. a) Comparison among Loop and Uniflow aircraft engines at full load (trapped AFR=20): scavenging parameters. For details, see [13]; b) Comparison among Loop and Uniflow aircraft engines at full load (trapped AFR=20). For details, see [13]; c) Comparison among Loop and Uniflow aircraft engines at full load (trapped AFR=20): performance parameters. For details, see [13]

9. Conclusion

The 2-Stroke cycle combined with Compression Ignition is a promising solution for high speed engines, particularly for small passenger cars and for light aircraft (power < 140 kW). In comparison to a corresponding 4-Stroke engine, the double cycle frequency yields the following advantages: higher power density, with ensuing possibility of downsizing and/or down-speeding the engine; higher mechanical efficiency, in particular with the piston-controlled ports (no valve-train); lower soot and NOx emissions at partial load, thanks to the higher air excess; possibility of having a high content of residuals within the cylinder without an external EGR system.

Since 1990, many prototypes have been designed and built, according to quite different concepts. The two most interesting designs, in the authors' opinion, are the Uniflow scavenging, with exhaust poppet valves, direct injection with bowl in the piston, and Loop scavenging, with piston controlled ports, direct injection and bowl in the cylinder head. Both solutions adopt an external supercharging system, so that lubrication can be the same of a conventional 4-Stroke engine.

The uniflow design is closer to the 4-Stroke engine, and its bigger advantage is to share most of the components with mass production engines. In particular, the combustion system and the valve-train is the same of passenger car Diesels. Conversely, the loop design requires a much bigger effort, since the combustion system must be developed according to new concepts, and a number of minor issues concerning piston rings and liner durability must be carefully addressed. The reward for properly addressing these issues is a very compact design and an excellent mechanical efficiency.

As far as scavenging is concerned, it is a widespread opinion that Uniflow is always better than Loop, in terms of efficiency. This is not the final outcome of the authors' investigations, whereas it was found that a strong support by CFD simulation can help the designer to close the gap between the two designs and even get a higher quality of the gas exchange process. The same CFD support is the key to develop efficient combustion systems, without need of a swirl motion within the cylinder.

In this document, the development of two different 2-Stroke High Speed Direct Injected Diesel engines is presented for two applications: small passenger cars (1.05 liter of capacity, 3-cylinder, power target 80 kW@4000 rpm) and light aircraft (1.8 liter of capacity, 3-cylinder, power target 110 kW@2600 rpm). In both cases, the design guidelines are discussed.

The superiority of the 2-S design in comparison to the 4-S stroke cycle is demonstrated by means of CFD analyses, performed by means of experimentally calibrated models. In particular, the passenger car 2-S engine is able to provide, from 2250 to 4000 rpm, a brake power higher than the peak value of the reference 4-Stroke engine (1.25 liter, 4 cylinder, turbocharged, peak power 65 kW@4000 rpm). Furthermore, engine-out soot emissions at partial load are about one order of magnitude lower, while NOx can be controlled without an external EGR system. As far as the aircraft engine is concerned, the 2-S design yields a big weight saving in comparison to a 4-Stroke engine delivering the same power; furthermore, the aircraft application requires strong modifications from the design used for passenger car engines, so that the transformation of an off-the-shelf Diesel engine has a cost very close to a brand new project.

At the moment of writing this document, the field of High Speed Compression Ignition 2-Stroke engines is an open research area. A lot of ground still has to be covered in order to develop reliable prototypes, able to practically demonstrate the theoretical advantages found by means of CFD simulation. In the automotive field, the concept may also find an application to the so-called “range-extenders”, i.e. internal combustion engines designed to recharge the batteries of electric vehicles. Here, the compactness and the low pollutant emissions level of the 2-Stroke cycle could play a fundamental role. In the aircraft field, the effort will be focused for keeping the engine design simple and reliable, in order to be competitive with the 4-Stroke SI engines also in terms of production and installation cost.

Nomenclature

1D/3D One/Three-Dimensional

AFR Air-Fuel Ratio

$A_{eff,av}$ Effective average area of transfer and exhaust ports

A_p Area of the piston (cross section)

BDC Bottom Dead Center

BMEP Brake Mean Effective Pressure

BSFC Brake Specific Fuel Consumption

CFD Computational Fluid-Dynamics

DI Direct Injection

DR Delivery Ratio

EGR Exhaust Gas Recirculation

EPO/EPC Exhaust Port Opening/Closing

EVO/EVC Exhaust Valve Opening/Closing

FMEP Friction Mean Effective Pressure

HSDI High Speed, Direct Injection

IDI In-Direct Injection

IMEP Indicated Mean Effective Pressure

Rpm Revolutions per minute

SE Scavenging Efficiency

S.I. Spark Ignition

TDC Top Dead Center

TE Trapping Efficiency

TPO/TPC Transfer Port Opening/Closing

Up Mean Piston Speed

Δp Pressure drop across the cylinder

ϱ Charge density

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