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Development of a 2-Stroke GDI Engine

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Abstract

Nowadays, high-pressure gasoline direct injection (GDI) can be considered a standard technology, due to the wide application on 4-stroke turbocharged engines. This technology – in combination with other specific solutions - has been successfully applied to a 500 cc, 30 kW 2-stroke engine, initially developed as a range extender. The engine is valve-less and cam-less, being the scavenge and exhaust ports controlled by the piston. An electric supercharger delivers the required airflow rate, without need of a throttle valve; the lubrication is identical to a 4-stroke. The current study reviews the development process, assisted by CFD simulation, that has brought to the construction of a prototype, tested at the dynamometer bed of the University of Modena and Reggio Emilia (Engineering Department “Enzo Ferrari”). An experimentally calibrated CFD-1d model is applied to predict full load engine performance. The results show an excellent fuel efficiency and a very low level of thermal and mechanical stress despite the high power density.

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Keywords: 2-stroke cycle; GDI; CFD; range-extender; electric supercharger

1. Introduction

One of the most critical challenges of recent electric cars is the driving range achievable with the current battery technology, which allows the vehicle to store only a small fraction of the energy provided by conventional fuels, such as gasoline or diesel. An effective way to address the so called “range anxiety” is the adoption of a range

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extender, that is a small internal combustion engine designed to recharge the traction batteries when their available energy content drops below a specified threshold [1,2].

The basic requirements for the range extender are:

- peak power sufficient to allow the car to reach a target highway speed (typically, 120 km/h), with empty batteries
- compactness and lightness, in order to permit an easy and not expensive installation on a conventional vehicle;
- excellent 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, a range extender can be directly derived from a mass production engine. Conversely, for city cars requiring 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, generally adopting a conventional 4-stroke cycle [3-6].

The two-stroke cycle, with a new specific design, represents an unconventional but not exotic alternative to 4-Stroke. The main 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. This concept has been explored by the manufacturer PRIMAVIS, who proposed a first engine version, rated at 30 kW at 4500 rpm, weighting about 35 kg. This project is described in several previous technical papers [7-12], so that in the next section only the fundamentals will be recalled. The current paper focuses on the development of the new prototype, featuring several improvements, in particular the adoption of an electric supercharger instead of a piston pump

2. The new prototype by PRIMAVIS

This section reviews the design process of the proposed 2-Stroke engine. The main differences from conventional small motorcycle engines are: direct fuel injection (no fuel losses through the exhaust ports, full control of the air-fuel ratio, capability to stratify the charge, then to run under lean conditions); lubrication as in 4-strokes (no oil dispersed in the exhaust flow, good lubrication at any operating condition); external air metering instead of the crankcase pump (much better control of the airflow, no need of a throttle valve); adoption of a patented device to improve the scavenging quality. In terms of pollutant emissions, the proposed 2-stroke can adopt a 3-way catalyst, like a 4-stroke, being free from the typical issues affecting small motorcycle engines (dispersed oil, fuel burning in the exhaust pipes, et cetera). The only concern for pollutants is related to NO_x emissions, since the air by-passing the cylinder at the end of the scavenging process may “freeze” the reduction of Nitrogen oxides. However, the concentration of this pollutant may be strongly reduced in the raw gas (before the catalyst), thanks to the control of exhaust gas trapped within the cylinder (internal EGR), as well as by adopting specific combustion strategies [13-15]. The power target of the engine is 30 kW (64 Nm) at 4500 rpm.

2.1. Layout

The first step of the project, started in 2012, was the choice of the scavenging type. Crankcase scavenging was immediately discarded, because of the lubrication oil dispersed in the fresh charge and bypassing the cylinder, as well as for 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.

As in the Ricardo Dolphin engine [16], or in the more recent ALCOR [17], an additional cylinder can be adopted for pumping air from ambient to the power cylinder; the second piston also enables an excellent balance of the crankshaft, as occurs in a V2-90° 4-Stroke engine. In 2014, PRIMAVIS built and tested a first prototype adopting a piston pump with different types of automatic valves installed in the pump head. The prototype is visible in figure 1.

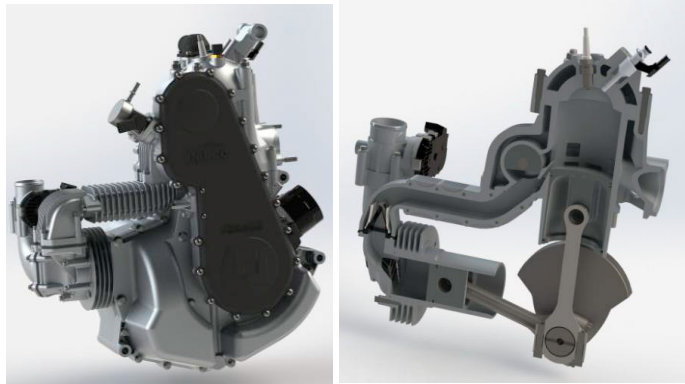


Figure 1: Views of the first prototype built by PRIMAVIS

Unfortunately, the experimental campaign carried out on the above mentioned prototype revealed that automatic valves are quite critical, in terms of reliability. Therefore, in 2016 it was decided to move from a piston pump to an electric supercharger, as the ones recently adopted to support conventional turbocharging [18-20]. When the required power is low, these machines can be very compact; furthermore, they don't need an additional battery pack, being a big one already installed on the electric vehicle.

2.2. The Patented Valve

The capability of a 2-Stroke engine to operate with piston controlled ports is a significant advantage in terms of compactness and cost, but it also yields some downsides, mainly associated to the symmetrical port timing and to risk of short-circuit of fresh charge across the cylinder. A fundamental help to improve the scavenging process in a cylinder with piston controlled ports can be obtained from a rotary valve called "Charge Control Device" (CCD) patented by PRIMAVIS [21]. The valve, visible in figure 2, is made up of a rotor, mounted on roller bearings and revving at the same speed of the engine. It controls the flow through a set of auxiliary transfer ports opposite to the exhaust ports, and it allows the designer to define the time-area diagram in a quite free manner. In particular, it is possible to adopt asymmetric timings, able to increase the amount of fresh charge stuffed within the cylinder, while contemporarily reducing the short-circuit. It is important to notice that the installation of CCD does not generate critical constraints on ports design, that can rely on the well-established engineering practice on 2-S high speed engines [17, 22]

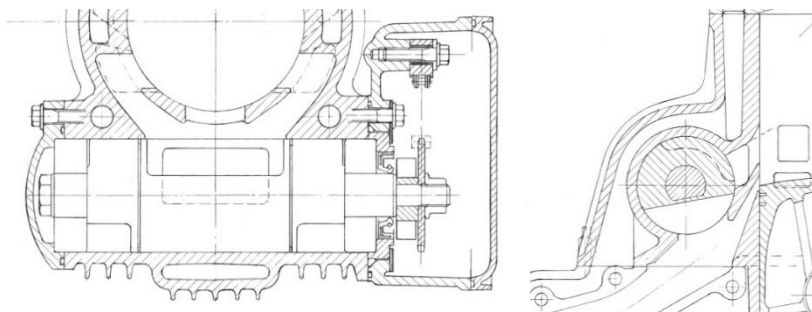


Figure 2: installation of the CCD valve [21]

2.3. Cylinder Design

The optimization of the scavenging and combustion system is the most critical part of the project, and it requires a strong support from both CFD simulation and experiments. CFD-simulation plays a key role to address the design process, while keeping the development cost under control. The followed approach is presented in figure 3.

A first cylinder design is defined on the basis of standard engineering practice, and refined by means of an iterative process, including both 1D and 3D CFD simulations. For 1D engine thermo-fluid-dynamic numerical analyses, a well assessed commercial software is used (GT-Power), while for 3D calculations a customized version of the KIVA-3V code is employed. This software was already used in a number of previous similar projects [23-26] Two types of multi-dimensional investigations are required, the former focused only on the scavenging process, the latter encompassing the whole engine cycle, including spray and combustion.

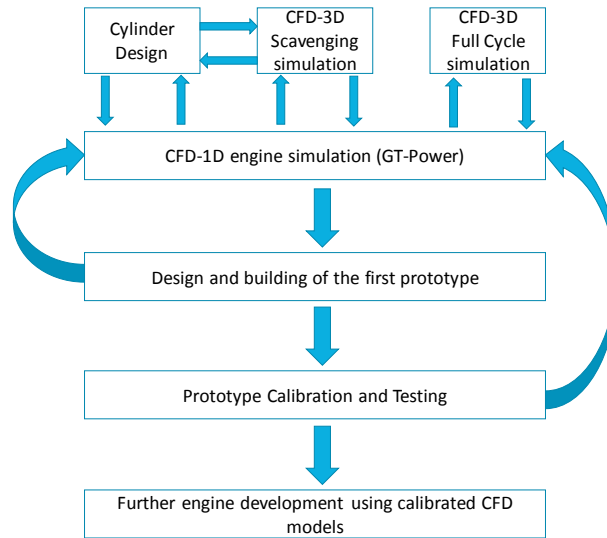


Figure 3: sketch of the design process

Once the ports geometry is “frozen”, thanks to the guidance provided by the CFD-3D scavenging analyses, full cycle CFD-3D analyses are employed to investigate the spray and combustion processes. Again, the support of GT-Power is fundamental, in order to provide accurate boundary conditions when the ports are open.

The most critical operating condition is at maximum engine speed (4500 rpm), full load: here, the time available for fuel evaporation and mixing is minimum, and the amount of fuel is the maximum one. In order to guarantee a regular combustion, it is necessary to achieve a homogeneous air-fuel mixture at about 20° before Top Dead Center. Failing this goal means not only a poor combustion efficiency, but also a high risk of knocking (where mixture is lean, temperature is generally higher, and auto-ignition may occur more easily). The composition of the charge before TDC depends on a huge number of variables: engine speed, delivery ratio, injector nozzle geometry and position, injection strategy (pressure and timing), air-fuel ratio... In order to limit these variables, only one commercial injector has been considered (Magneti Marelli RDI 206) and the influence of injection strategy has been analyzed only at the most critical operating conditions. It was found that the proper location of the injector, along with the high injection pressure (200 bar), enable a uniform distribution of fuel vapor within the cylinder, without any loss at the exhaust.

As an example of full cycle CFD-3D simulation, Figure 4 shows the distribution of fuel concentration (left column) and gas temperature (right column) on a cross section of the cylinder, around top dead center, calculated at 4500 rpm, full load. First of all, it may be observed that the charge composition at Start of Ignition is almost perfect; moreover, combustion is very fast, thanks to the high values of turbulence (a strong tumble vortex is automatically generated during loop scavenging, squish also helps) as well as to the compactness of the chamber (spherical shape, almost central spark plug). The compactness of the chamber is a consequence of the lack of constraints, leaving total freedom to the designer.

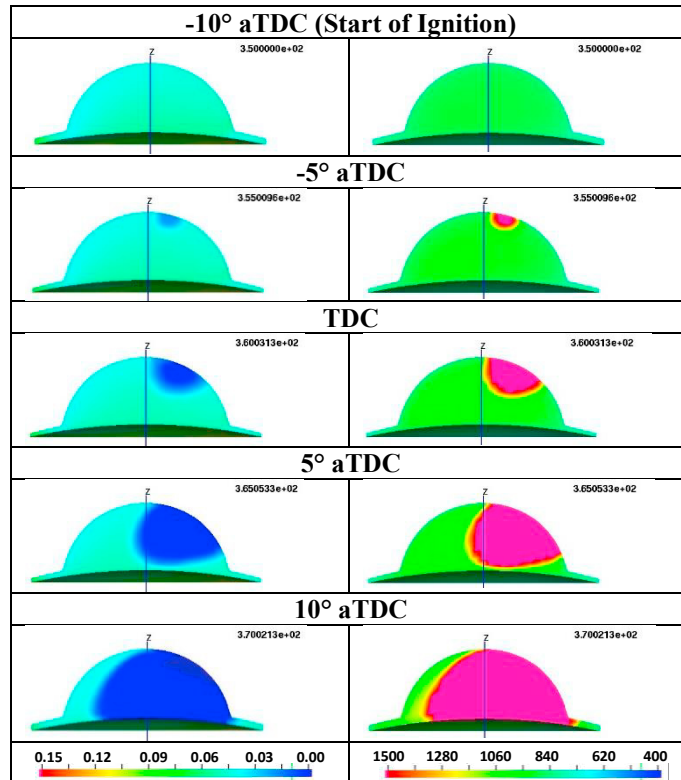


Figure 4: Simulation of the combustion process at 4500 rpm, full load (Fuel concentration and gas temperature [K])

2.4. Development of the Prototype

As already mentioned, after the experimental campaign on the first prototype, presented in figure 1, some modifications were introduced. In particular, an electric supercharger replaced the previous piston pump; automatic valves are not required anymore; load is now controlled by the speed of the electric supercharger (no throttle valve). A view of the new prototype is visible in figure 5.

The new version of the engine has been built and tested at the Department of Engineering “Enzo Ferrari” The experimental facility has been comprehensively described in previous papers [27-28], so that only the fundamentals will be reviewed here. The testing bed features an Apicom FR 400 BRV eddy-current brake and the Apicom Horus software for system control and data acquisition. Besides the standard pressure and temperature transducers, the laboratory instruments also include flow meters for measuring air and fuel consumption, and a Lambda sensor. A high frequency indicating system has been specifically designed and installed in order to record in-cylinder pressure traces; the system is made up of a Kistler piezoelectric transducer, integrated in the spark plug, a charge amplifier and an optical encoder. Pressure and phase signals are processed by means of the NI CompactRIO hardware and the Alma Automotive software (Obi).

As done in the previous experimental campaigns, also in this case the measured data are used to improve the accuracy of the GT-Power model representing the engine in the current configuration. The comparison between experimental and computational results is already shown in other papers, thus it will not be repeated here. The full load performance of the new prototype, predicted by using this calibrated model, are shown in figures 6-7. Trapped air-fuel ratio is set at the stoichiometric value within the cylinder, while the supercharger speed is regulated in order to maintain a constant charging efficiency (ratio of trapped air mass to reference mass at ambient condition) close to

0.6. In this way, the indicated torque is about constant, while the brake power reaches the target of 30 kW. The electric supercharger enables many other strategies: as just one example, the delivery ratio can be controlled in order to maintain a constant fraction of exhaust residuals within the cylinder (internal EGR). Figure 6 presents the plots of: volumetric efficiency (or delivery ratio), charging efficiency, fraction of residual gas within the cylinder, imep, bmep, fmep, and the specific work lost to drive the electric supercharger (auxiliary mean effective pressure). All the parameters are plotted as a function of engine speed.

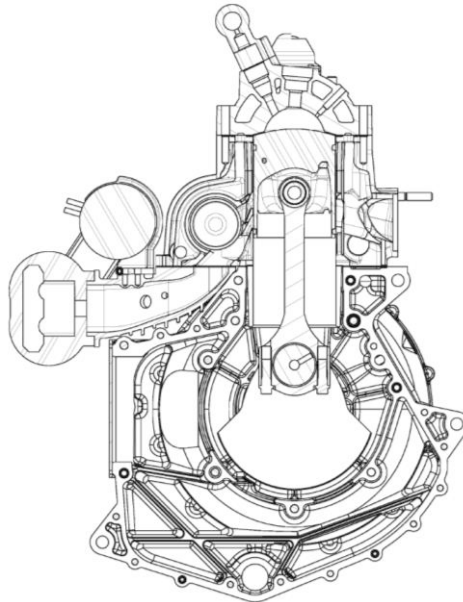


Figure 5: Drawing of the new prototype with the electric supercharger

In figure 6, it is observed that volumetric efficiency (of delivery ratio [17]) decreases as engine speed goes up, in order to maintain a constant value of volumetric efficiency of trapped air (or charging efficiency [17]). The reason for trapping efficiency (ratio between charging efficiency and delivery ratio [17]) is monotonically dependent on engine speed can be found in the total lack of dynamic effects (the exhaust system is not tuned, and the maximum engine speed is relatively low). As a result, the longer the ports remain open, the larger the fraction of fresh charge lost through the exhaust. Not surprisingly, the higher values of trapping efficiency correspond to the maximum values of residuals within the cylinder. The specific indicated work (imep) is quite constant (between 8 and 9 bar), since the amount of trapped charge is kept constant. However, heat losses have a larger weight at low engine speed, thus the curve goes up as engine speed increases. It should be noted that the values of imep are very low, in comparison to a 4-stroke engine with a comparable power density; thermal loads are then lower, with ensuing advantages in terms of reliability and lightness of the engine construction. The effective specific work available at the engine crankshaft is the difference between imep and the sum of friction and auxiliary mean effective pressures. The auxiliary loss is the energy required to drive the electric supercharger, calculated considering the isentropic efficiency of the compressor, as well as the efficiency of the electric motor (0.95) and of the transmission (0.9). The bmep curve is shifted downward from imep of an almost constant quantity (about 1 bar). It should be noted that the supercharger has a stronger impact at low engine speed, since a higher volumetric efficiency is required here to compensate the low values of trapping efficiency. The most critical point occurs at 3000 rpm: charging efficiency is slightly lower here, because of the limit on the electric power adsorbed by the supercharger.

Figure 7 shows brake power, brake specific fuel consumption and the electric power required by the supercharger. First, it is observed that the peak power target of 30 kW at 4500 rpm is reached easily, requiring an electric power of less than 0.75 kW. The limit on the electric power is set at 1.2 kW, in order to have a compact and light supercharger. The supercharger hits the limit only at 3000 rpm. Specific fuel consumption is very good, in particular at high speed. Again, the main reason is the smaller losses due to the supercharger and heat transfer.

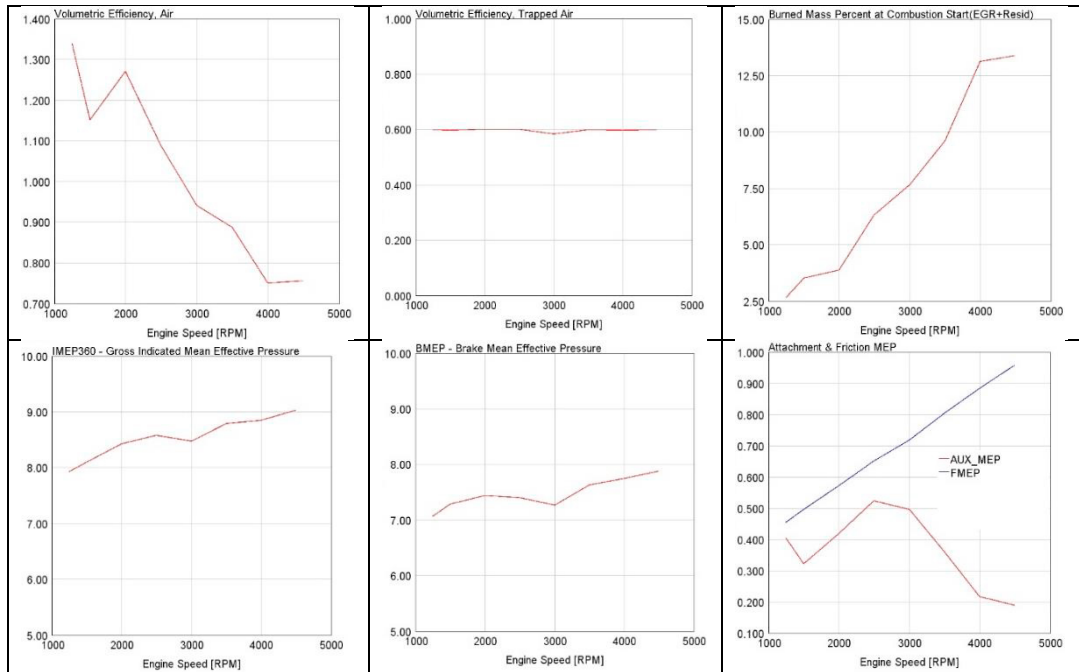


Figure 6: Full load engine performance predicted by GT-Power

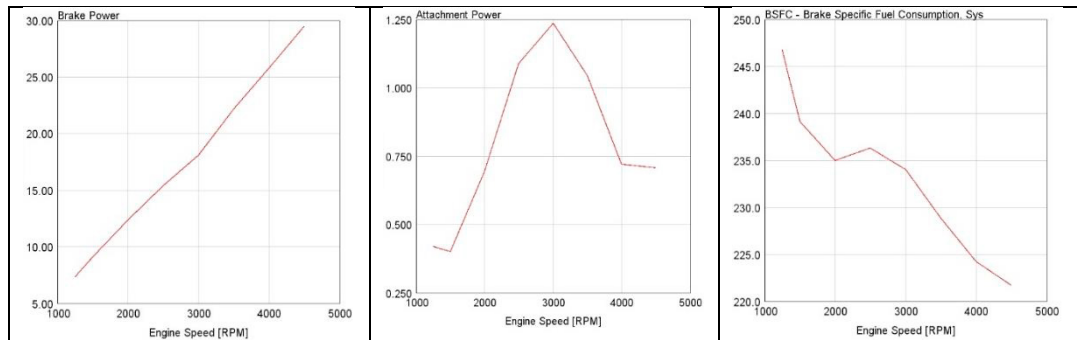


Figure 7: Full load engine performance predicted by GT-Power

3. Conclusions

The paper reviews the development of a 2-stroke GDI cam-less and throttle-less engine, able to provide a specific power of 60 kW/l, along with excellent fuel efficiencies (222 g/kWh as best point). Air is delivered by a small electric supercharger (maximum electric power: 1.2 kW). The maximum value of imep is just 9 bar. The engine can be coupled to a three-way catalyst, thanks to the absence of lubricant oil in the exhaust flow (lubrication is identical to a 4-stroke engine) and to the total control of the trapped charge enabled by the electric supercharger and by a patented inlet valve. The engine has been initially developed as a range-extender, but it can find also other applications where compactness, light-weight, low noise and high efficiency are fundamental. A prototype of the engine is currently under testing at the Department of Engineering “Enzo Ferrari”, in Modena (Italy).

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