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## Design of a fuel-efficient two-stroke Diesel engine for medium size passenger cars: assessment of the best suited scavenging architecture, stroke-to-bore ratio and air-loop layout

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### Abstract

The European REWARD project (REal World Advanced technologies foR Diesel engines) funded by the European Union within the Horizon 2020 framework aims to develop a new generation of Diesel engines complying with stricter post Euro 6 legislation under real driving conditions, and with lower fuel consumptions. Among the different technologies investigated, a fuel-efficient two-stroke Diesel engine suited for B- and C- class vehicles is designed and experimentally evaluated.

Two-stroke engines offer high power densities resulting from doubling the combustion frequency compared with four-stroke engines. Despite this attractive advantage, the combustion occurring each revolution involves a very short time for the gas exchange processes and thus, a quasi-overlapping of the exhaust of burnt gases and the intake of fresh gases. This process is called scavenging and has to be properly optimized as it directly affects the overall efficiency of the engine.

**Keywords:** Two-stroke Diesel engine; Uniflow scavenging; Stroke-to-Bore ratio; Air-loop layout.

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## 1. Introduction

The fight against the global warming and the reduction of the atmospheric pollution are meaningful environmental concerns in our industrial societies and involve pursuing the continual effort of reduction of the fuel consumption and of control of emissions induced by internal combustion engines. The REWARD collaborative project (REal World Advanced Technologies foR Diesel engines), funded by the European Union within the Horizon 2020 framework, takes part to this technical challenge and aims more specifically to develop a new generation of Diesel engines even more efficient and complying with future stringent regulations of pollutant emissions. Among the different technologies developed, a fuel-efficient two-stroke Diesel engine suited for B and C-segment passenger cars is designed and experimentally evaluated. The partners involved in these activities are: IFP Energies Nouvelles, CMT-Universitat Politècnica de València, Czech Technical University, Delphi, AVL-LMM and Groupe Renault.

Two-stroke Diesel engines offer high power density, associated with a low weight and a small engine size. These particular qualities are especially well suited for Marine engines, which can reach very high thermal efficiency due among others to the very low rated speed of the engine (about 100RPM) and the large stroke-to-bore ratio (up to 4). Regarding on-road light duty applications, four-stroke engines represent almost the entire market of Diesel applications even if a renewed interest for the two-stroke Diesel engine has recently come up. The development of two-stroke engine suited for passenger cars is motivated by the reduction of the mass, the bulk and the cost of the powertrain thanks to the double combustion frequency. Thus, the number of cylinders can be reduced with a very low impact on the NVH and keeping a high unitary displacement and then high combustion efficiency. In addition to those advantages, two-stroke engines have also the potential of reducing engine-out NO<sub>x</sub> emissions, as pointed out by Xin (2011), thanks to the natural operation of two-stroke engine with large amount of residuals gases. Indeed, this property is particularly interesting for transients because the latency inherent to a classical Exhaust Gas Recirculation (EGR) system is avoided and lower NO<sub>x</sub> emissions can be thus anticipated. Moreover, Warey et al. (2016) have underlined that two-stroke Diesel engine may also yield a benefit in terms of CO<sub>2</sub> emissions. Indeed, Warey et al. have compared, through system simulations, two concepts of two-stroke engine and a hypothetical state-of-the art four-stroke Diesel engine that is anticipated to be in production in 2020. Their results show that the CO<sub>2</sub> emissions over two normalized driving cycles are at least reduced by 5%. In this frame, several attempts have been done to develop an efficient two-stroke engine suited for passenger cars. The works of Knoll (1998), Masuda et al. (1999), Nomura and Nakamura (1993), Tribotté et al. (2012), Brynych et al. (2014) and Redon et al. (2014) may especially be mentioned.

The basic difference between two-stroke and four-stroke engines is that the combustion occurs each revolution. This special feature implies that the transfers of gases have to be performed around the bottom dead center (BDC) and within a small available time and the technical solution consists of almost fully overlapping the phases of exhaust of burnt gases and of intake of fresh gases. This gases transfer process is called scavenging, and has to be well optimized because it strongly affects the overall efficiency of the engine (reduced volumetric efficiency due to a poor removal of residual gases). The main challenge consists of trapping a maximal amount of fresh gases while removing as much burnt gases as possible and avoiding, at the same time, the waste of fresh gases by a direct by-pass to the exhaust. From a local point of view, this involves that the fresh gases should push the in-cylinder burnt gases toward the exhaust without mixing, and that the flow is stopped once all the burnt gases have exited the cylinder. Therefore, a perfect scavenging is basically featured by no mixing and an excellent timing. Several engine architectures, described by Schweitzer (1949), and illustrated in Fig. 1, were developed in order to produce this ideal process:

- Engines with both intake and exhaust ports as shown in Fig. 1(a), without camshafts, which strongly reduces engine friction losses and presents the interest of a simplified mechanical layout. Two main scavenging configurations are available according to the arrangement of the ports around the cylinder. The first solution consists of setting the ports in opposite sides of the cylinder. This configuration called cross-flow scavenging is generally implemented with a deflector on the piston in order to compel the fresh gases to flow toward the top and thus scavenge the burnt gases from the top of the cylinder. The second arrangement is the loop scavenging configuration, with the ports located on the same side of the cylinder. This particular distribution of the ports aims to impel a U-turn movement of the flow into the cylinder. However, the proximity of the intake and exhaust facilitates the short-circuiting of fresh gases, but this issue may be solved through a detailed optimization and design of the intake ports. Mattarelli et al. (2016) performed such optimization on basis of 1352 CFD simulations, related to different transfer ports inclinations and different operating points. Lastly, one

may mention opposed piston engines which also operate with ports only. These engines may reach a very efficient scavenging, but the management of a very long stroke and of two crankshafts introduce a noticeable difficulty, especially for light-duty applications which require simple and compact solutions.

- The poppet valves engines, as displayed in Fig. 1(b), enable the loop scavenging. The main advantage of these engines is the mechanical configuration, very close to a conventional and well-known four-stroke engine. Despite this advantage, the loop flow motion makes the development of a proper swirl motion very difficult and the associated combustion system has to deal with this constraint. Moreover, Ternel et al. (2014), who developed a twin cylinder poppet valve engine, also emphasize that the design of the air loop is a primary parameter for achieving an efficient scavenging. Finally, permeability of valves is low compared to ports; this implies larger differential pressures between the intake and exhaust, which strongly penalizes the engine fuel consumption due to the energy required by an additional mechanical compressor.
- An intermediate architecture involves both ports and valves as illustrated in Fig. 1(c). The associated scavenging is called uniflow because of the induced main axial flow motion in the cylinder. Abthoff et al. (1998) have benchmarked fully-valves and uniflow scavenged engines dedicated to passenger cars application. Their results show that lower fuel consumption and larger specific power are achievable with the uniflow engine. The main reasons are the best fitted in-cylinder aerodynamics for combustion and the more efficient scavenging. Laget et al. (2013) have also shown by Computational Fluid Dynamics (CFD) simulations that an efficient scavenging can be reached with the uniflow scavenging if the ports are properly designed. In the frame of two-stroke gasoline engines, Ma et al. (2014) have also optimized by CFD the ports of a uniflow scavenged engine and the CFD outputs have been implemented in a system code model for assessing the engine performances.

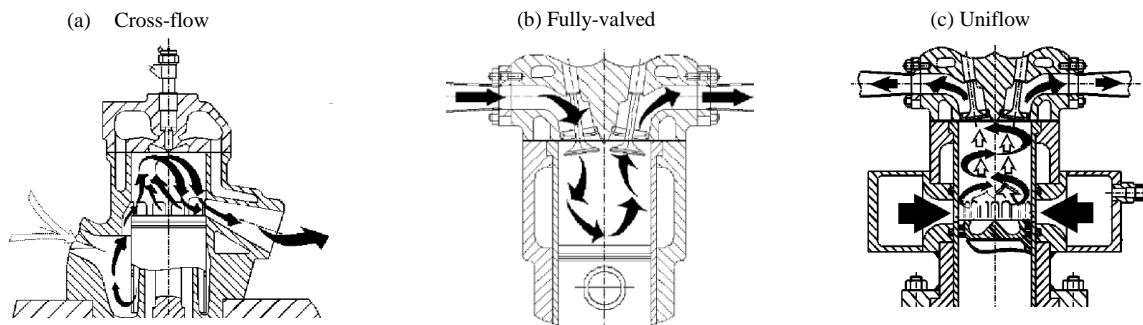


Fig. 1 Main two-stroke engine architectures

Due to its better efficiency mentioned by Abthoff et al. (1998), the uniflow scavenging architecture has been selected for this project. The two possible arrangements are benchmarked: either the standard uniflow configuration with the intake ensured by ports and the exhaust by poppet valves or the opposite, called reverse configuration. From the benchmark results, standard configuration is selected and several designs of the intake ports are investigated through 3D CFD simulations using CONVERGE. The best solutions are selected for being assessed on a single cylinder engine.

Another known leverage for improving the scavenging of the standard uniflow scavenged engine is to choose a large stroke-to-bore ratio for the engine architecture. However, a larger stroke-to-bore ratio raises questions about the effective gain on the scavenging, the feasibility of an efficient combustion with low emissions and the back effects of intake and exhaust permeability losses.

After defining the engine architecture, the activity is focused on the selection of the best air-loop layout based on 0D-1D simulations using GT-Power, considering a twin-cylinder engine with a specific power rate targeted at 60kW/L. A full set of possible architectures are evaluated: double stage super-turbocharging or turbo-supercharging and single supercharging, control by VGT or WG or SC bypass, coupled with exhaust VVT.

For the purpose of determining the engine load points which will be considered for both simulation and experimental investigations, a new tool has been developed. This tool statistically evaluates engine load according to the track profile, virtual engine map and car specification including gear box. The result of this evaluation is the time duration at each engine load point when the defined car is going through the defined track. RDE cycle has been evaluated for B- and C-class vehicles in this case. The load points with the longest time duration are then considered.

## 2. Engine architecture

### 1.1. Benchmark between standard and reverse uniflow

The comparison between the standard and reverse uniflow configurations is achieved out by introducing system code simulations which allow in studying the impact of setting the ports for the intake or the exhaust on the one hand and by analyzing 3D CFD results which yield information about the scavenging process from a flow point of view on the other hand. System simulations are performed with LMS Imagine.Lab Amesim, considering three engine operating points. Simplifying assumptions are adopted and the same scavenging law, the same combustion process and same intake and exhaust pressure ranges are considered for both standard and reverse configurations. No EGR is considered and acoustics effects are not taken into account since the air-loop is not explicitly represented. The main considered criterion is the corrected ISFC that accounts for the supercharger work which is withdrawn from the engine work. Constant turbine and compressor efficiencies are considered, respectively equal to 0.7 and 0.6, and the supercharger efficiency is assumed to be equal to 0.65. These global efficiencies are related to realistic values achievable with on-market technologies such as those mentioned by Froehlich and Stewart (2013). The after-treatment system is modeled by a global pressure drop.

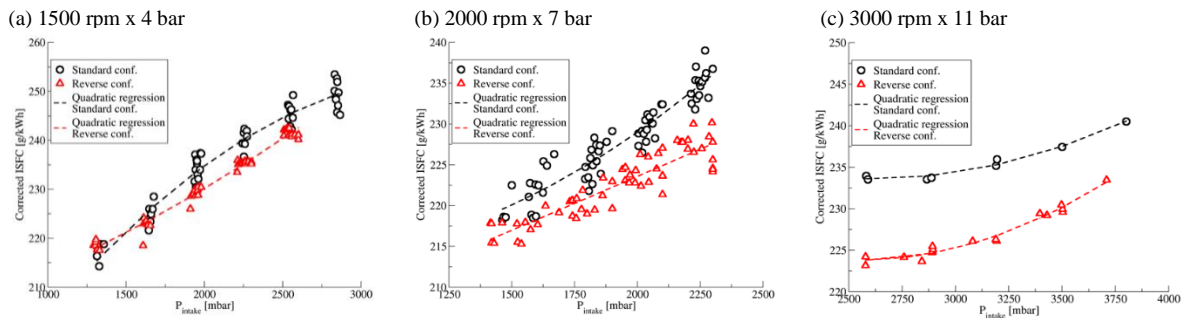


Fig. 2 Corrected ISFC vs. intake pressure for three operating points and comparison between the standard and reverse configurations

The Fig. 2 shows the comparison of the corrected fuel consumption for the three computes operating points. The reverse configuration presents lower fuel consumption considering above mentioned hypothesis, but this advantage appears mainly for the full load engine operating point. This is explained by an advantage in term of expansion to compression ratio with a lower penalty on the charging ratio at high load by Galpin et al. (2017).

This benchmark is completed with 3D CFD analysis using CONVERGE version 2.2. Three geometries, as shown in Fig. 3 are compared, considering intake and exhaust diagrams and valve lifts chosen from the previous system simulation results. The case 1 is devoted to the standard uniflow while the cases 2 and 3 compare two different geometries for the reverse uniflow, which differ by the intake ducts. The intake pipes of case 2 mainly promote the filling without generating aerodynamic swirl motion. Regarding case 3, the ducts are inspired from conventional four-stroke Diesel engine. Two tangential and two helicoidal ducts are arranged in order to impel a coherent in-cylinder swirl motion. The main considered criteria for scavenging efficiency assessment are the trapping ratio ( $TR$ ), which quantifies the efficiency of the trapping of the intake gases and thus yields information about the amount of fresh gases short-circuited, and the scavenging ratio ( $SR$ ), which is the proportion of fresh gases with respect to the in-cylinder total mass at the engine closure, and quantifies the efficiency of the scavenging of in-cylinder burnt gases. The trapping and scavenging ratios are respectively given by the following equations:

$$TR = \frac{m_{FG \text{ trapped}}}{m_{FG \text{ intake}}} \quad (1)$$

$$SR = \frac{m_{FG \text{ trapped}}}{m_{trapped}} \quad (2)$$

Where  $m_{FG \text{ trapped}}$ ,  $m_{FG \text{ intake}}$  and  $m_{trapped}$  are respectively the total in-cylinder mass of trapped fresh gases, the intake mass of fresh gases and the total in-cylinder mass of gases at the end of the scavenging.

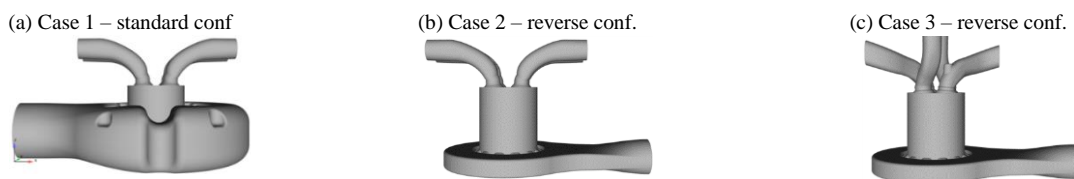


Fig. 3 Investigated geometries by CFD

For the three geometries, the computed scavenging curves, defined by the evolution during the scavenging process of the normalized ratio between burnt and fresh gases in the cylinder ( $Y_{BG-Cyl}$ ) relative to the one in the exhaust ( $Y_{BG-Exh}$ ) are shown in Fig. 4 and the trapping and scavenging efficiency ratios are reported in Table 1. The Case 1 provides the closest scavenging curve to the target, for which the values of the trapping ratio and the scavenging ratio are expected to be higher than 90%. Thus, it may be concluded that the best values are reached for the Case 1 which is the standard uniflow configuration.

From these 3D CFD simulations results analysis, the standard uniflow configuration allows getting the most efficient scavenging process among the different studied cases. As discussed by Galpin et al. (2017), the scavenging of the reverse configuration is strongly penalized by the vortex shedding that develops downstream the valves, which enhances the mixing between the in-cylinder burnt gases and the intake fresh gases.

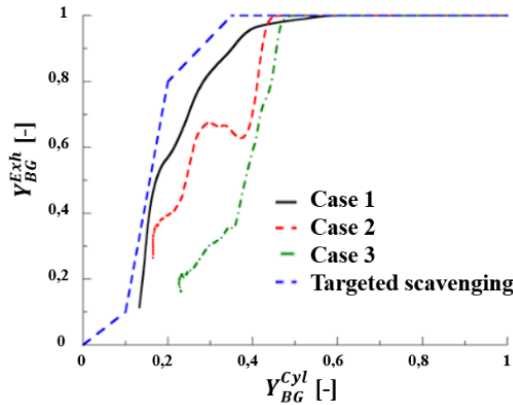


Fig. 4 Investigated geometries by CFD

Table 1. Trapping and scavenging ratios assessed by CFD

	Case 1	Case 2	Case 3
Trapping ratio (Eq. 1)	91%	82%	68%
Scavenging ratio (Eq. 2)	88%	83%	77%

Several geometrical options are evaluated with the aim of improving the scavenging efficiency of the reverse uniflow configuration, including different intake valve diameters, intake pipes geometry and even introducing mask in the valves or in the cylinder head.

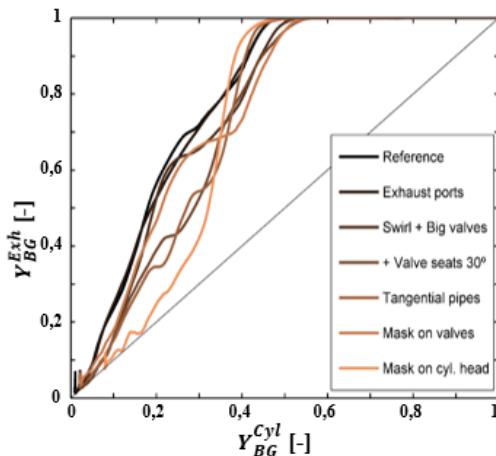


Fig. 5 Scavenging curves and efficiency parameters obtained with different reverse uniflow configurations

Table 2. Scavenging efficiency parameters with different reverse uniflow configurations

Trapping ratio [%]	Scavenging ratio [%]
38.41	98.95
39.56	98.73
37.51	97.84
38.20	97.62
37.01	97.44
45.74	97.63
47.10	91.93

The results shown in Fig. 5 and Table 2 confirm that the reverse uniflow configuration does not provide benefits over the standard uniflow configuration in terms of the scavenging curve, and this solution has been thus given up. Moreover, these simulation results highlighted the importance of optimizing the intake geometry to improve the scavenging process, the exhaust design having a very low impact over the air flow through the cylinder.

Considering the standard uniflow configuration (intake by ports and exhaust by valves), an intensive optimization of the intake ports design has been performed and very different geometries have been investigated. As results of huge amount of 3D CFD simulations with CONVERGE, in term of scavenging performances, four intake ports designs selected for being assessed on a single cylinder engine for experimental evaluation are presented in Fig. 6. The corresponding trapping and scavenging efficiency ratios are reported in Table 3. Two total ports height are considered with two design types:

- Double rows type: wider ports on the bottom are designed for intake permeability and scavenging performances and the narrow ones on the top are designed for promoting the swirl movement of the in-cylinder fresh gases.
- Single twisted row type: both performances and swirl movement are achieved with the single ports row. The swirl movement is produced and controlled by the twisted geometry of the ports.

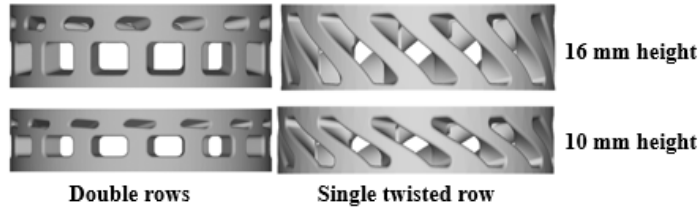


Table 3. Scavenging efficiency parameters with standard uniflow configuration

	16 mm height		10 mm height	
	TR	SR	TR	SR
Double rows	84%	91%	88%	89%
Single row	69%	95%	83%	88%

Fig. 6 Best intake ports designs and scavenging efficiency parameters obtained with standard uniflow configurations

Additional to the intake ports final optimization on the single cylinder engine, since the exhaust is ensured by poppet valves, the valves law is adapted to each port height design and a VVT system is also considered in order to increase the scavenging efficiency by optimizing the opening and closing timing of the exhaust valves. A complete testing program with a combination matrix of these scavenging design parameters will be performed on the single cylinder engine in order to find out the best standard uniflow definition with the highest scavenging efficiency.

### 1.2. Definition of the best suited Stroke-to-Bore (S/B) ratio

In order to find out the best bore diameter and piston stroke regarding the scavenging performances, three stroke-to-bore ratios ( $S/B=1.45, 1.24, 1.0$ ) are investigated by CFD simulations on the basis of a single cylinder engine with the same displacement of 500cm<sup>3</sup>. This sensitivity has been achieved with a reference geometry which is stretched according to the S/B ratios. The obtained results are shown in Fig. 7 and clearly underline that:

- An increase of the S/B ratio leads to a mutual increase of the scavenging and trapping efficiency, scavenging ratios (see Fig. 7(a)).
- An increase of the S/B ratio leads to an increase of the trapped mass of the fresh air (see Fig. 7(b)).

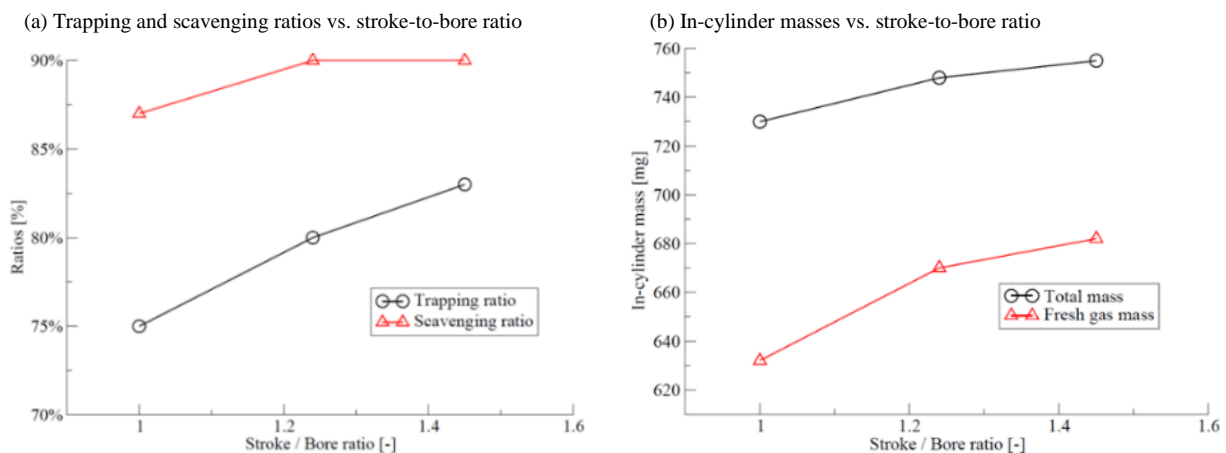


Fig. 7 Scavenging performances vs. stroke-to-bore ratio

There is a clear improvement of the scavenging with large values of S/B ratio; this can be explained by the following elements:

- The by-pass of intake fresh gases to the exhaust is delayed with a large S/B ratio due to the large distance to be covered from the bottom of the cylinder up to the top.
- The mixing between fresh and burnt gases is reduced with a large S/B ratio because of the decrease of the mixing volume (as a first approximation, it depends only on the bore diameter).

The Table 4 is summarizing the final engine architecture retained for the rest of the project.

Table 4. Features of the engine configuration

Displaced volume	500 cm <sup>3</sup>
Bore	76 mm
Stroke	110 mm
Stroke-to-Bore ratio	1.45
Scavenging architecture	Standard Uniflow

### 3. Air-loop definition

Scavenging of two-stroke engine requires a positive pressure difference between inlet and exhaust systems. The excess of scavenged fresh air – the greater, the poorer scavenging efficiency is – dilutes burnt gas, decreasing its temperature upstream of the turbine. Lower exhaust gas temperature reduces turbine power, decreasing achievable pressure difference between compressor outlet and turbine inlet. Moreover, the inlet-exhaust pressure difference has to insure the necessary flow-rate during scavenging, which is significantly depending on the available flow area of the engine valves during scavenging period. The missing pressure difference has to be covered by a supercharger, which is consuming part of the engine brake power. The higher is the pressure and the mass flow rate to be reached, the greater is the engine break power consumption, as described by Brynych et al. (2014).

Two-cylinder engine, thus 1L total displacement, has been selected for all following 1D simulations using GT-Power. The same combustion law and exhaust valve lift curve are considered for all configurations simulated. Engine friction losses are estimated based on data from state of the art four-stroke and four cylinders Renault K9K 1,5L engine. The scavenging law and the intake ports area are changed according to 3D CFD simulation results.

Honeywell VNT turbocharger and Eaton TVS-VR250 supercharger are taken into account in different arrangement (high-pressure stage or low-pressure stage) depending on the investigated air loop layout. Maximum combustion pressure is limited by the value of 160 bar. Supercharger drive with continuously variable transmission is considered. This solution avoids supercharger by-pass need at some operation points and thus improve BSFC. In the future, electric driven supercharger with reasonable efficiency could be investigated.

The engine model contains from intake piping including air filter, low pressure (LP) compressor (or supercharger), LP intercooler, middle pressure EGR loop including intercooler, high pressure (HP) supercharger (or compressor), HP intercooler, engine intake ports and cylinders, four exhaust valves per cylinder, 2 liters exhaust mixing volume, turbine, exhaust gas aftertreatment (AFT) system downstream the turbine and exhaust piping including a muffler. The exhaust aftertreatment system is considered downstream of the turbine with the target to keep a standard four-stroke AFT system design.

The main goal of all of the simulations was to optimize the whole engine air loop to be able to obtain the lowest possible brake specific fuel consumption (BSFC) at defined engine operating points. This should be achieved next to the others by minimizing of the supercharger power and finding an appropriate turbocharger.

The considered engine load points for air loop optimization task are presented in Table 5. The methodology applied for the selection of these engine operation points is described in section 4.

Table 5. Engine operating points.

		OP 1	OP 2	OP 3	OP 4	OP 5
Engine Speed	[rpm]	1500	1500	2500	2500	3000
IMEP	[bar]	4	10,4	4	15,1	~13,5 (~60kW/L)
EGR rate	[%]	20	10	20	0	0
FAER	[-]	0,6	0,7	0,6	0,77	0,77

Two basic air loop layouts as shown in Fig. 8 were simulated:

- Fig. 8(a) - Turbo-Super configuration: Turbocharger at LP and supercharger at HP. This configuration is more feasible from the turbocharger point of view. There is almost the same volumetric flow through the compressor and turbine thus the size of both wheels is very similar. Moreover, the supercharger could be of smaller size which should have also smaller input power.
- Fig. 8(b) - Super-Turbo configuration: Supercharger at LP and turbocharger at HP. It may be quite difficult to combine the small compressor with a big turbine in this case. The issue is more complicated, when the machine



size differs in such a way that each machine should come from other product row. Then there may occur issues with assembly of the machine casings.

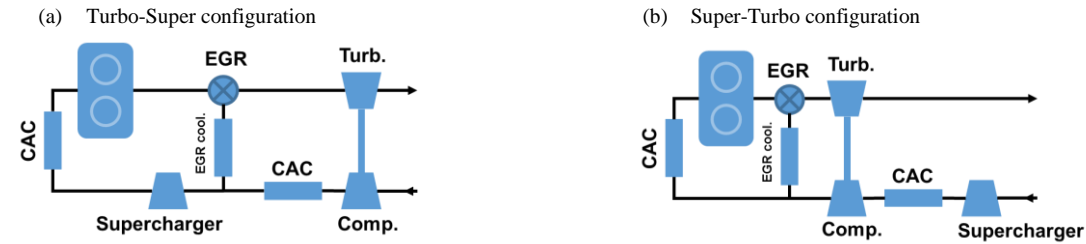


Fig. 8 Investigated air-loop layouts

No significant difference in fuel consumption of both configurations is found out. Nevertheless, as we can see on the bottom right hand side of Fig. 9, with super-turbo configuration, the supercharger is out of its map at 2500rpm/15.1 bar of IMEP, because of higher pressure ratio needed for this engine operating point.

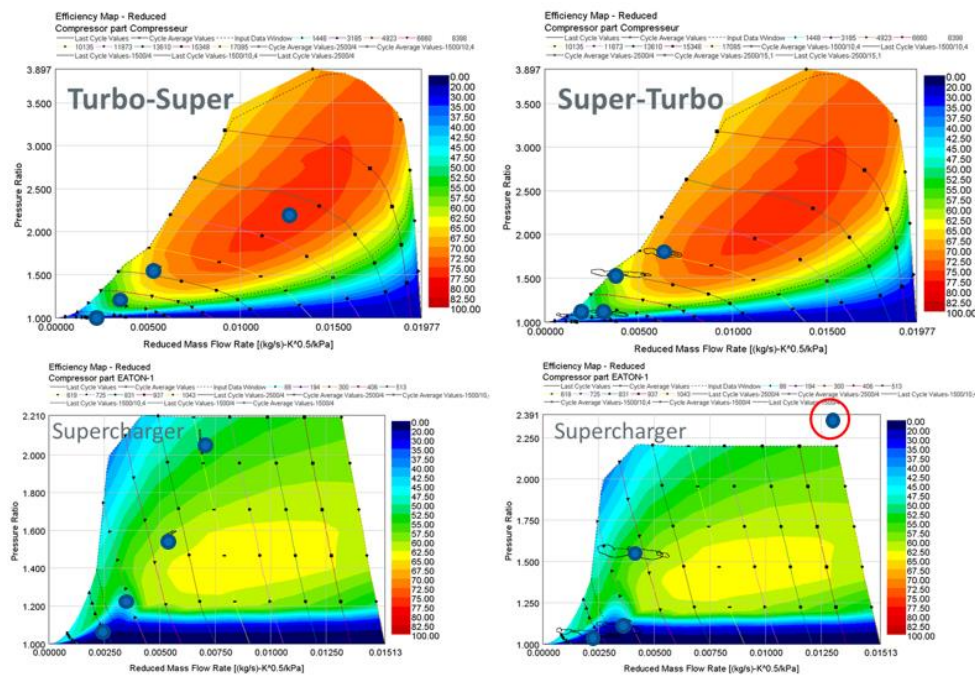


Fig. 9 Boosting device operating points plotted in maps of a turbocharger compressor and a supercharger

After the engine global optimization, Turbo-Super configuration is found as the best solution. Indeed, it leads to a slightly better BSFC and no issues with turbocharger availability are identified contrary to the Super-Turbo configuration, for which a relatively small compressor wheel could be needed. Additionally, the supercharger can be smaller, which could lead to lower friction losses. The supercharger power input depends proportionally on air inlet temperature. That is why efficient intercooling is also recommended.

Comparing VNT and wastegated turbine with Turbo-Super configuration, Table 6 and Table 7 below are showing that VNT offers better solution at high load and speed points, mainly thanks to higher turbine efficiency achievable. Nevertheless, keeping the peak pressure limit seems easier using wastegated turbine, but it is paid for by lower engine efficiency at the rated power.

Table 6. CZ C09 Wastegated Turbine results.

Turbo-Super Configuration – CZ C09 Turbine						
Operating point		1500rpm 4bar IMEP	1500rpm 10,4bar IMEP	2500rpm 4bar IMEP	2500rpm 15,1bar IMEP	3000rpm - 60kW Brake
BSFC	[g/kW.h	258	227	270	240	257
Max. Pressure	[bar]	57	110	63	159	157
Brake Torque	[Nm]	53	148	48	197	191



Table 7. Honeywell VNT results.

Turbo-Super Configuration – Honeywell VNT						
Operating point		1500rpm 4bar IMEP	1500rpm 10,4bar IMEP	2500rpm 4bar IMEP	2500rpm 15,1bar IMEP	3000rpm - 60kW Brake
BSFC	[g/kW.h]	258	226	277	230	246
Max. Pressure	[bar]	55	106	74	158	168
Brake Torque	[Nm]	53	148	49	208	191

#### 4. Engine operating points definition

The methodology applied for the selection of the engine operation points considered for all investigations is consisting in calculating the relative weight of engine speed and load map areas when the considered vehicle is rolling on a given drive cycle. If needed, more detailed information about this methodology can be found in publications of Steinbauer et al. (2016) and Macek and Morkus (2017).

For current investigations, WLTC cycle is considered with B-class vehicle with the peak power limit for test of 50 kW (rated power of four-stroke engine 66 kW). Evaluation of the most used modes is based on the engine work (in W.h), and performed inside a small range of engine speed and power. Fig.10 shows an example of results using a state of the art four-stroke engine in both positive power and braking part of the engine map. The most working modes, recalculated to two-stroke engine with the same mean piston speed are presented in Table 8.

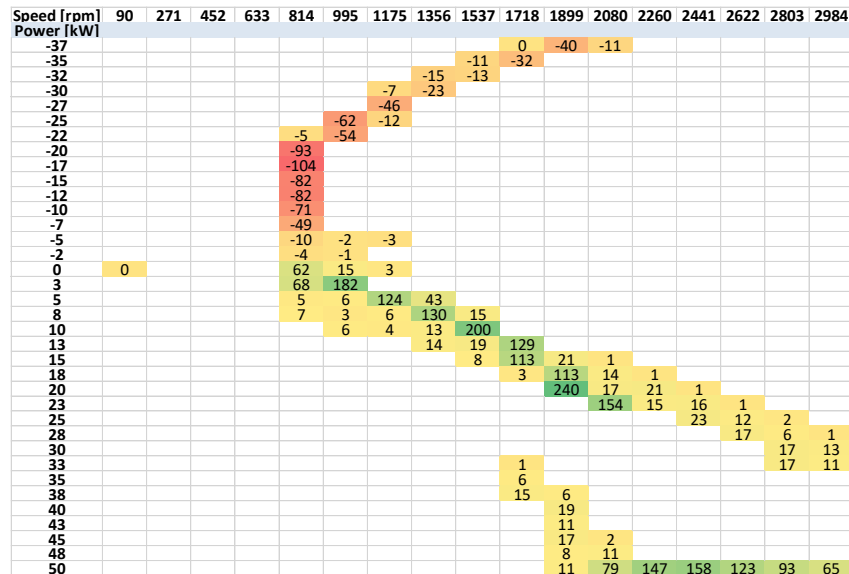


Fig. 10 Engine Work (in W.h) during WLTC at different operation modes

Table 8. Most used modes of operation at speed reduced for two-stroke engine and with rated power of 60 kW.

Selected Modes of Operation										
Speed	[rpm]	1899	1537	995	2441	2080	2260	1356	1718	1175
Torque	[N.m]	101,8	63,7	26,4	196,6	104,5	212,3	54,6	70,9	42,6
Power	[kW]	20,2	10,2	1,7	50,2	22,7	50,2	7,7	12,7	5,2
BMEP	[Bar]	6,55	4,09	1,70	12,64	6,72	13,65	3,51	4,56	2,74

The selected engine operating points for all investigations, as presented in Table 5, has taken those modes into account. Moderate speeds were represented for 1500 rpm, high speeds for 2500 rpm. High-torque mode at low speed and maximum rated power at rated speed were also added.

#### 5. Conclusions and perspectives

This article summarizes the detailed design from scratch of a fuel-efficient two-stroke uniflow Diesel single cylinder engine. This engine once assembled will be intensively assessed during experimental test campaigns and constitutes the first step toward an industrialized multi-cylinder engine suited for medium size passenger cars. The

first task achieved in this design process is dedicated to the definition of the scavenging architecture. After investigations using 1D and 3D CFD simulations, the standard uniflow configuration is selected. Moreover, several designs of the intake ports were investigated through 3D CFD simulations using CONVERGE. The four best solutions shown in Fig. 6 are selected for being assessed on the single cylinder engine. The second task was to fix the best suited Stroke-to-Bore (S/B) ratio. The highest investigated value, which is 1.45 is selected. The third task was to investigate the best suited air loop layout for this two-stroke Diesel engine. After investigations using 1D simulation with GT-Power, Turbo-Super configuration is recommended with wastegated turbine and efficient intercooling.

The next step of the project will consist in experimental investigations on the single cylinder engine for assessment of both scavenging and combustion performances. The global objective is to demonstrate low fuel consumptions and low engine-out emissions ability of two-stroke Diesel engine.

The tasks in this project are split between the partners involved in the dedicated workpackage of the REWARD project dealing with the development of a fuel-efficient two-stroke Diesel engine.

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