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Optimization of a Supercharged Single Cylinder Engine for a Formula SAE Racing Car

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ABSTRACT

The paper reviews the development and optimization of a SI high performance engine, to be used in Formula SAE/Student competitions. The base engine is a single cylinder Yamaha 660cc motorcycle unit, rated at about 48 HP at 6000rpm. Besides the reduction of engine capacity to 600cc and the mounting of the required restrictor, mechanical supercharging has been adopted in order to boost performance.

The fluid-dynamic optimization of the engine system has been performed by means of 1D-CFD simulation, coupled to a single-objective genetic algorithm, developed by the authors. The optimization results have been compared to the ones obtained by a well known commercial optimization software, finding a good agreement.

Experiments at the brake dynamometer have been carried out, in order to support engine modeling and to demonstrate the reliability of the optimization process.

INTRODUCTION

As well known, Formula SAE and its equivalent in the UK (named Formula Student) is a competition among University students who have to design and build and compete with a car complying with specific rules.

As far as the engine is concerned, the technical regulations may be summarized as follows [1].

One or more four stroke engines running on gasoline or E85 may be used, with a total displacement not exceeding 610 cc in all, and the air for all engines must pass through a single air intake restrictor. This restrictor must be of circular shape, and placed between the throttle and the engine(s). The maximum restrictor diameter is 20 mm for gasoline fueled cars and 19 mm for E85. Any device that has the ability to throttle the engine downstream of the restrictor is prohibited.

Turbochargers or superchargers are allowed, but only if the competition team designs the application (i.e. engines originally equipped with a turbocharger are not allowed to compete with the turbo installed). The restrictor must be placed upstream of the compressor but after the carburetor or throttle valve. The intake air may be cooled with an intercooler. Only ambient air may be used to remove heat from the intercooler system. Air-to-air and water-to air intercoolers are permitted.

The car must be equipped with a muffler in the exhaust system to reduce noise to an acceptable level, that will be measured during a static test. Measurements will be made with a free-field microphone placed free from obstructions at the exhaust outlet level, 0.5 m from the end of the exhaust outlet, at an angle of 45 degrees with the outlet in the horizontal plane. The test engine speed for a given engine will be the velocity that corresponds to an average piston speed of 914.4 m/min. The maximum permitted sound level is 110 dBA, assessed through a fast weighting technique.

The throttle must be actuated mechanically, i.e. via a cable or a rod system. The use of electronic throttle control (ETC) or "drive-by-wire" is prohibited.

Water-cooled engines must only use plain water, glycol-based antifreeze or water pump lubricants of any kind are strictly prohibited. No agents other than fuel and air may be induced into the combustion chamber.

Most of the competitors adopt 4-cylinder in-line engines, derived from road motorbikes, but in the last years single cylinder engines, typically taken from Cross and Endurance bikes, are gaining popularity [2]. On the one hand, multi-cylinder engines rev at high speed and may easily provide high values of peak power (up to 80-100 HP, when restricted), but they are generally heavy and large. Furthermore, low end torque is difficult to be obtained when the engine is tuned at high revving speed. On the other hand, single cylinder engines are simple, compact and light (30 kg may be saved only for the power-train), but they cannot reach the same level of power. This result is due to several factors. First, off the shelf single cylinder engines with high values of specific power (exceeding 100 HP/l) are generally designed for Cross motorbikes, where the typical capacity is 450 cc, and bore size is close to 100 mm. Since bore is so large, an increase of displacement can be obtained only by enlarging the crank radius, thus reducing top revving speed. Therefore, the crankshaft modification does not enhance performance very much, while it boosts costs and failure probability (it is reminded that all the modifications must be designed by students). Second, single cylinder engines are affected by stronger pressure and velocity pulses across the restrictor, in comparison to multi-cylinder engines (because of the lower frequency of the intake process). As a result, mass flow chocking occurs at lower speeds, limiting engine revolutions.

A possible way to overcome the limitations of the single cylinder engine, while preserving its advantages, is the use of supercharging [14,16,17]. Turbocharging is penalized by the throttle that must be placed at the compressor inlet. Conversely, a mechanical compressor does not stall when throttled, and it has the further advantage of providing a boost pressure almost independent on engine speed. As a result, low end torque may be very high, an issue particularly appreciated by drivers on winding tracks, such as the ones where Formula cars have to compete.

Obviously, matching a supercharging system to an existing engine is not a task to be taken lightly, particularly by a team of students. Besides the design of the gas-dynamic system, a critical issue is the increase of thermal and mechanical stress within the cylinder, as well as on the whole drivetrain. From this point of view, an off-the-shelf Cross engine is not very suitable as a baseline, being designed to endure high revving speed but relatively low IMEP and torque. Endurance motorbike engines, with a capacity of 600-650 cc are, in the authors' opinion, a straighter choice, being generally much heavier and less sophisticated than Cross power units, but also stronger.

The paper reviews the development and optimization of a Formula SAE supercharged single cylinder engine, carried out by the team of the University of Modena and Reggio Emilia. The base engine is a Yamaha 660cc motorcycle unit, rated at 48 HP at 6000rpm, without restrictor (figure 1). Because of the capacity limits enforced by technical regulations, the engine displacement had to be reduced to less than 610 cc. It was decided to decrease the crank radius, and to design a new and longer connecting rod. In this way, no modification on the engine head, cylinder block or crankcase is required, while higher revving speed may be reached. It is also observed that the reduction of displacement for a given combustion chamber volume makes compression ratio slightly decrease (from 10 to 9.3), without further modifications on the combustion chamber or on the head gasket.

Supercharging has been achieved by means of a small Roots compressor (390cc, 5 kg) driven by a belt, and an inter-cooling air-air system.

The injection system too has been modified, for meeting the higher fuel rates required by supercharging. In order to enhance gasoline atomization within the airflow, the injector has been twinned, and each fuel jet has been carefully oriented toward the valve back (one injector per intake valve).

The fluid-dynamic optimization of the engine system has been performed by means of 1D CFD simulation, coupled to a single-objective genetic algorithm, developed by the authors. The optimization results have been compared to the ones obtained by a well known commercial optimization software.

Experiments at the brake dynamometer have been carried out, in order to support engine modeling and to demonstrate the reliability of the optimization process.

Figure 1: The base engine (Yamaha XT660 R)

ENGINE MODEL VALIDATION

A 1D CFD model of the original engine (Yamaha XT660 R) has been built by using GT-Power [3], a commercial software released by Gamma Technologies and extensively used by the authors.

For the sake of brevity, only the more relevant issues of the model will be described in the paper. Since no specific data were available about combustion, friction losses and the flow through the valves, information has been derived from a database on engines, and with the help of some empiricism. A lot of care has been devoted to the modeling of the muffler, represented as a net of 21 volumes and 10 series of orifices (tail noise is considered a performance parameter, since it is limited by technical regulations).

The calibration of the model parameters has been carried out by comparison with experiments at the dynamometer on the base engine, in the original configuration. The more relevant measured quantities are: engine torque, speed, fuel rate and Air-Fuel Ratio (by means of a UEGO sensor).

After this calibration, the model has been updated to represent the modifications required by the Formula SAE regulations, i.e. with the 20 mm restrictor and the total displacement reduced to 610 cc. Again, a comparison has been made between the results provided by simulation and the values measured at the dynamometer, finding a very satisfactory agreement (this comparison is shown in figure 2).

Obviously, the validated GT-Power model of the naturally aspirated Formula SAE engine is the base on which the model of the supercharged engine has been built.

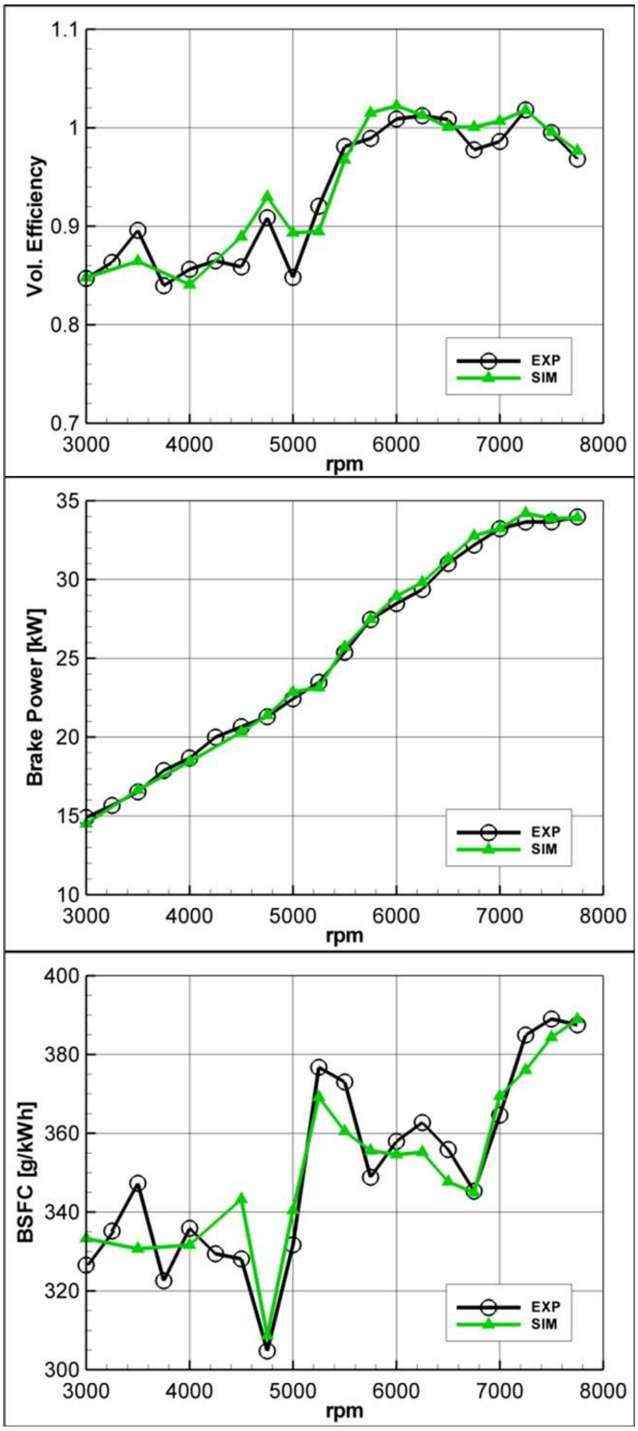

Figure 2: Model validation of the naturally aspirated Formula SAE engine

ENGINE PARAMETRIC DESIGN

The fluid-dynamic optimization of a supercharged Formula SAE engine is far from trivial, since many conflicting issues must be matched.

First, maximum boost pressure and top revving speed should be accurately defined, in order to meet performance and reliability and fuel efficiency. It is reminded that in Formula SAE, high speed operations at Wide Open Throttle (WOT) are important only in the acceleration test, while they have a very limited use on the track, both in autocross and in endurance competitions (beside the circuit is very windy, the car must also slalom among cones, therefore the engine runs throttled most of the time). The adopted design strategy is to keep compressor delivery pressure in a low-medium range (about 1.5 bar, absolute), but let the engine rev up to 7500 rpm. In this way a high power peak may be reached for a few seconds when needed, while under average operating conditions the engine should be safe against knocking. Furthermore, engine top rotational speed is very easy to be electronically controlled (through the advance map in the ECU), while boost pressure would require a specific valve.

Low delivery pressure helps fuel economy too, for two reasons: 1) compressor isentropic and volumetric efficiencies are generally better than at high pressure ratios; 2) as a thumb's rule, the higher is boost pressure, the richer must be the air-fuel mixture, in order to keep in-cylinder temperature under control.

As well known, for a mechanical supercharging system with intercooler, delivery pressure depends on: ratio of compressor capacity to engine capacity, transmission ratio between engine and compressor, ratio of compressor volumetric efficiency to engine volumetric efficiency (referred to intake port pressure) and intercooler outlet temperature. Therefore, once the choice of compressor and target delivery pressure and intercooler outlet temperature is made, on the base of a number of design constraints, the value of transmission ratio is subsequent.

Another critical issue is the trade-off between flow-dynamic tuning and engine transient response. On the one hand, large volumes are suitable to get a proper tuning of the intake system (in particular, a Venturi nozzle is necessary after the restrictor to recover some kinetic energy; furthermore, a large plenum should be placed between the throttle and the engine, in order to dump the pressure and velocity pulses through the restrictor and to provide an open end reflection to the waves traveling along the intake runner). On the other hand, the higher is the volume between the throttle and the engine, the slower is transient response (and the more difficult is injection parameters calibration). The University team has found a very good trade-off between these issues by adopting an unusual configuration for the plenum. Instead of using a single large volume, two Helmholtz resonators are connected to a short intake runner, as visible in figure 3. Even on a single-cylinder naturally aspirated engine, the influence of these resonators on the intake system dynamics is very complex, since the two pipes (intake duct and air scoop) and the two volumes (cylinder and resonators) form a vibrating system, with two degrees of freedom and two resonant frequencies [19].

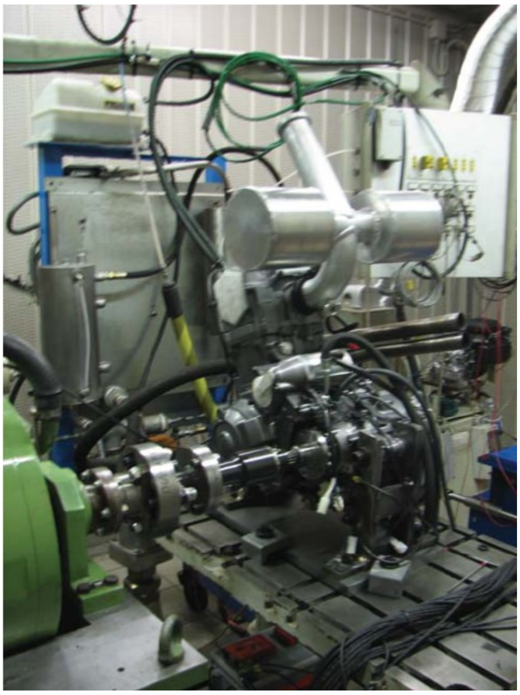

Figure 3: View of the Formula engine with the distinctive plenum made up of two Helmholtz resonators.

A first attempt to tune this system may be done by referring to the electrical analog theory, in which capacitors represent volumes and inductors pipes. A sketch of the model is depicted in figure 4. It is observed that in a supercharged and inter-cooled engine, the upstream duct corresponds to the pipe between the compressor outlet and the resonators.

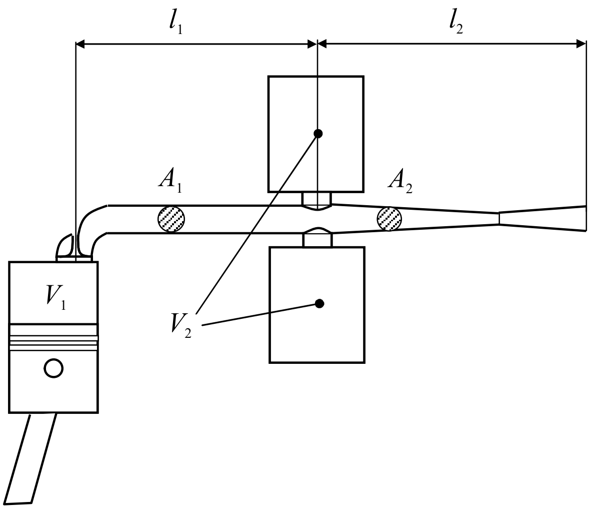

Figure 4: Sketch of the Helmholtz resonator model

According to the electrical analog [20], the equation for the calculation of the resonant engine speed is:

$$
\begin{aligned}
N_{1,2} &= X_{1,2} N_0 \\
X_{1,2} &= \sqrt{\frac{(\alpha\beta + \alpha + 1) \mp \sqrt{(\alpha\beta + \alpha + 1)^2 - 4\alpha\beta}}{2\alpha\beta}} \\
N_0 &= \frac{a}{2\pi k} \sqrt{\frac{1}{L_1 V_1}}
\end{aligned}
\tag{1}
$$

where:

$$\alpha = \frac{l_2 A_1}{l_1 A_2}\ ;\ \beta = \frac{V_2}{V_1}\ ;\ L_1 = \frac{l_1}{A_1}\ ;\ k \cong 2$$

and the average cylinder volume is assessed through:

$$V_1 = V_d \frac{(r_c + 1)}{2(r_c - 1)}$$

where V_d is the engine displacement and r_c the compression ratio.

It may be observed that, when adopting a conventional intake system (made up of an inlet runner, a large plenum and an air scoop), the highest engine speed at which the tuning occurs is almost coincident with N_0, while the lowest speed falls out of the range of interest. Conversely, an intake system with two small Helmholtz resonators yields two tuning peaks, at both low and high engine speed, and presents the further advantage of a faster transient response, due to the reduced volume between throttle and cylinder.

The authors acknowledge the strong limits in the accuracy of equation (1), which can be used only as a starting point for the development of the intake system. However, the experiments carried out by the University team at the dynamometer and on the track, have fully confirmed the effectiveness of this solution.

A number of constraints is placed upon the dimensions of pipes and volumes. According to the technical regulations, all parts of the engine air and fuel control systems must lie within the surface defined by the top of the roll bar and the outside edge of the four tires. Furthermore, the exhaust pipe outlet(s) must not extend more than 60 cm behind the centerline of the rear axle, and shall be no more than 60 cm above the ground. Many other constraints are given by the specific lay-out of each car, as well as by the need of packaging the whole engine system into an assembly as compact and light as possible, easy to install and access for inspection. Last, but certainly not the least, the cost must be as low as possible (the limit for the cost of the whole car, calculated according to the regulations, is $25,000 US, thus the use of expensive technologies, beside some exceptions, is not a practical proposition). As a result, on each car the degrees of freedom for the design of the intake and exhaust system are strongly limited.

The parameters selected for the optimization in the 2008 car of Modena and Reggio Emilia are described below.

As far as the intake system is concerned, the part considered for the optimization, visible in figure 5, lays between the intercooler outlet (left end) and the engine head inlet (right end). The intake runner and the pipe from the intercooler have the same constant diameter, named DI1, while their length may be different (LI1 and LI4). The two Helmholtz resonators are identical, and they are defined by a set of 4 parameters (DI2, DI3, LI2 and LI3). Finally, the volume of the junction between the resonators and the runner is called VI1.

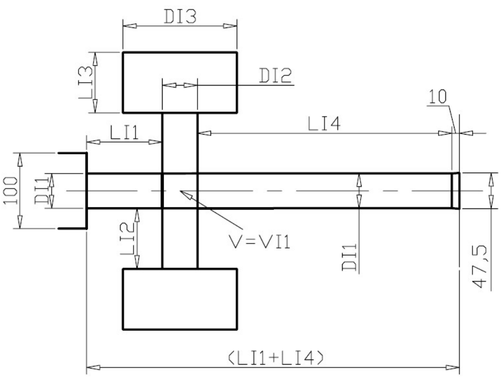

Figure 5: The optimized part of the intake system.

The exhaust system schematic is presented in figure 6. The twin ducts leaving the cylinder head have a constant diameter, DE1, and a length named LE1. After the junction (whose length is empirically assumed as the sum of the inlet and outlet diameters), the pipe has a diameter DE3 and a length LE3. The silencer is made up of two chambers (the first one stuffed by wool), a perforated duct (diameter is still DE3), and a short duct having a diameter of 33 mm. In the GT-Power model, both the perforated duct and the first chamber are divided into 8 parts, connected by multiple holes and openings through which the flow is free of losses. The length of each component along the silencer axis is called LSIL1, while LSIL2 is the length of the second chamber. Finally, the external diameter of the silencer is DSIL.

It is remarked that the modeling of the muffler is suitable for brake performance analyses, but it is not very accurate for acoustic predictions (according to the technical regulations, the measured Sound Pressure Level at about 6000 rpm must be less than 110 dB). For the last purpose, a higher refinement of the numerical model would be necessary, with unacceptable increase of the computational time. Therefore, empirical criteria have been adopted for muffler design, and a further simulation with a more sophisticated model is required to verify the actual noise level.

Table 1 presents the range of variation allowed for each parameter that must be optimized. Three further constraints are placed upon this set of parameters: the total length of the exhaust pipes (LE1+LE2) must be less than 1.5m; DSIL must be at least 20mm larger than DE3; the total length of the silencer must not exceed 600mm.

Variable	Unit	Min.	Max.
DE1	mm	25	42
DE3	mm	30	70
DI1	mm	38	58
DI2	mm	28	58
DI3	mm	60	200
DSIL	mm	50	200
LE1	mm	300	800
LE3	mm	300	800
LI1	mm	50	300
LI2	mm	30	130
LI3	mm	40	160
LI4	mm	100	500
LSIL1	mm	10	70
LSIL2	mm	40	400
VI1	mm^3	100,000	2,000,000

Table 1: List of optimized parameters and range of variation

THE GENETIC ALGORITHM

The influence of the 15 parameters described in the previous section has been explored by means of a single-objective genetic algorithm (GA). Since the optimization goal is to find the best combination of a number of independent parameters, it is necessary to define how good each configuration is on an analytical basis, i.e. a merit value has to be assigned.

The genetic algorithm optimization technique tries to reproduce the concepts of natural selection and evolution, and their principles are applied to find solutions for those problems which depend on a huge number of variables. In the field of Internal Combustion Engine CFD simulations, a number of applications are reported in literature, as an example in [10-13] and in [15]. According to J.Holland's definition of genetic algorithm [4], GAs are methods for moving from one population of chromosomes to a new population by using a kind of 'natural selection', together with other genetic operators (crossover, mutation, inversion). More in detail, a GA works with a population of individuals, each of which is a candidate solution for the optimization. Each individual owns a defined number of chromosomes, which are the genetic representation of the independent variables. Chromosomes consist of binary strings, whose length depends on the variety of values that chromosome can assume. For instance, an 8-bit chromosome leads to 2^8 = 256 possible different instances of it. Each bit in the chromosome represents a gene, and each gene is the instance of a particular 'allele' (which, in the binary representation, can only be

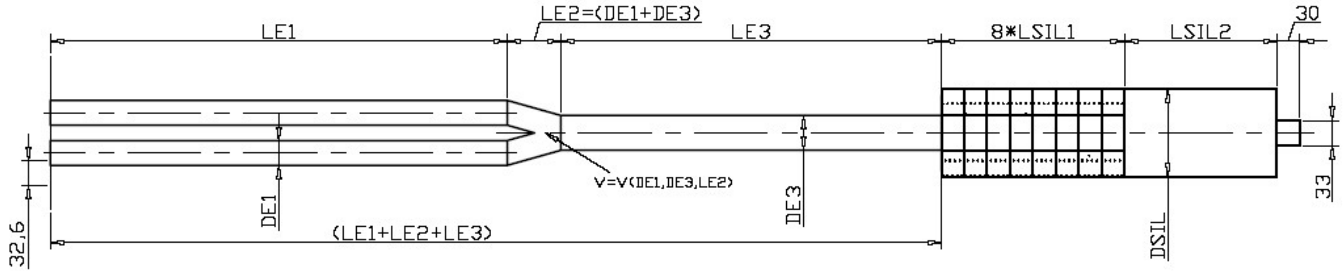

Figure 6: Schematic of the exhaust system.

0 or 1). It should be considered that the whole variable allowed range – e.g., [v_{min}, v_{max}] – has to be represented as an n-gene chromosome string, so determining a distribution of 2^n possible values the variable can assume within the range (matching to a chromosome possible configuration which can vary from 000….0, equivalent to v_{min}, to 111…1, equivalent to v_{max}); then, the variable value v matching a generic n-bit chromosome is computed as follows:

$$v(n-bit\ chr.) = v_{min} + bin2dec\{n-bit\ chr.\} \cdot \frac{v_{max} - v_{min}}{2^n - 1}, \quad (2)$$

where the *bin2dec* operator is a function that converts into a base-10 number the value represented by the base-2 n-bit binary chromosome (which range spans the integer {0, 1, 2, …, 2^n-1} set) [6].

The selection of the individuals for reproduction is fitness-proportionate: the probability that an individual is chosen for reproduction is proportional to its fitness value, i.e. to the value of the merit function as computed from the values of the independent variables extracted from the chromosome strings. In the case of a 1-D CFD simulation, the code itself calculates the fitness function. The values given by chromosomes are inputs, while the output fitness value is extracted from the predicted engine performance. For the current optimization, the following fitness function has been adopted:

$$f(\mathbf{x}) = fitness = \frac{1}{N}\sum_{i=1}^{N} P_i \ [kW], \quad (3)$$

i.e. the average output brake power, upon the simulated engine speeds. As a matter of fact, six different engine revs have been investigated: from 2500 to 7500 rpm, with a 1000 rpm step between each other. Reproduction of individuals is similar to natural reproduction: two parents own two different genotypes (i.e. different sets of chromosomes), and each chromosome of the son is randomly chosen from one of the two parents. Nevertheless, there is a defined probability that some other genetic operation occurs. The mutation of genes, for instance, randomly changes from 0 to 1 or vice versa the allele values of some locations in the chromosomes; conversely, crossover exchanges the subparts of two chromosomes – one from each parent – at a randomly chosen cut point.

A new generation differs from the previous one by a number of individuals who have been generated, and that replaces those individuals of the previous generation who had the worst fitness values. Usually, 50-200 generations are calculated. A schematic of the GA procedure adopted for the optimization may be summarized as follows [7]:

1. Start: first randomly-generated population, made of n individuals.
2. Calculation of fitness function values for each individual in the population.
3. Fitness-proportionate selection of $2 \cdot f_R \cdot n$ individuals for reproduction.
4. Reproduction and, thus, generation of $f_R \cdot n$ new individuals. Possible occurrence of mutation and crossover of chromosomes, with – respectively – p_M and p_C probabilities.
5. Substitution of the worse $f_R \cdot n$ individuals with the new generated ones. Start of a new generation.
6. Calculation of fitness function values for each of the new individuals.
7. Go to 3.

The choice of the most appropriate number of individuals and generations to simulate is important for the results of the genetic optimization, as well as a correct binary representation of the variable ranges.

The genetic algorithm has been coupled with GT-POWER by means of a Fortran program which integrates the genetic algorithm with 1-D simulations. Since a GT-Power run reads an input file containing model data, all the variables involved in the optimization have been parameterized within the model, so that a string for each of them appears in the input file (tagged as "dat"). Their values are changed by the Fortran program each time a run (i.e. an "individual", in GA lexicon) is started, as a function of values provided by GA. Then, the script is run to get brake power from the output file at each simulated operating condition, and to compute the fitness value of each individual. A schematic view of that procedure is reported in Figure 7.

As a term of comparison for the in-house developed GA, a similar optimization has also been run employing one well known commercial GA-based optimization software (modeFRONTIER 4.0 by ES.TEC.O). In this case, even being the objective of the optimization still focused upon the average brake power, a multi-objective genetic algorithm (MOGA) has been applied, and the single output power values at the simulated operating conditions have been chosen to be optimized. Since the same weight was given to each of the power values at the different engine speeds, the fitness value of the same individual resulted to be equal in both the in-house developed and the commercial optimizations. So, adopting a multi-objective GA has proved to be useful not in terms of performance of the optimization, but instead because it allows the user to use a series of analysis tools which investigate the influences of the input variables upon the single brake power values, more than on the average brake power only. Also in this case a procedure similar to the one reported in Figure 7 has been used to couple commercial code with GT-POWER.

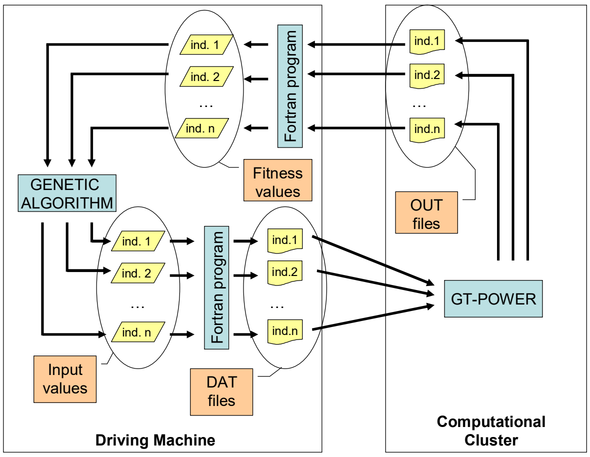

Figure 7: schematic showing the genetic algorithm and GT-Power coupling procedure

OPTIMIZATION RESULTS

A base configuration for the supercharged engine has been defined through an empirical optimization process, considering the parameters of table 1 and the following guidelines, suggested by experience. First, the length of the intake runner and the volume of the resonators have been set in order to maximize volumetric efficiency at high engine speed. Second, the length of the exhaust pipe has been calculated for a proper breathing from medium to high speed. Third, the transmission ratio between engine and compressor was set in order to get an average boost pressure of about 1.6 bar.

It should be mentioned that the modeling of friction losses and combustion and heat transfer has been set as in the naturally aspirated engine, so that any comparison between different configurations is independent on these issues. The authors acknowledge that the increase of intake pressure and temperature affects heat release rates, however the influence on brake performance should be limited, as far as knocking is kept under control. For this purpose, in comparison to the naturally aspirated engine, the cooling system of the supercharged unit has modified, while injection parameters and spark advance have been properly calibrated at the dynamometer bench.

Brake power of the base supercharged configuration is reported in figure 8, compared to the performance of the naturally aspirated engine. The fitness of the former, computed according to (2), adds up to 34.483 kW.

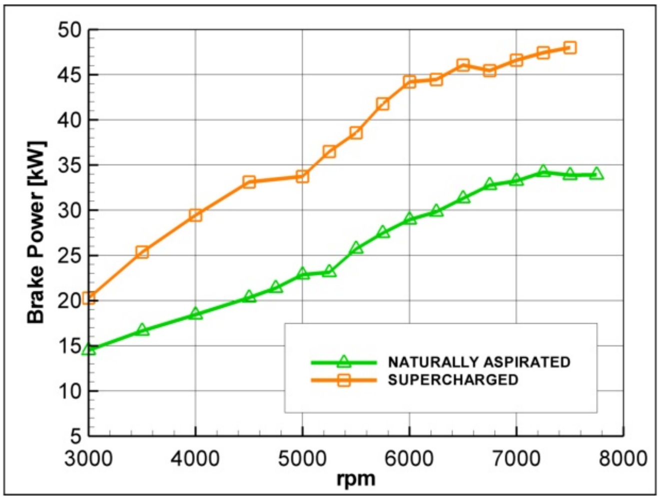

Figure 8 – Comparison between naturally aspirated and base supercharged brake power output, at full load (CFD-1D simulations)

Then, the optimization of the engine model has been performed by using the in-house developed single-objective GA, described in the previous section. A total of 65 generations has been simulated, each one consisting of 150 individuals. Probabilities of crossover and mutation have been set both to $p_M = p_C = 0.4$, while the fraction of the population to be reproduced between one generation and the next has been chosen to be f_R =0.35, resulting in 52 new individuals in each new generation. The definition of these parameters has been decided taking into account the guidelines provided in [4,8,9], as well as some other optimizations carried out previously. The variables' allowed ranges have been subdivided into 256 intervals, i.e. 8-bit chromosome strings have been used to genetically represent the variables values. As a result, a total of 9750 individuals have been simulated, and the best individual has been found by the GA after 44 generations, yielding a fitness value of 35.475kW, as shown in figure 9.

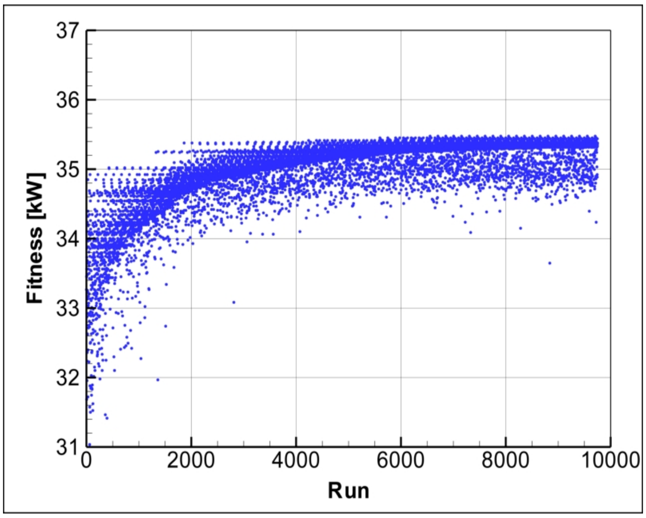

Figure 9: fitness values vs design ID for the in-house GA optimization

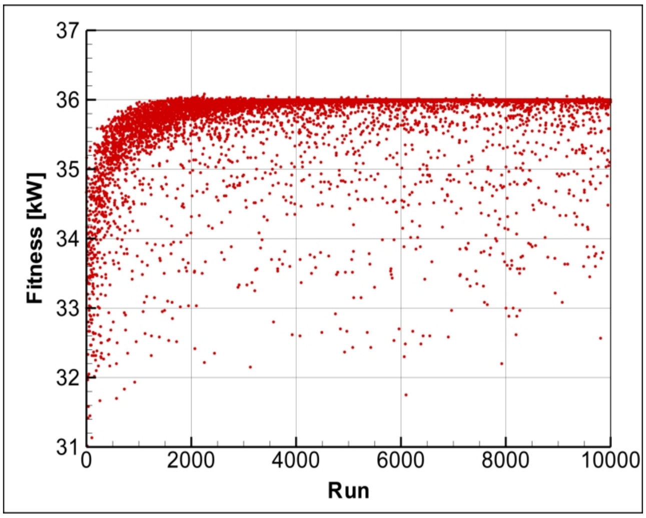

Figure 10: fitness values vs design ID for the commercial code GA optimization

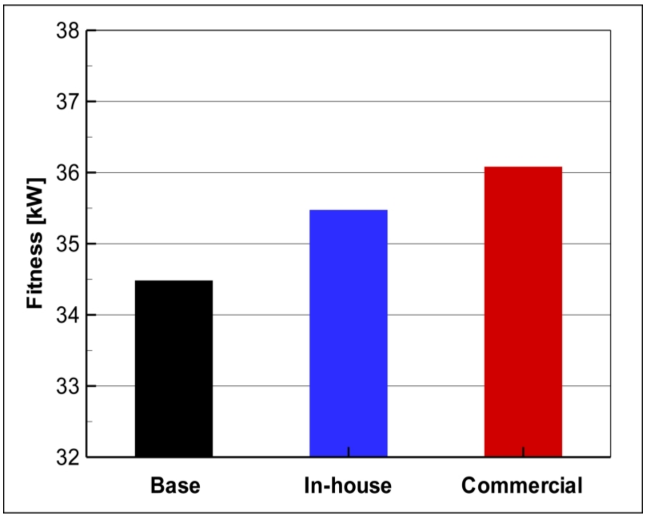

Figure 11: Fitness values of the best configurations

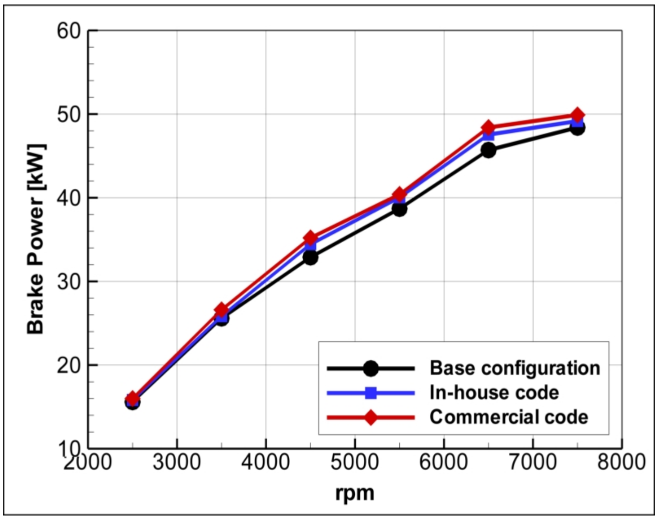

Figure 12: Brake Power output of the optimum configurations.

At the same time, the optimization using modeFRONTIER 4.0 has been run considering the same dimension of the population (150 individuals), while different genetic parameters have been chosen. As well known [4], genetic parameters such as mutation, cross-over, and reproduction probabilities, interact among them in a non-linear way, and it is not still clear how such parameters should be properly combined. As a matter of fact, they cannot be optimized one by one, then the values that worked well in previous reported cases are generally adopted. Thus, the parameters set up has been performed using a group of values suggested by developers in the users' guide [18] and refined through a trial and error process. In details, the probabilities of crossover and mutation have been eventually set to 0.5 and 0.1, respectively, while the fraction of the population to be reproduced between generations is 0.95. In the analyzed case, simulations stopped after about 10000 individuals, at the reach of convergence (figure 10). It is mentioned that the best individual, owning a fitness of 36.083 kW, has been found since generation 15.

Figure 11 shows a graphical comparison among the optimized configurations in terms of fitness, while, in figure 12, the brake power curves are plotted.

In order to further assess the performance of the GA developed by the authors, figure 12 shows a comparison between the results of the in-house code with the ones yielded by the commercial optimizer. The comparison is made in terms of percent variation of the geometric parameters listed in Table 1. It may be observed that for almost all the parameters the trend suggested by the in-house code is confirmed by the commercial optimizer, the only significant exceptions being the dimensions of the Helmholtz resonators (LI2, LI3, DI3). However, it is remarkable that the optimum volume of the resonators is almost coincident, while the ratio of neck cross section to length ($DI2^2/LI2$) is similar. Since the resonance frequency is given by [5]:

$$f = \frac{c}{2\pi}\sqrt{\frac{S}{VL}} \tag{4}$$

(where c is speed of sound, S is neck cross section, L its length and V is the resonator volume), the two configurations are very close from an acoustic point of view.

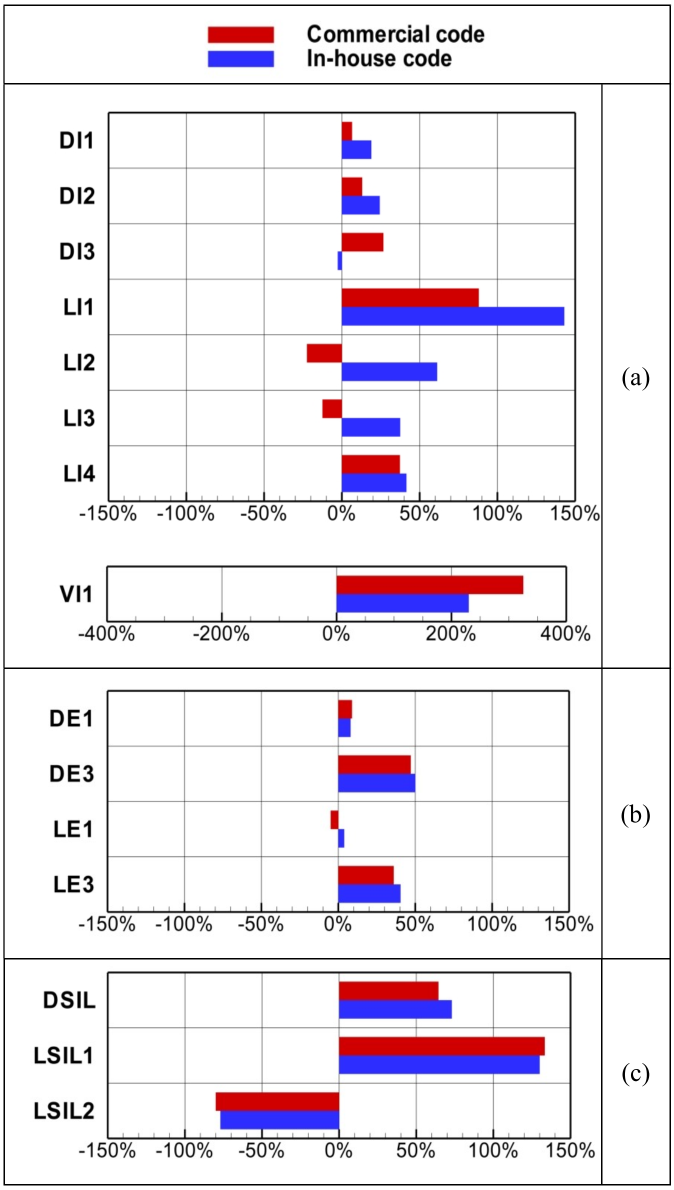

Figure 13: Differences between the values of the parameters in the genetic optimisations, and the base configuration, for intake (a), exhaust (b) and silencer (c) geometries (percent).

Figure 14 presents a comparison among the three optimized configurations in terms of engine performance. It may be noticed that the geometry of intake and exhaust system affects boost pressure: both the GA optimizers allow lower pumping losses than on the BASE layout, as demonstrated by the PMEP graph, so that the compressor delivery pressure is relieved. The automatic optimizations yield slightly higher airflow rates at low-medium engine speeds, while the improvement of IMEP at high speed is mainly due to the lower pumping losses and the reduction of power absorbed by the compressor (same airflow rate, lower delivery pressure). Since FMEPs are almost constant for all the configurations (the difference in terms of in-cylinder maximum pressure are small, being the values of boost pressure very close), the enhancement of indicated work also improves mechanical efficiency a little bit. As a result of better pumping and organic efficiency, fuel consumption is slightly lower in the configurations optimized by Genetic Algorithms. Finally, it is observed that the improvement of engine performance yielded by optimization is quite significant, considering the good level of the base configuration and the high number of constraints. However, sophisticated multi-objective genetic algorithms, as the ones employed in the commercial software, not always produce relevant benefits, in comparison to simple single objective algorithms, as the one developed by the authors.

CONCLUSION

The paper reviews the development of a Formula SAE/Student engine, supported by CFD-1D simulations and experiments at the dynamometer bench. Measurements have been performed on a naturally aspirated unit, in order to calibrate a GT-Power base model. Furthermore, a single objective Genetic Algorithm has been developed by the authors and coupled to CFD simulation.

The calibrated engine model has been modified in order to include a mechanical supercharger and an intercooler, and the intake/exhaust system has been optimized in three different ways: manually, i.e. through a series of simulations controlled by the authors; coupling GT-Power to the in-house GA optimizer; coupling CFD simulation to a well known commercial optimizer (modeFRONTIER 4.0), adopting a Multi-Objective GA. All the automatic optimizations have been run considering 14 variables.

The commercial optimizer achieves the best result in terms of fitness value, yielding an improvement of 4.6% if compared to the manually optimized configuration. However, also the in-house GA provides a configuration with 2.9% more average power than the base configuration. Furthermore, it has been observed that for almost all the parameters of the intake/exhaust system the variation trend suggested by the in-house code is fully confirmed by the commercial optimizer.

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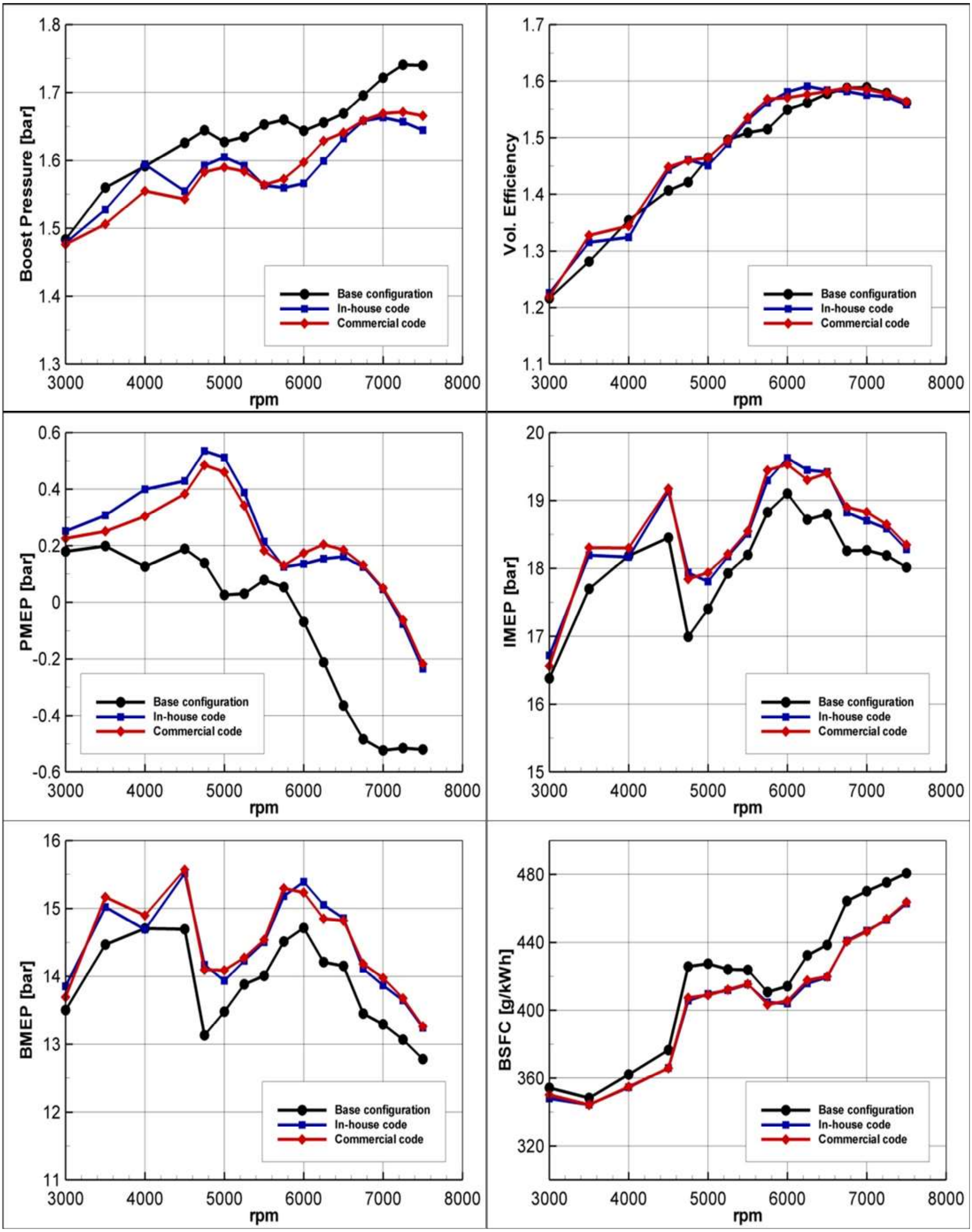

Figure 14: Comparison among the three optimized configurations in terms of engine performance

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

BMEP = Brake Mean Effective Pressure [bar]

c = speed of sound [m/s]

f = resonance frequency [Hz]

FMEP = Friction Mean Effective Pressure [bar]

f_R = reproduced fraction of the population

GA = genetic algorithm

IMEP = Indicated Mean Effective Pressure [bar]

L = resonator’s neck axial length [m]

N_i = i-th resonant engine speed [rpm]

p_C = probability of crossover

p_M = probability of mutation

S = resonator’s neck circular surface [m^2]

V = resonator’s volume [m^3]

V_d = engine displacement [m^3]