

Comparison between 2 and 4-Stroke Engines for a 30 kW Range Extender

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ABSTRACT

The paper compares two different design concepts for a range extender engine rated at 30 kW at 4500 rpm. The first project is a conventional 4-Stroke SI engine, 2-cylinder, 2-valve, equipped with port fuel injection. The second is a new type of 2-Stroke loop scavenged SI engine, featuring a direct gasoline injection and a patented rotary valve for enhancing the induction and scavenging processes.

Both power units have been virtually designed with the help of CFD simulation. Moreover, for the 2-Stroke engine, a prototype has been also built and tested at the dynamometer bench, allowing the authors to make a reliable theoretical comparison with the well assessed 4-Stroke unit.

Even if the optimized design of each one of the two engines is similar to that of existing prototypes, the paper is not intended to be a benchmarking, but a general study, aimed to define the fundamental project guidelines and compare different solutions under the same conditions, including the unavoidable arbitrary hypotheses.

The main results of the comparison may be summarized as follows: the 2-Stroke engine is more compact and light (–38% of frontal area, 35 vs. 50 kg); its fuel efficiency is slightly better, and further improvements are possible running on stratified charge; the reduction of NOx in the 2-S catalyst may not be complete, due to the unavoidable air short-circuit.

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INTRODUCTION

The electrification of powertrain is assuming an increasing importance in the worldwide race towards CO₂ reduction. One of the main challenges for its market success is the electric driving range achievable with the current battery technology, which allows the vehicle to store only a small fraction of the energy available using conventional fuels, such as gasoline or diesel. An effective way to address the so called range anxiety is the adoption of a range extender, that is a small internal combustion engine designed to recharge the traction batteries when their available energy content drops below a specified threshold. The range extender is also a practical solution to further issues. As an example, the range extender enables a more efficient use of heating and air-conditioning functions when the vehicle stops, thus helping to reduce CO₂ emission.

The basic requirements for the range extender are:

- compactness and lightness, in order to allow an easy and not expensive installation on a conventional vehicle construction;
- good fuel efficiency, to maximize the benefits on CO₂ reduction of the electric vehicle;
- low pollutant emission levels, to meet stringent regulations;
- excellent NVH behavior: ideally, the driver should not be able to tell when the combustion engine is running;
- low cost.

For vehicles needing a power higher than 40-50 kW, many carmakers already have got, in their own products portfolio, engines able to fulfill the above mentioned requirements. Conversely, for city cars needing less than 30 kW, even the smallest existing automotive engine is not suitable. Apparently,

the easiest solution is to convert engines, already existing for other applications (motorcycles, small gen-sets, etc.), into range extenders. However, the engineering cost for these conversions would be not much lower than a development from scratch. Moreover, a specifically developed engine allows the designer to select more suitable technologies and solutions. This is the reason why some important players of the automotive industry, not only OEMs, decided to develop their own Range Extender.

The goal of the paper is to compare two different design concepts for the development of a range extender rated at 30 kW (64 Nm) at 4500 rpm. Higher speeds are not suitable for the ensuing NVH issues. This power target is ideal for a C-segment passenger car, as demonstrated by the choice made by Mahle Powertrain [1] and FEV-KSPG [2], whose products are the current reference for the category.

It should be noted that, even if the optimized design of each one of the two engines is quite similar to that of existing prototypes, the paper is not intended to be a benchmarking, but a general study, aimed to define the fundamental project guidelines and compare different solutions under the same conditions, including the unavoidable arbitrary hypotheses. The authors acknowledge that any modification to these conditions may change the outcome of the study.

4-STROKE CYCLE FOR RANGE-EXTENDERS?

When considering conventional 4-Stroke engines, cost is the paramount criterion of choice. Turbocharging and mechanical supercharging are immediately discarded, also for the increased complication and overall dimensions of the intake/exhaust systems. Without turbocharging, Compression Ignition engines are not very attractive, because of the low power density, as well as the cost and weight of the after-treatment system.

With a naturally aspirated SI engine, the displacement required to meet the above mentioned performance target depends on the chosen technology. Typical automotive 2-valve PFI engines yield a specific maximum torque of 80-85 Nm/l below 4500 rpm; with a 4-valve pentroof combustion chamber the figure may rise up to 100 Nm/l; higher values can be reached only with devices able to vary the valve timings or the intake runner geometry, as well as by adopting a direct fuel injection.

Because of the cost constraints, the most suitable technology appears to be a simple PFI 2-valve combustion system, with in-block camshaft and pushrods. The "antiquated" push rod technology bears no restrictions to the functionality of the engine as high engine speeds are not desired due to NVH considerations. Conversely, the dimensions of the engine head may be reduced, in comparison to a conventional overhead camshaft.

The displacement resulting from the combination of the performance target and the choice of valves is about 0.8 l. With such a capacity, the number of cylinders must be no less than 2, in order to adopt conventional components. Since both Mahle and FEV-KSPG [1, 2] selected a two-cylinder lay-out, only this solution will be considered in the current paper.

As far as the cylinder lay-out is concerned, a V-90° design appears better than a 2 in-line, because of the possibility to fully balance the first order mass forces by means of crankshaft mounted counter-weights, instead of installing a balancer shaft, that cause additional parasitic losses. Regarding its space requirements, a V-90° concept with a disc-shaped generator directly flanged to the crankshaft is also attractive.

For the V2-90° range extender, FEV-KSPG proposed a one-piece high pressure die cast aluminum crank case in conjunction with low friction roller bearings. As a result, an extremely low friction level is expected, while the dimensions of the oil pump may be significantly reduced. An additional advantage of the cracked crankshaft is the fact that the connecting rods can be made of one piece, so that they can be manufactured as low-cost sheet-metal formed parts.

2-STROKE CYCLE FOR RANGE-EXTENDERS?

An unconventional but not exotic alternative to the 4-Stroke design described above is represented by a the two stroke cycle with piston controlled ports, combined with external scavenging and Gasoline Direct Injection (GDI).

The first reason of interest for this type of machines is the double cycle frequency, allowing the designer to either draw a very compact and light unit for the given power target, or limit the maximum rotational speed, with ensuing advantages in terms of mechanical efficiency, noise and vibrations.

Differently from the rotary engines like the Wankel, a 2-Stroke can easily share many technologies and components with a 4-stroke engine. In particular, a standard automobile GDI system can be adopted. Obviously, the position of the injector and its nozzle geometry must be carefully designed for the specific application. The main issue, in comparison to a 4-Stroke, is the start of injection timing, at full load and high speed: for obvious reasons, injection can't start before bottom dead center, as generally done in conventional automotive engines, so that air-fuel mixing must be much faster. However, in a 2-stroke engine, the injector position and the shape of the combustion chamber are almost free of constraints, since there are no valves on the cylinder head and the head cooling can be much more effective. Furthermore, a 2-Stroke range extender does not need to run over 4500 rpm.

As reported by other authors [3, 4, 5, 6, 7, 8], direct injection allows the 2-stroke engine to address many of the typical issues related to fuel consumption and emissions, even if other solutions have been investigated [9]. At low load and speed, a 2-Stroke GDI engine can work with lean/stratified mixture and a relatively high amount of residuals, thanks to the high turbulence level, the compactness of the chamber and the injector capability to generate a proper stratification of the charge. As a result, all the raw pollutant emissions can be dramatically reduced, while fuel efficiency is improved in comparison to standard partial load operations (penalized by induction throttling). The adoption of a conventional three-way catalyst should allow the complete abatement of CO and HC, while NOx are already very low.

At high loads and speeds, this strategy may not be able to comply with stringent emissions regulations, however the 2-Stroke engine can be set to operate with a stoichiometric mixture within the cylinder. In comparison to a 3-way catalyst installed on a 4-Stroke engine, a worsening of the NOx emission is envisioned, due to the peaks of oxygen concentration in the exhaust flow occurring at the end of the scavenging process (short-circuit cannot be avoided in a 2-Stroke engine, and the instantaneous air excess is expected to freeze the reduction of NOx). The lower is the trapping efficiency, the stronger is this negative effect. However, it should be considered that the formation of NOx in the 2-Stroke engine is limited by the presence of exhaust gas within the cylinder (internal EGR), and that for the most part of the exhaust process the Oxygen concentration in the catalyst is the same of a 4-Stroke engine (short circuit generally occurs only at the end of scavenging).

The presence of air in the exhaust flow can be turned into an advantage during the engine warm-up phase: setting a slightly rich mixture within the cylinder, the additional fuel may burn in the catalyst, strongly reducing the light-off time. Therefore, the catalyst may be placed relatively far from the cylinder, leaving more freedom for fluid-dynamic tuning and packaging.

Finally, it is important to bear in mind that a Range Extender engine is operated only at a few steady points. Therefore, the optimization is much easier, and potentially critical operations for emissions, reliability or fuel consumption may be simply discarded from the set of working points.

For a number of reasons, the main issue that must be addressed on a 2-Stroke engine is the efficiency of scavenging process. Crankcase scavenging seems not suitable for modern automotive applications, because of the lubrication oil dispersed in the fresh charge and bypassing the cylinder, as well as by the increase of friction losses and wear at high load. Conversely, an external compressor enables the use of a conventional and efficient oil sump, leaving a lot of freedom in the choice of the induction and exhaust systems. Most of

externally scavenged engines have Roots-type blowers, but other solutions proved to be reliable over a wide range of operating conditions. As in the Ricardo Dolphin engine [10], or in the more recent ALCOR [11], an additional cylinder can be adopted for scavenging purpose, and also be used to achieve an excellent balance of the crankshaft, as on the V2-90° 4-Stroke engine. Furthermore, piston pumps are generally much more efficient than rotary blowers.

The capability of a 2-Stroke engine to run with only piston controlled ports is another significant advantage, even in front of some downsides. On the one hand, an engine without camshafts and poppet valves is less expensive, more compact and efficient (less friction losses, much more freedom in the design of the combustion chamber and the injector nozzle, generation of a well structured tumble motion within the cylinder, after ports closure). On the other hand, it is difficult to get a good scavenging quality (high trapping efficiency, low residuals), and a number of minor issues arise when designing the piston-rings-liner assembly. A fundamental help for scavenging a cylinder, without poppet valves but in conjunction with a piston pump, can be obtained from a patented rotary valve called "Charge Control Device" (CCD) [12], visible in figure 1.

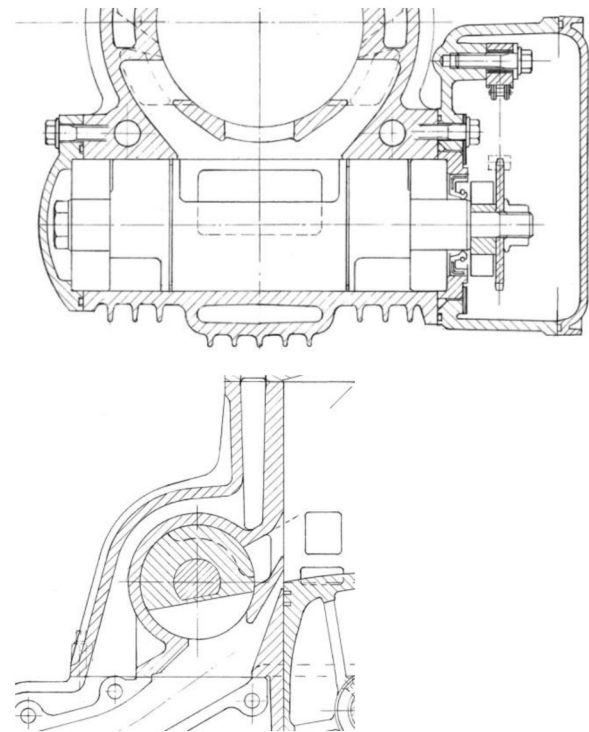


Figure 1. drawings of the patented rotary valve (CCD) for induction control

The valve is made up of a rotor, mounted on roller bearings and revving at the same speed of the engine, controlling the flow upstream of a specific set of transfer ports, called "supercharging ports". The shape of the valve rotor's cross section allows the designer to define the time-area diagram of

the supercharging ports in an almost free manner, enabling the adoption of effective scavenging strategies. In particular, the supercharging ports can remain fully open when the piston is closing the exhaust port (compression stroke), taking the maximum advantage from the action of the piston pump, that is pushing fresh charge toward the engine at the same time. Moreover, the supercharging ports are slightly higher than the exhaust port, in order to provide an additional amount of fresh charge after exhaust closing. In [figure 1](#) it can be also observed that, besides the supercharging ports, there is a set of conventional transfer ports, quite small in comparison to the standard design.

The installation of the CCD device does not generate critical constraints on ports design, that can rely on the well-established engineering practice on 2-S high speed engines. The parasitic loss of the device is negligible, in comparison to a 4-stroke valvetrain, since the load on the bearings is very low and the construction tolerances are not tight.

Finally, CFD-3D analyses on the in-cylinder flow show that the CCD valve further increases the intensity of the tumble vortex within the cylinder, promoting air-fuel mixing and flame speed.

In conclusion, the key features of the 2-Stroke range extender engine analyzed in this study may be summarized as follows:

- V2-90° lay-out and oil sump, as for the 4-Stroke engine;
- external scavenging pump, provided by one cylinder;
- GDI, with full electronic control;
- Piston controlled ports (both scavenging and exhaust);
- Adoption of the patented rotary induction valve CCD;
- 3-way catalyst

[Figure 2](#) shows the 2-Stroke engine concept.

ENGINE SPECIFICATIONS

The main design targets and constraints considered for the two types of engines (2-Stroke and 4-Stroke) are listed below:

- 30 kW at 4500 rpm or lower speed
- V2-90° lay-out
- Maximum combustion pressure: 80 bar
- Maximum Sound Pressure Level (SPL) measured at 0.5 m from the exhaust tailpipe (at 45°): 95 dB(A)
- Maximum temperature at the catalyst inlet: 800 °C
- No knocking

It is observed that the analyzed engines have got the same lay-out and power target, for a straightforward comparison.

The limitation of in-cylinder pressure is a fundamental key for achieving a light design, while the control of the maximum pressure gradient and of the SPL are basic NVH requirements, set upon experience. Finally, the catalyst maximum temperature inlet is set to guarantee the reliability of the after-treatment system.

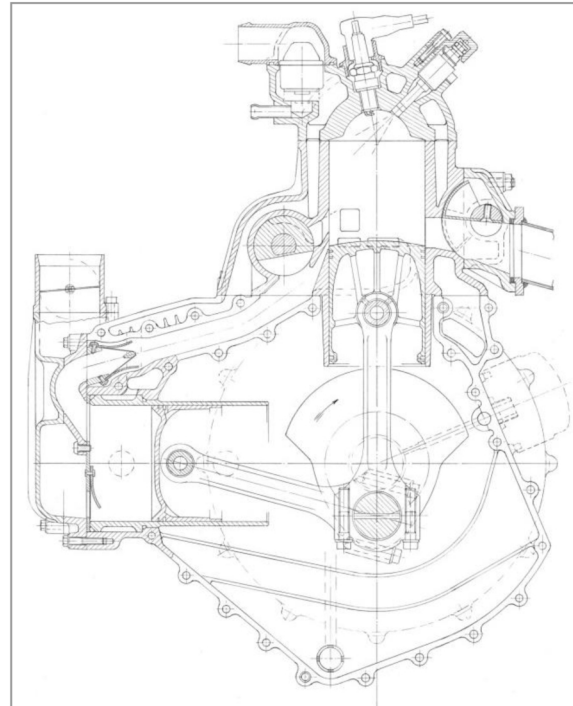


Figure 2. the drawing shows the 2-Stroke range extender engine concept.

4-S ENGINE DESIGN

For this type of engines, the ideal bore to stroke ratio should be close to 0.9 (bore 77 mm, stroke 85 mm), for the following reasons:

- the volume to surface ratio of the combustion chamber is proportional to stroke, thus relatively long strokes are desirable for reducing heat losses;
- on the basis of the authors' experience, at top engine speed, mean piston velocity should be higher than 10-12 m/s, in order to achieve high turbulence levels within the cylinder and to enhance dynamics effects within the intake runners;
- friction mean effective pressures are about proportional to mean piston speed, thus, according to the authors' experience, it is not recommendable to exceed 15 m/s
- as bore decreases, the flame path is shorter: this feature may help to reduce knocking probability
- small bores (<80 mm) generate tighter limitations on valve dimensions, in comparison to medium size chambers (80-100 mm)

As far as the combustion system is concerned, design was oriented toward the well-known Heron chamber (flat head, bowl in the piston), for the following reasons:

- this design is a perfect match for swirl, easily generated by the two valve configuration;
- combustion velocity is supported by the turbulence provided by both swirl and squish
- the chamber is compact and regular, since the flame path is short (assuming a relatively deep bowl, and the spark plug installed close to the cylinder axis), while the valve pockets carved on the piston crown are small (big average clearance between valves and piston, due to the bowl);
- since the geometry is very simple, it could be optimized through a parametric CFD analysis.

The minimum clearance between piston and head (squish height) was assumed 1.0 mm, a typical value for these engines. For the design of the valves and ports, the following constraints are imposed:

- spark plug diameter: 10 mm (for using off-the-shelf items)
- maximum spark plug offset (from cylinder axis): 10 mm (to limit the flame path)
- minimum clearance between valves and spark plug: 2 mm
- minimum clearance between valves and liner: 2 mm
- (Valve max. diameter - seat diameter)= 4 mm
- Minimum clearance between valves and piston at TDC: 2 mm (for avoiding deep valve pockets on the piston crown)

A geometrical study has been carried out in order to determine the maximum dimensions and the corresponding weight of the valves for a bore of 77 mm. Accordingly, two reference valve lift profiles have been determined, from the authors' database, see [figure 3](#). These profiles comply with all typical design constraints, reaching the limit on maximum and minimum valve acceleration. Starting from this basis, the actual valve lift profiles may be obtained according to the following equation [22]:

$$h\left(\frac{\theta}{\Delta\theta}\right) = h_0 \left(\frac{\theta}{\Delta\theta_0}\right) \cdot \left(\frac{\Delta\theta}{\Delta\theta_0}\right)^2 \cdot \left(\frac{D_{v0}}{D_v}\right)^2 \quad (1)$$

where h is the valve lift, θ is the cam angle (starting from 0), $\Delta\theta$ is the angular opening duration; D_v is the valve maximum diameter, the subscript "0" indicates the limit reference profile.

Assuming that the valve mass is proportional to the square maximum diameter, any lift curve generated by [equation \(1\)](#) comply with the same inertia constraints enforced in the reference profile. Furthermore, all the profiles may use the same set of springs. Obviously, any other less "aggressive" profile may also be adopted.

The intake system of the engine must be as compact as possible: this paramount requirement yields a limitation on the maximum length of the intake runners, assumed equal to 200 mm.

Also the length of the primary exhaust pipes is limited to 500 mm, in order to reduce the light-off time of the catalyst. The latter may be packed within the silencer, as done in most of the recent motorcycles.

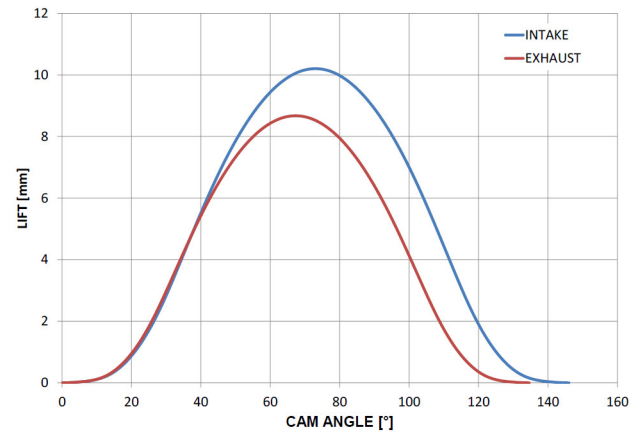


Figure 3. reference valve lift limit profiles

2-S ENGINE DESIGN

The design of the 2-Stroke range extender engine has been already described in some previous papers [13, 14, 15], thus only some fundamentals are reviewed here, for the sake of brevity.

The main goal of the design process is to achieve the performance targets minimizing overall dimensions, weight and construction costs.

First, the geometry of the inlet and exhaust ports and of the cylinder has been defined, in order to achieve a good scavenging quality. In practical terms, scavenging is good when: a) for a given delivery ratio (defined as the ratio of delivered air to a reference mass, the latter calculated as the product of inlet density and cylinder displacement), the trapping ratio (trapped mass to delivered mass) is always higher than the value obtained under perfect mixing conditions; b) the average pressure drop across the cylinder at the maximum delivery ratio is as low as possible. The first aspect mainly depends on the geometric details of ports and ducts, while the second is related to the equivalent mean effective area of the transfer and exhaust ports, referred to the cylinder cross section area, and to the peak mean piston speed [11]. Scavenging optimization required a strong support from CFD-3D simulation, as extensively reported in [15], and it was specifically carried out for a cylinder having a bore of 84 mm and a stroke of 90 mm.

Once the geometric details are defined, the main development tool becomes engine cycle 1D simulation. The keys for the engine optimization are listed below.

1. Proper displacement of the piston pump. As the pump bore increases, also the delivery ratio increases: higher power, but lower trapping efficiency and higher fuel consumption
2. Proper design of reed valves, one set at each end of the piston pump; in particular, a trade-off is required between overall dimensions, affecting the pump compression ratio, and the valve mean discharge coefficient
3. Fluid-dynamic tuning: ports and rotary valve timings, combined with the geometry of the intake and the exhaust system must generate proper conditions within the cylinder, and at the transfer and exhaust ports, in order to maximize the amount of trapped air at 4500 rpm, WOT
4. Charge cooling between pump and engine cylinder, in order to prevent knocking issues.
5. Exhaust valve (see [figure 2](#)) for the control of the exhaust effective area. This device helps to reduce the amount of fresh charge by-passing the cylinder, improving both engine performance and the catalyst conversion efficiency, under stoichiometric operations.

The last phase of engine design requires again the support of CFD-3D simulation, for the optimization of fuel spray and combustion. As reported in [15], the geometry of the combustion chamber and the positions of the injector and of the spark plug are almost free of constraints: these conditions allow the designer to define an almost ideal combustion chamber, with the injector close to the spark plug, both in the center of the chamber. Such a lay-out, supported by a very high turbulence level, enables an excellent control of the charge stratification: a stable and efficient combustion was observed for values of relative air-fuel ratio in the trapped charge even higher than 1.5.

Particular care must be devoted to find the optimum injection strategy at full load, high speed. A homogenous mixture seems the best option, not only to achieve a high combustion efficiency, but also to prevent the formation of hot spots, causing knocking. In comparison to a 4-Stroke engine, mixing time is much shorter, but the specific flow field within the cylinder, combined with the injector nozzle geometry, enable the adoption of a relatively advanced injection (close to BDC), without losing fuel from the exhaust.

ENGINE CFD SIMULATION

The development of both engines has been supported by CFD simulation, both 1D (GT-Power) and 3D (KIVA). In this paper only the 1D results will be presented.

A 2-S engine prototype has been recently built and tested at the BRC laboratories in Cherasco (Italy), as reported in [15]. The experimental results are used here to calibrate the GT-Power model, as shown in [Appendix 1](#). Starting from this calibrated model, a new improved configuration has been developed.

As far as the 4-stroke engine is concerned, the GT-Power model was built on the basis of empirical hypotheses, supported by a few 3D analyses.

DISCHARGE COEFFICIENTS

The values measured at the steady flow bench on a similar 4-Stroke engine (SI, 2-valve) have been entered, see [figure 4](#)

FMEP

The Chen-Flynn correlation [17] has been used to model the mechanical losses:

$$FMEP = A + B \cdot P_{max} + C \cdot U_p + D \cdot U_p^2$$

(2)

Where A, B, C and D are calibration parameters, P_{max} is the average in-cylinder max. pressure, U_p is the mean piston speed. For matching the experimental data of the authors, the best set of parameters is:

$A = 0.3$ bar; $B = 0.006$; $C = 0.09$ bar*s/m; $D = 0.0$

COMBUSTION

The predictive combustion model included in the 7.4 release of GT-Power has been used. A calibration has been performed by means of a comparison with cfd-3d results. For the sake of brevity, the model calibration is presented in [Appendix 2](#).

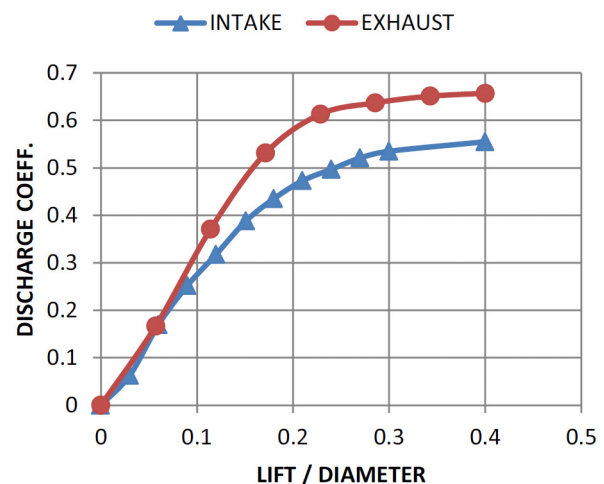


Figure 4. experimental discharge coefficients plotted versus non dimensional valve lift for a 2-valve SI engine. Reference area: max. valve diameter

4S ENGINE OPTIMIZATION

The 4S engine has been optimized by means of a multiple factor DOE analysis [16]. Three operating points (2000, 3000 and 4500 rpm), have been selected for the 1D engine cycle simulation, using GT-Power. The optimization goal was the minimum fuel consumption, as an average among the 3 cases.

A total of fourteen parameters has been investigated, however each analysis was run with a maximum of seven factors, to optimize the computational efficiency. The first set of DOE parameters are:

1. Intake valve timing (from 440 to 500 cad after TDC)
2. Exhaust valve timing (from 220 to 280 cad after TDC)
3. Intake lift multiplier (from 0.8 to 1.0)
4. Exhaust lift multiplier (from 0.8 to 1.0)
5. Crank angle at which 50% of charge has burnt (from 5 to 26 cad after TDC)
6. Compression ratio (from 9 to 13)
7. Throttle opening (from 50 to 100%)

This set of parameters was chosen for their well-known influence on BSFC and knocking and brake performances. In particular, the compression ratio was expected to have the strongest impact on both fuel efficiency and knocking tendency, balanced by ignition timing and throttle opening; the valve parameters are paramount for achieving the power target. The parameters range is defined on the basis of a set of preliminary runs, which were essential to keep computational cost at a reasonable level.

Once the optimum value for the first seven parameters was found, a further set was considered:

1. Intake valve diameter (from 32 to 36 mm)
2. Exhaust valve diameter (from 24 to 28 mm)
3. Intake lift multiplier (from 0.8 to 1.0)
4. Exhaust lift multiplier (from 0.8 to 1.0)
5. Intake runner main diameter (from 30 to 60 mm)
6. Intake runner length (from 170 to 290 mm)
7. Exhaust pipe length (from 100 to 400 mm)
8. Exhaust pipe diameter (from 30 to 60 mm)
9. Stroke (from 77 to 85 mm)

As visible, intake and exhaust lift multipliers have been considered in both sets of parameters, because of the correlation with valve diameter, expressed by [equation \(1\)](#). Furthermore, bore has been kept constant (77 mm), since its influence on the valves dimension and the geometry of the

combustion chamber is quite complex to be analytically determined. It should be noted that the maximum values of the valves diameter are calculated in order to comply with the geometrical constraints, while the influence on discharge coefficients is neglected.

Combining all the DOE analyses, it was possible to identify the best setting for the parameters, while complying with all the constraints. The most stringent limits were found to be the ones on power rate, knock and catalyst temperature. [Table 2](#) reviews the final results of the whole DOE analyses, carried out at 2000, 3000 and 4500 rpm.

Table 1. Best values of the 4-S engine parameters, optimized by using DOE at 2000, 3000 and 4500 rpm

1. Intake valve timing (c.a. at max. lift)	450°
2. Exhaust valve timing (c.a. at max. lift)	250°
3. Intake lift multiplier	1
4. Exhaust lift multiplier	1
5. 50% burnt angle (thb_50)	Variable with rpm
6. Compression ratio	12:1
7. Throttle opening	100%
8. Intake valve diameter	36 mm
9. Exhaust valve diameter	28 mm
10. Intake runner main diameter	40 mm
11. Intake runner length	170 mm
12. Exhaust pipe length	100 mm
13. Exhaust pipe diameter	35 mm
14. Stroke	81 mm

As expected, the first seven parameters are the most important ones, and the simulation shows a robust tendency to the values above.

The reference lift profiles and the maximum valves diameters are found to be suitable for the specific application. A reduction of their values does not guarantee the compliance with the power target, due to the ensuing drop of volumetric efficiency. As far as the valves timings are concerned, it appears to be convenient to adopt small closing retards of the intake valve (about 50°), along with small advance of exhaust opening (about 60°). This tendency is explained by the low max. engine speed (4500), that does not require large exhaust advance for pushing out the spent charge, and, at the same time, it makes the ram effects so weak that the intake valve must close early, to avoid a strong backflow. It is observed that the overlapping period is unusually large, if compared to standard automobile