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OPTIMIZING BRAKE SPECIFIC FUEL CONSUMPTION OF A GASOLINE ENGINE BY VARYING THE VALVE EVENTS

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ABSTRACT

The Brake Specific Fuel Consumption (BSFC) represents the efficiency of fuel and is related to the overall efficiency of an ICE. This study evaluates strategies for varying the events in the intake and exhaust valves to improve the BSFC of a gasoline engine. An optimization study was carried out to obtain the lift curves that allow the minimum fuel consumption for operating conditions at partial loads, which are representative of driving a vehicle in an urban cycle. A one-dimensional model of the Etorq Evo 1.6L turbo engine was taken as a basis and converted to an aspirated engine model. The Brent optimization method, which is available in the standard optimizer of the software and intended for optimizations with only one independent variable, was used in conjunction with the Univariate search method, thereby allowing more than one variable to be incorporated into the optimization process. Thereafter, an analysis was made of the phenomena that led to reducing the BSFC and also the implications for torque, volumetric efficiency and pollutant emissions. The results obtained a maximum reduction of 22.5% for the BSFC for a engine operating with 2 bar of Brake Mean Effective Pressure and at a speed of 1500 RPM.

KEYWORDS: Gasoline engine; Variable valve events; Brake Specific Fuel Consumption; Optimization methods; Efficiency Enhancement.

1. INTRODUCTION

Recently, different solutions are being used to improve fuel consumption and to reduce the emissions of pollutants. Some well-known examples are the variable valve actuation technologies, the recirculation of exhaust gas, direct injection, and hybridization of vehicles.[1]

Variable valve actuation technologycan be used to achieve changes in important parameters of engines, such as specific fuel consumption, emissions, torque, and output power, in accordance with the working condition of the engine. This technology changes the opening and closing times of valves and the length of their lift and stroke.[2]

In today's automotive market, car manufacturers make use of this technology in different configurations and operations. For example, the BMW Vanos and Toyota VVT-i engines feature the VarioCam and VVT-i systems, respectively, which serve only to speed up or delay valve times. On the other hand, engines like Honda i-VTEC, Fiat MultiAir and Toyota VVTL-i present systems that are able to gradually adjust the elevations and timing of the valves.[1]

Despite this being a recognized system in the automobile industry, studies that focus on the variable forms of action of the valves are constantly seeking a better understanding of the effects of parameters on the engine. Arising from this, Parvate-Partil et al.[3] reviewed the literature on the variation in the events of the intake and exhaust valves and the implications of these events for the engine pressure-volume cycle, including volumetric efficiency, NOx and hydrocarbon emissions and fuel consumption. They simulated some of the combined delay and advance settings for opening and closing the intake and exhaust valves using GT-Power software. Their research succeeded in obtaining results equivalent to those achieved in experiments applied on real engines.

Zhao and Xu,[4] applied the Atkinson cycle to control engine load in order to reduce fuel consumption. A 1-D model of the Atkinson cycle engine was developed in GT-Power software. Then, the engine's fuel consumption was optimized using the Genetic Algorithm method. The errors obtained from the model in relation to the real engine were considered small and acceptable by the authors. Thus, by comparing the original Otto cycle engine with the optimized Atkinson cycle engine using break specific fuel consumption (BSFC) maps, improvements in fuel consumption were observed.

Deng et al,[5] studied the operation of a spark-ignition engine using a mixture of gasoline and 35% butanol. The authors noted that advancing the ignition timing led to improvements in output power, fuel consumption and emissions of hydrocarbons and carbon monoxide; on the other hand, NOx emissions increased considerably. Given this, a model of the engine was developed and calibrated in GT-Power software and the strategy of recirculating exhaust gases was applied by varying the valve overlap. The results showed a decrease in NOx emissions, without causing major losses to the torque and fuel consumption.

Tie Li et al.[6] applied the Miller cycle to a booster-modified direct injection spark ignition engine by delaying and advancing the closing time of the intake valve in order to suppress the engine's crash effect and to analyse its effects on fuel consumption, torque and output power compared to the original engine, which had a lower compression ratio at low and high loads. They achieved an improvement in the low load consumption by 7%.

In order to reduce the consumption of a spark ignition engine without deteriorating its performance, He et al,[7] used some strategies to vary the timings of the valves.

The authors developed a model of the Miller cycle engine which focused on the configuration that provided the best response among a set of parameters (knock, burn duration, brake specific fuel consumption, brake torque, etc.). The three configurations analysed by the authors used the late-intake-valve-closing cam strategy and the early-intake-valve-closing cam strategy and two different types of multi-stage boosting systems (a two-stage turbocharger and the combination of a turbocharger and a supercharger). For partial load operations the LIVC strategy presented better performance than the EIVC strategy in relation to the stability of combustion and fuel economy.

At full load, the results showed a decrease in the consumption of fuel while maintaining levels of torque similar to those of the original engine due to using the two-stage turbocharger.

Yangtao Li et al.[8] optimized a hydraulic valve actuation system (HVVA) to improve performance regarding power (total load) and fuel economy (under partial load). They proposed a new HVVA engine model which was calibrated by using experimental data and the optimization was performed using the Genetic Algorithm method. At full load, implementing HVVA resulted in a 10.4% improvement in output power. At partial loads, strategies for delaying closing the exhaust valve and advancing opening the intake valve were applied to intensify the recirculation of internal gas and thus to meet demands without using the throttle valve. The BSFC was reduced by 13.1% with a load of 7 Nm.

In order to reduce the losses from the intake system of a spark-ignition engine, Sun et al.[9] introduced a different speed control system to replace the throttle valve. The system consists of two intake valves connected in series to the intake port. The performance of the engine operating with the system developed by the authors (series valve speed control - SVSC) was compared to the engine operating with a throttle valve. The consumption of the engine with the SVSC system showed an economy of up to 12%.

Another study related to this theme was developed by Teodosio et al.[10] They studied the influence of strategies applied to the intake valve (EIVC, LIVC and Full Lift) of a turbocharged downsized spark ignition engine using 1D and 3D numerical approaches. The results of low load operations showed improvements of up to 5.6% of the BSFC when applying the EIVC strategy relating to the Full Lift strategy. Even in the low-load condition, the LIVC strategy provides lower benefits than the EIVC strategy. Applying the LIVC and EIVC strategies at high loads allowed reductions in consumption regarding the Full Lift profile strategy, besides reducing problems related to knock.

In the same context, Ghodke and Bari,[11] developed an optimization study of a single cylinder spark ignition engine that was naturally aspirated. They varied individually and simultaneously the intake runner diameter and intake valve timing with the aim of improving the volumetric efficiency. An increase of about 8.5% in the volumetric efficiency was obtained by varying the diameter of the intake runner. On the other hand, varying the valve timing led to a 3% improvement in the volumetric efficiency. Finally, the combination of both strategies afforded an average improvement of 12%.

In order to obtain some improvements in the BFSC while constantly maintaining the original performance of the engine, Sawant and Bari[12] analyzed a 500cc KTM internal combustion engine. The model was validated and operated at a wide-open throttle (WOT) mode. By varying the opening and closing times of the intake valve, and the profiles of lift and elevation, improvements in performance and a reduction in fuel consumption were achieved.

Li et al,[13] used GT-Power software to develop a 1-D model of an Otto cycle gasoline engine. The authors modelled and coupled a hydraulically actuated valvetrain to eliminate the throttle valve in the load control. Finally, they optimized the final model using the Genetic Algorithm method in order to obtain high performance at full load and high fuel efficiency in partial loads. To achieve these performance objectives, the authors applied the Atkinson cycle in partial load and Otto cycle operations at full load by changing the compression and expansion processes of the engine. Comparing the performance results between the optimized model and the original one, an increase of power in was verified; in partial load operations, the optimized engine obtained greater fuel economy compared to the original engine; this was a consequence of the lower pumping losses and higher thermal efficiency.

Despite the possible benefits from varying the valve timings of the engine, there are some problems that need to be studied and researchers are looking for ways to solve them. For example, by eliminating the throttle valve and establishing load control by using a Fully Variable Hydraulic System (FHVVS) to control load in a spark-ignition engine, Zhang et al.[14] observed slower and more unstable combustion, thus reducing thermal efficiency, despite the reduction of pumping losses. One of the existing problems is the low intensity of mixing and the poor performance of the combustion after applying the EIVC strategy at low loads (increase of turbulence dissipation from Intake Valve Closing to Bottom Dead Centre).

Silva et al.[15] conducted an optimization analysis of the geometry of an intake manifold of 4-cylinder spark ignition engine. The authors performed the optimization using a numerical approach based on the 1D GT-Power simulation platform. Their goal was to provide the engine's greatest volumetric efficiency and its brake-specific fuel consumption at each engine rotation speed in the range of 1000 - 6000 RPM. The authors found that the engine can achieve higher values of volumetric efficiency and lower brake-specific fuel consumption, depending on the speed conditions.

Engine performance is highly dependent on the engine valve motion and varying valve events has the potential to achieve improvements in several characteristics of an engine in order to make it more economical and efficient, bringing direct influences to the engine emissions. In this work we will evaluate the influence of valve variation on the operation of a 4-cylinder spark ignition engine. The valve events will be optimized in order to evaluate its effects in brake specific fuel consumption, considering the comparison of an aspirated engine model and a turbo engine model.

2. METHODOLOGY

A 4-cylinder spark ignition gasoline engine model was used to study the effects of valve events. The engine model was developed using GT-Power software and consists of an intake system made up of objects that simulate environmental conditions, objects of tubular sections (in which diameter, curvature, length, material characteristics, among other configurations are specified), orifices, an air filter, and butterfly valve, all of which are used so as to carry the air to the intake manifold. This manifold holds the plenum which receives the intake air, which is then distributed to the cylinders using ducts called runners. In the runners, the injected fuel is defined as a function of the inducted air flow and is measured by a sensor located upstream of the air filter. The air/fuel ratio is 15.5:1 and must be specified in the injector object inside the GT-Power model. The engine model (Figure 1) has four cylinders each with two inlet valves and two exhaust valves at the top. The exhaust system comprises four pipes connected in pairs to single pipes that couple to the exhaust section.

The engine used to develop the GT-Power model has originally a turbocharger assembly. In turbocharged engines the intake system pressures are higher than those of the exhaust, which therefore does not allow the recirculation of gases to be evaluated during the period in which the valves overlap. The elements representing the turbocharger assembly have been removed from the model to allow the analysis of variations on valve lift curves. This change makes the engine naturally aspirated. Figure 1 shows a simplified representation of the model.

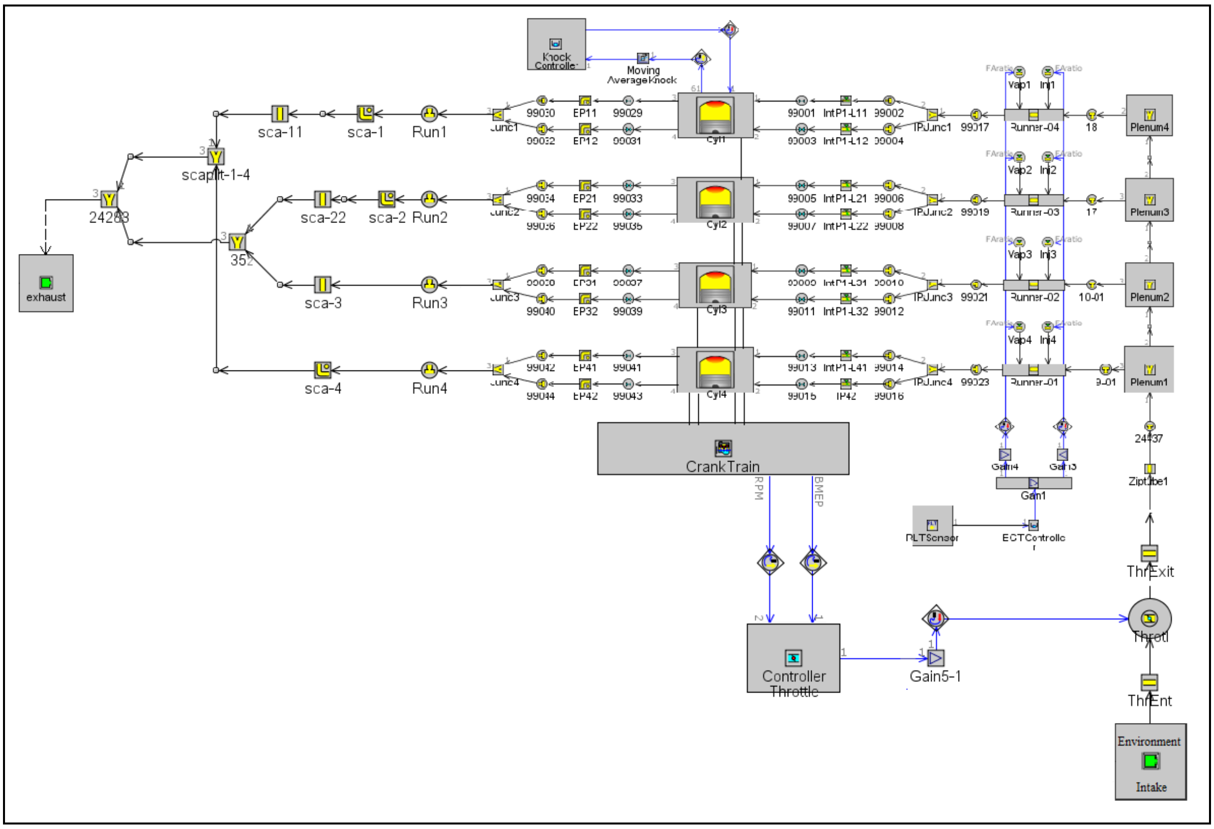


Figure 1: Representation of the engine model.

To evaluate the engine model at full load and partial loads, it is necessary to use the largest number of operating points, obtaining the BSFC map for all possible scenarios. However, the computational time to accomplish this task would be extremely high. In this work the choice of operating points was based on the concept of canonical points of the specific fuel consumption map, which represent the points most commonly used in an urban consumption cycle.[16] Table 1 shows the list of cases for which valve event optimizations were performed.

Table 1: Engine operating points.

Case	1	2	3	4	5	6	7	8
Speed (RPM)	1500	2000	2000	2000	2500	2500	3000	3500
Pressure (bar)	2	2	4	8	8	10	6	10

To determine the best valve event for each condition, the Optimizer Direct tool was used. This allows one or more variables in the model to be changed in order to analyze the behavior of another parameter of the model.

It begins by conducting successive simulations up to the iteration that provides the best parameter value, which can be a target value. The Brent optimization method was used for the simulations. This method optimizes one independent variable at a time.[17,18,19]

In this work, the objective function chosen to be optimized by varying valve events was the Brake Specific Fuel Consumption (BSFC). This parameter is directly related to the engine's efficiency.

Six independent variables related to the lift curves were used in the optimization process: the valve positions on the cycle, their amplitudes, and the maximum elevations of the intake and exhaust valves. The number of iterations in the optimization process increases greatly as more variables are added to the analysis. To analyse several variables in the optimization step, it was used the univariate search technique. This method is a technique which consists of optimizing the objective function in relation to each design variable, one by one. A starting point is chosen and one variable is selected to optimize the function, while the others are kept constant. The process continues by alternating the variables and updating the values achieved in the optimizations, until the optimum value is obtained, i.e., when the variation of the optimized characteristic between consecutive iterations is smaller than the convergence criteria. An alternative to the Univariate method would be to apply a sensitivity analysis to prioritize the dominant variables, i.e., those that cause the greatest changes in the objective function. However, since there was not a mathematical expression for the objective function concerning the design variables, all variables were maintained in the analysis.[20]

Within the GT-Power software, the positions of the lift curves were associated with the variables VVT and $VVT2$ which define the angular distances of the lift matrices of the intake and exhaust valves, respectively, relative to the reference point, established as the top dead centre; the amplitudes of the elevation curves for the intake and exhaust valves were controlled by multiplicative factors defined by the variables $range$ and $range2$, respectively; and the maximum intake and exhaust valves elevations were controlled by multiplicative factors defined by the parameters $lift$ and $lift2$, respectively. All created variables appear in the case setup option, in which their initial values were defined and updated after every optimization iteration (Univariate method).

The specific fuel consumption optimization is performed with one variable at a time and the procedure is repeated until it achieves the tolerance value of less than 5% between optimization cycles or after reaching a total number of 36 simulations (this means six optimizations for each variable), in order to limit the simulation time. The initial values of the independent variables are updated during the simulations in the Case Setup.

To start the optimization process, it is necessary to specify other additional parameters. The first parameter to be specified is the resolution which determines the convergence of optimization and the sizes of the steps in the domain exploration. It was set at 2%. The other parameter to be defined relates to the variation range of the independent variable. Large intervals were defined for the first iterations. The variation range is reduced as the variations of the independent variables became small throughout the simulations. In some cases, the range had to be adjusted to avoid absurdly low values of specific fuel consumption.

After the optimization step, an analysis of the specific consumption was performed relating to the aspirated engine operated with the original camshaft using the GT-POST, a graphic-plotting and the data-processing tool that is used by the GT-SUITE. Other characteristics such as volumetric efficiency, pumping losses, exhaust gas composition and exhaust gas recirculation (EGR) were included in the analysis. According to the literature, these parameters are also influenced by the change in valve events.

3. RESULTS AND DISCUSSION

This section analyzes the profiles of the lift of the intake and exhaust valves in order to identify how the changes in the original profile resulted in a lower specific fuel consumption. In addition, the behavior of other parameters such as torque and engine emissions under urban operating conditions will be examined, focusing on two extreme cases with respect to engine speed and effective mean pressure.

The optimization results will be presented for the cases with lower and higher effective mean pressure and engine speed among the set of cases studied: they are the operation points in the conditions of 1500 RPM and 2 bar and the condition of 3500 RPM and 10 bar. The optimized lifting profiles for these two operating conditions provided the maximum percentage reduction for operation at 1500 RPM and 2 bar and minimum decrease at 3500 RPM and 10 bar.

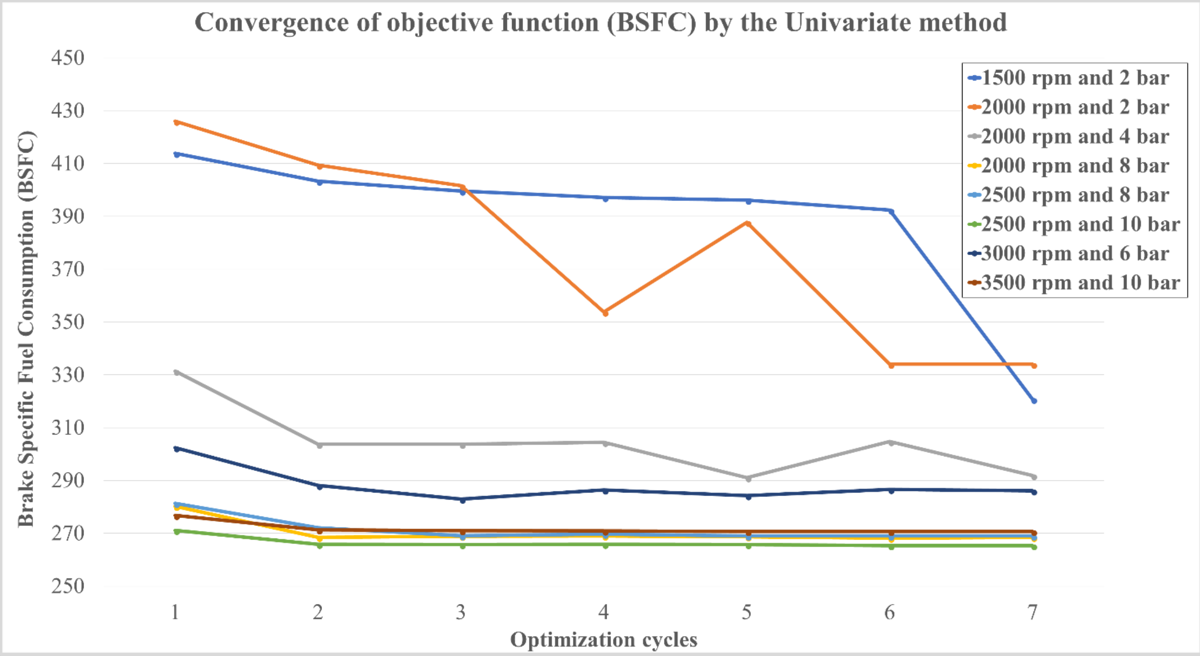
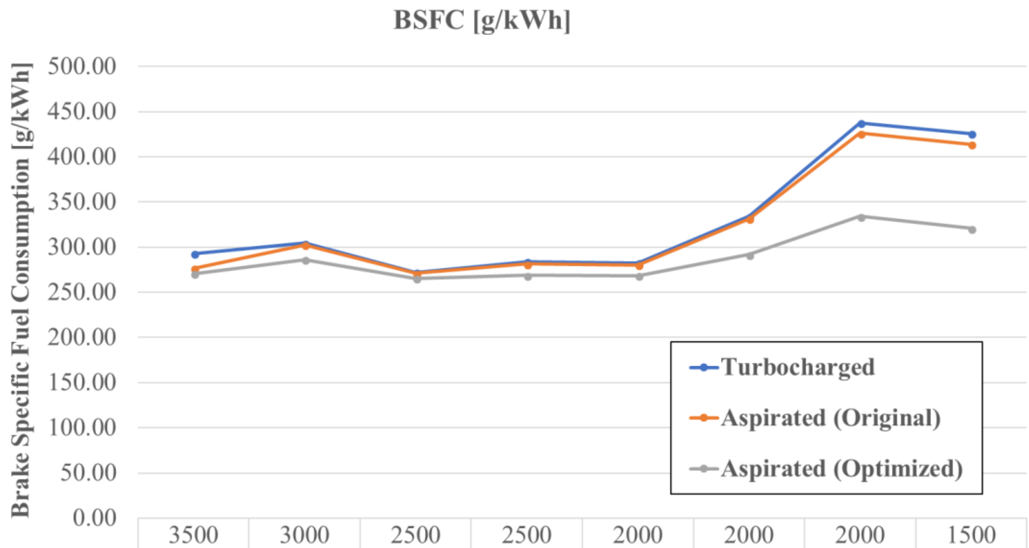


Figure 2: Specific fuel consumption obtained during the optimization step for the selected operating conditions by the Univariate method.

Figure 2 shows the behaviour of specific fuel consumption during the optimization cycles (each cycle being a set of six optimizations for the six independent variables applied to the Univariate method). Note that, in the case with the engine running at 1500 RPM and 2 bar of brake effective mean pressure, the value of the objective function still shows a very large variation between consecutive cycles, but due to the criterion of convergence that was established, the optimizations were stopped. In the case of 3000 rpm of engine rotation and 6 bar of brake effective mean pressure, the lowest value of the objective function was obtained in the third optimization cycle and then higher values were reached, thus demonstrating that the method may not be as efficient for all situations. Note also that for high engine speeds and brake effective mean pressures, the reduction in consumption was smaller and the objective function showed fast convergence. It is worth noting that, despite the long simulation periods, the number of optimizations established as a stopping criterion may not be enough to obtain results as close to the global optimum value as wanted. Nevertheless, significant reductions in the objective function were achieved, which represented local optimum values that obtained a variation of less than 5%, between consecutive optimization cycles except for the operation point at 1500 RPM and 2 bar.



	3500 10	3000 6	2500 10	2500 8	2000 8	2000 4	2000 2	1500 2
Turbocharged	293.10	304.32	272.22	283.87	282.69	334.18	437.78	425.79
Aspirated (Original)	276.74	302.41	271.10	281.32	280.24	331.44	426.04	413.88
Aspirated (Optimized)	270.70	286.09	265.29	268.97	268.49	291.89	334.06	320.79

Speed [RPM] | BMEP [bar]

Figure 3: Specific fuel consumption for the turbocharged model, aspirated engine model with the original cam profile and the aspirated engine model with the optimized cam profile.

Figure 3 shows how the BSFC curves were plotted for partial load and speed. These curves relate the turbocharged engine model with the profile of the original cam, the aspirated engine model with the original lifting profile and the aspirated engine model with the optimized lifting profile. The highest percentage reductions were obtained for low speed and brake effective mean pressure (BMEP), reaching maximum values of 22.5% reduction with the motor operating at 1500 RPM and with an effective mean pressure of 2 bar. At pressures above 6 bar and rotations above 2000 RPM the percentage reductions did not exceed 6%. This means that the cam profile of the original camshaft was designed based on an average operating condition close to these operating points.

To further investigate the reasons for the lower BSFC values when modifying the lift profiles of the intake and exhaust valves, the original and optimized valve curves for the aspirated engine were plotted as shown in Figure 4.

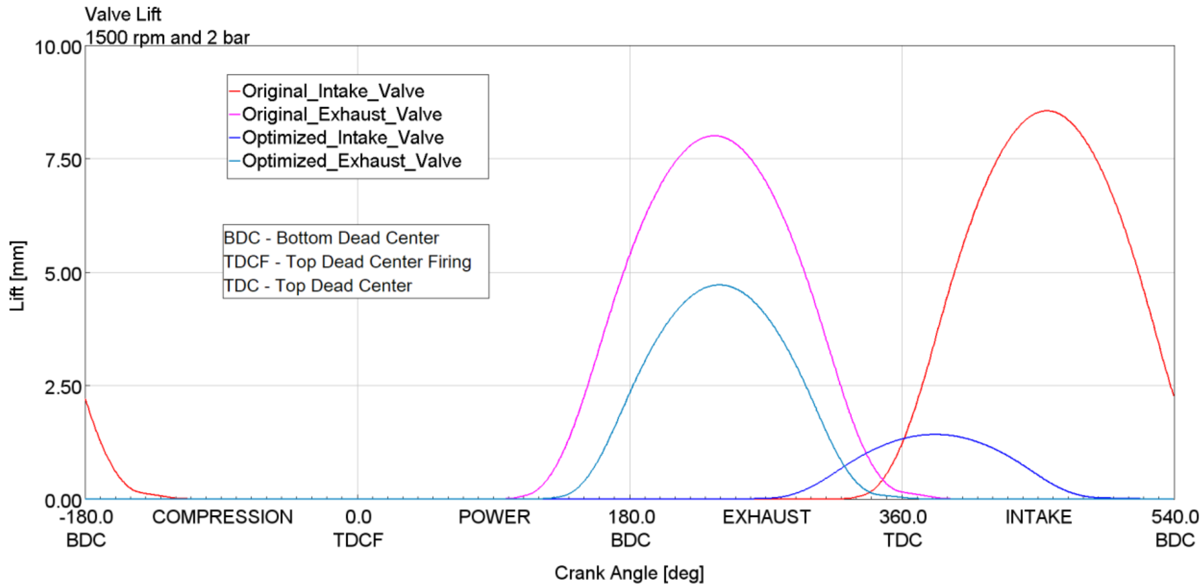


Figure 4: Valve lift curves before and after optimization for operation at 1500 rpm and 2 bar.

The operating condition that obtained the greatest drop in consumption was at 1500 RPM and 2 bar is shown in Figure 4. The opening and closing of the intake valve was anticipated at 58o and 13o respectively. There is also a great reduction in its elevation in relation to the original curve.

The early opening of the inlet valve may lead to an increased overlapping of the exhaust and intake valves. For an aspirated engine operating at partial loads, the consequence of this anticipation is the reflux of exhaust gases into the cylinder which in turn reduces the amount of mixture admitted at the end of the intake stroke, since part of the volume is occupied by a mixture of burnt gases.[21,3] Figure 4 shows that the period of valve overlap has increased.

From the graph of the mass flow through the inlet and exhaust valves (Figure 5), it can be noted that, despite the greater angular cross-section of valves, both the duration and the maximum value of the reverse burned gas flow rate of the exhaust system to the cylinder decreased, but the period of reflux of cylinder gases to the intake system increased, resulting in an increase of the percentage mass of burned gases trapped in the cylinder before combustion (16% to 26.9%). Also, the pressure rises from 0.34 bar to 0.88 bar at the intake port (reduction of air-fuel mixture intake work), the maximum temperature obtained at combustion reduces from 2456 K to 2135 K, and the mass of air imprisoned drops from 110 mg to 96 mg.

Anticipates the closing of the inlet valve is another strategy to control the admitted load. This interrupts the flow of air to the cylinder, and just a smaller amount of the mixture is allowed. This change also contributes to admitting less air, and to reduce specific fuel consumption and pumping losses.[3] Moreover, during the remaining displacement to the BDC, the entrapped mixture undergoes a slight expansion which causes a reduction in the pressure and, consequently, reducing the temperature of the fresh mixture. This also contributes to reducing the maximum temperature inside the cylinder.

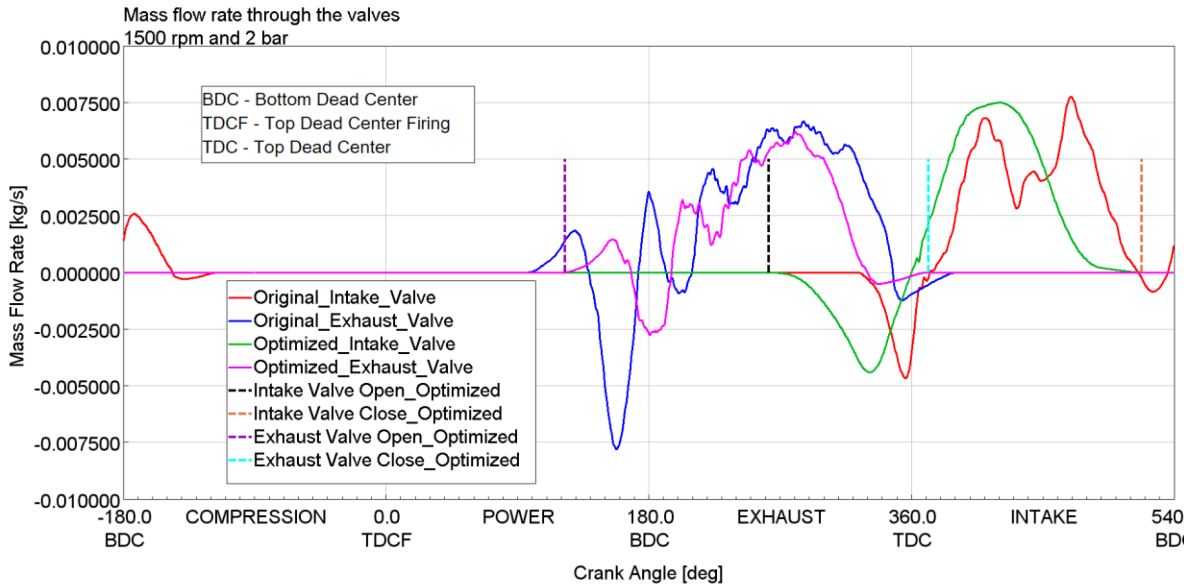


Figure 5: Mass flow through the intake and exhaust valves before and after optimization at 1500 rpm and 2 bar.

Depending on the geometry of the valve, the air flow area varies from a circular cone trunk to the area of the circular section between the valve stem and the valve port, from the slightly open position to the maximum elevation, respectively. The lower the elevation, the smaller the amount of mixing inducted in the cylinder due to the smaller flow area.[22,23] In order to verify if there was a reduction of the average flow area through the intake valve due to reducing the maximum elevation, the average flow velocity and the mean volumetric flow rate were used to determine the average area. The result showed a reduction of the mean area from 0.7 cm2 to 0.32 cm2. As the intake valve lift is very small for the operation at 2 bar and 1500 RPM, the possible implications would be a reduction in volumetric efficiency, an increase in torque, a reduction of BSFC, an increase in thermal efficiency, small reductions in hydrocarbon and CO emissions and a small increase in the emission of NOx.[24]

The elevation curve of the exhaust valve suffer a delay of its opening and an anticipation of its closing related to the original configuration. The opening of the exhaust valve at the exhaust stroke causes the exhaust gases to be expelled because of the difference between the interior pressure of the cylinder and the pressure of the exhaust system, a phenomenon known as blowdown. This favors the reduction of the pumping losses to expel the gases and reduces the temperature of the exhaust gases. Also, this phenomenon causes the disadvantage of power being wasted because the gases expand to the Bottom Dead Center.[1,21,25] Even with the valve opening closest to the Bottom Dead Center, the power output and engine torque were lower due to a smaller amount of fuel and a weaker blend being burned.

Closing the exhaust valve before or near the top dead center reduces the overlap period of the valves and the residual gas may get trapped in the cylinder due to the shorter exhaust stroke. For the case studied, the difference in closure angle between the optimized and original cases was small (about 20 degrees of crank angle), which may have contributed slightly to trapping residual gases, but did not reduce the valve overlap period relative to the original lift curve, since the anticipation of the intake valve opening was a 58o crank angle. It is worth remembering that the retention of residues in the cylinder can cause a reduction of the indicated thermal efficiency, increase fuel consumption, and decrease the formation of NOx (reduction of the temperature of the mixture before the combustion) by making the admitted fresh mixture weak when it is diluted with burned gases.

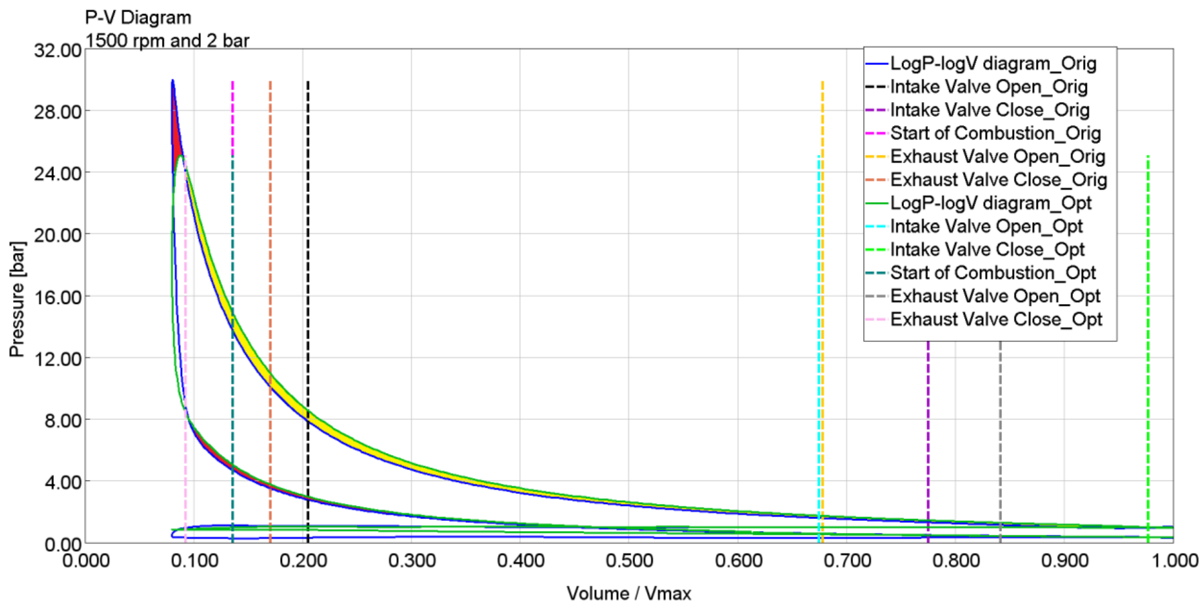


Figure 6: P-V diagram of the engine before and after optimization at 1500 rpm and 2 bar.

The p-V diagram was plotted (Figure 6) to show the optimization result in terms of volumetric efficiency and power output at 1500 RPM and 2 bar. The area painted in yellow represents the power gain indicated relating to the original case. The red area indicates a loss of power due to a lower pressure peak in the optimized case because of the smaller amount of fresh mixture inducted in the cylinder. The area painted in red at the lower loop means that there was a reduction in the work of pumping the gases into and out of the cylinder. This last change is justified by the increase of the inlet pressures (closer to the atmospheric pressure). Figure 7 shows the p-V diagram plotted on a log-log basis, facilitating identifying the occurred changes.

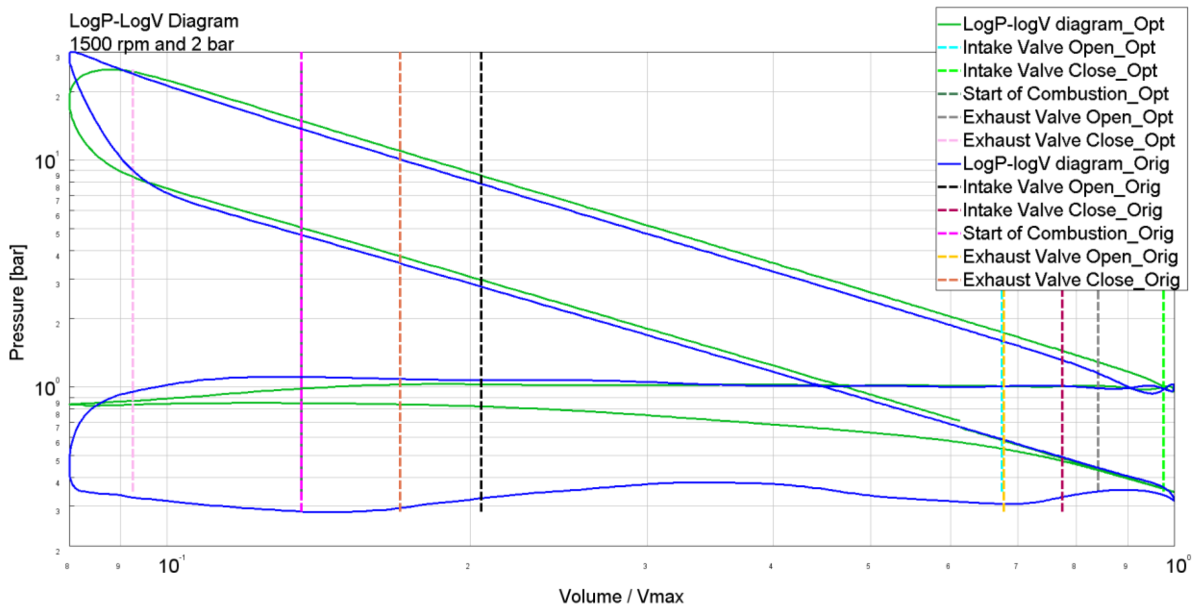


Figure 7: log(p)-log(V) diagram of the engine before and after optimization at 1500 rpm and 2 bar.

It is important to notice that, since the optimization problem involved six independent variables, it is difficult to quantify the weight of each one in the result obtained.

Table 2 shows some important parameters that changed after optimization. The results show that there was a significant increase in NOx emissions, despite the reduction of the maximum temperature inside the cylinder, which could have been due to the delay of the exhaust valve opening (it took longer for the gases to be oxidized), higher pressures along the course of expansion, poorer mixture (air-fuel ratio went from 12.015:1 to 12.268:1) and the kinetics of NOx formation. They were also obtained because of the optimization, what was also obtained was: a 6% reduction in torque, a decrease of hydrocarbon emissions and an increase in the mass of gases burned inside the cylinder before combustion which regulates the amount of mixture admitted, as explained above, and which implied a fall in volumetric efficiency.

Table 2: Properties and parameters of operation for the turbocharged engine with the original cam profile and the aspirated engine with the optimized cam profile at 1500 RPM and 2 bar.

	Turbocharged	Naturally Aspirated Engine	Naturally Aspirated Engine Optimized	Percent Increase/Decrease	Index
BSFC [g/kWh]	425.79	413.88	320.79	22.5%	Decrease
Torque [N-m]	25.43	25.42	23.95	5.8%	Decrease
Volumetric Efficiency	24.80%	24.09%	18.86%	21.7%	Decrease
Hydrocarbon Emissions (ppm)	119.25	104.22	87.71	15.8%	Decrease
NOx within the cylinder at the time of opening of the exhaust valve (ppm)	0.05890	0.16947	0.25870	52.7%	Increase
Percentage of mass burned at the start of combustion (EGR + Residual)	25.9%	16.0%	26.9%	68.1%	Increase

The graph of the optimum lift curves for the intake and exhaust valves at 3500 RPM and 10 bar is plotted in Figure 8. The percentage reduction in fuel consumption for this case was 2.2%. The changes to arrive at this result were: anticipations in the opening and closing of the intake valve of 34.6o and 34.7o, respectively, and maximum elevation 25% lower; delays in the opening and closing of the exhaust valve of 21.5o and 5.4o of crank angle and an increase in maximum elevation of 6.9%. The less expressive increase of the overlapping period in relation to the previous case can be justified by the fact that, once the load and the speed increased, the need for burnt gases trapped in the cylinder decreased because of the greater power output needed to comply with the demand from the driver.

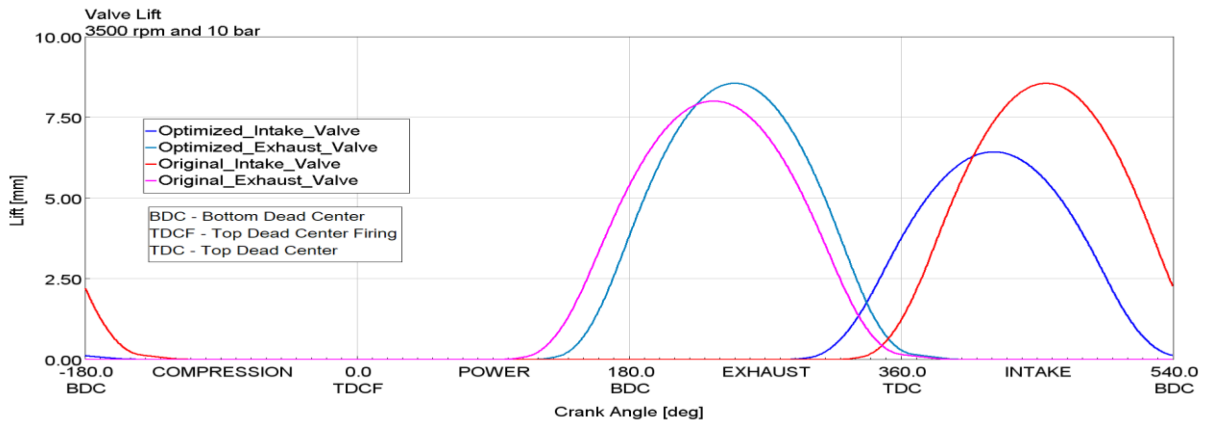


Figure 8: Valve lift curves before and after optimization for operation at 3500 RPM and 10 bar.

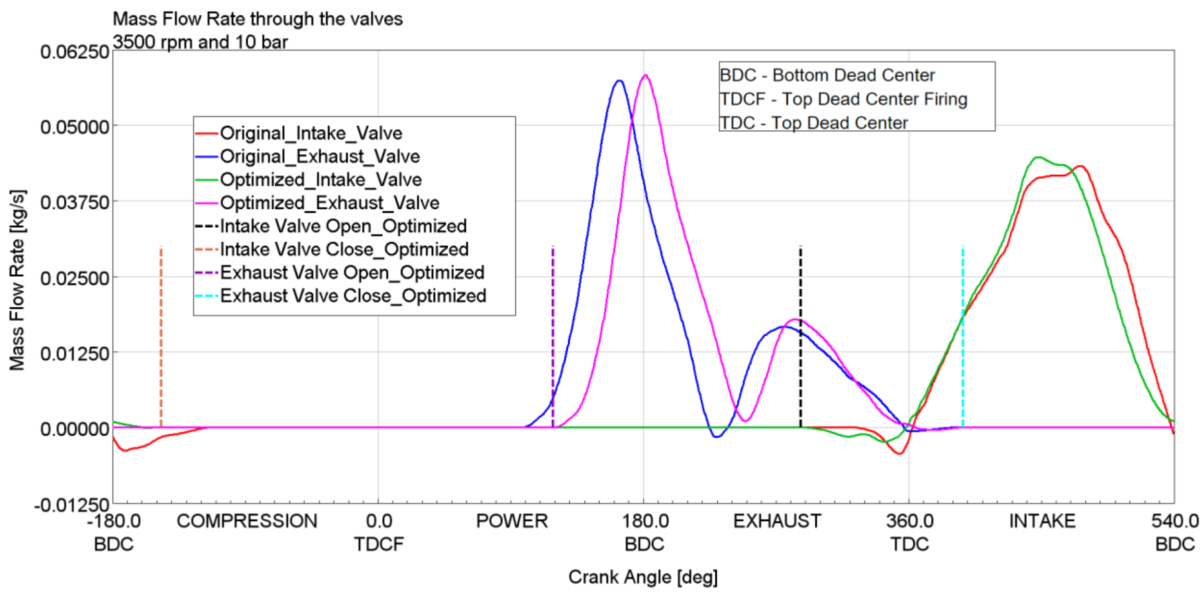


Figure 9: Graphic of the mass flow through the intake and exhaust valves before and after optimization at 3500 RPM and 10 bar.

The earlier opening of the inlet valve allowed the overlap period to be increased, but in this situation the reverse flow of exhaust gases through the exhaust valve was very small (Figure 9) because of the higher pressure inside the cylinder at the end of the exhaust stroke (1.904 bar) with respect to the pressure in the exhaust manifold (1.061 bar). On the other hand, the high pressure inside the cylinder during the interval in which the valves overlap favored the increase of the reflux period of cylinder gases towards the intake manifold, thereby increasing the pressure in the valve port by approximately 0.08 bar, despite the lower maximum flow rate related to the original case, and also contributing to increasing the amount of gas retained in the cylinder before combustion from 4.5% to 5.7% (the lowest percentage increase among all the operating conditions studied, since the increase in the average pressure on admission was very small). The p-V diagram in Figure 10 shows that both the increase in the amount of retained gases and the increase of pressure contributed at the beginning of the intake stroke to reducing the area of pumping losses, but the subsequent pressure drop caused a near overlap of the original engine curves and its optimized version, thus increasing the work of gas exchange.

The valve closing after and closer to the bottom dead center (crank angular position of 573o in the engine cycle) prevented the expulsion of cylinder gases, which occurred in the original engine case (Figure 9). However, the mass of air trapped in the cylinder fell from 369.3 mg to

361.6 mg, justified by the additional mass of flared gases (greater internal recirculation of burned gases). Thus, the volumetric efficiency fell by around 2%.

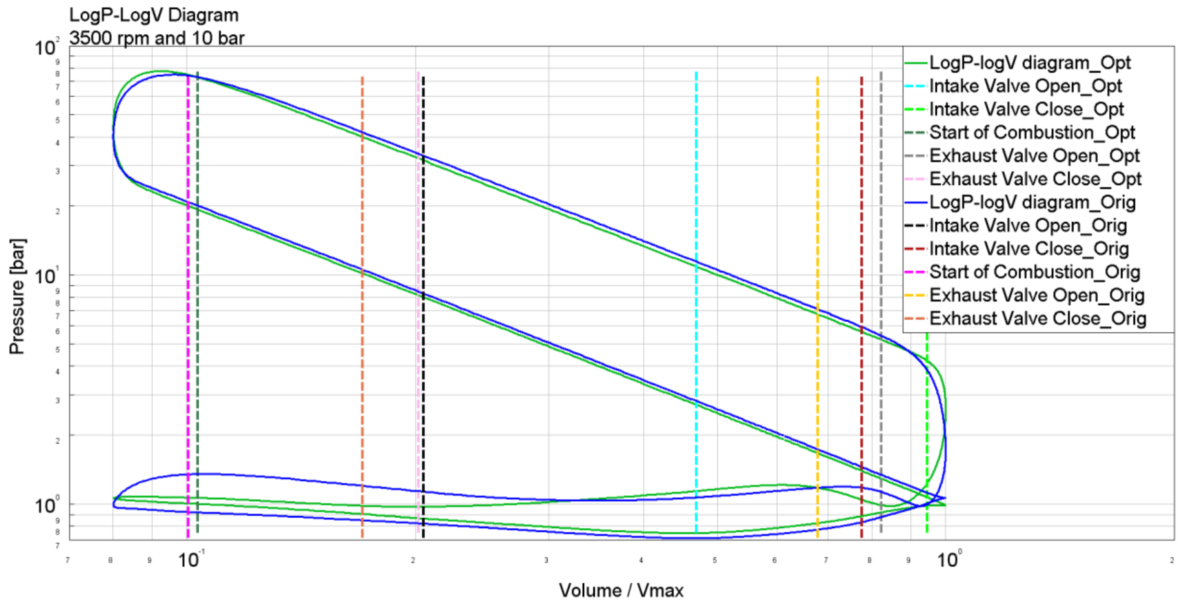


Figure 10: log(p)-log(V) diagram of the engine before and after optimization at 3500 rpm and 10 bar.

Related to the exhaust valve, its later opening compared to the original engine case allowed a better compromise between the expulsion of gas because of the blowdown and the absorption of the gas energy during the course of expansion, which helped to attenuate the specific fuel consumption and maintenance of the torque in the value of 127 N-m, despite the lower volumetric efficiency.[26]

Table 3 shows that the NOx emissions decreased by 55% and the hydrocarbons emissions increased by 2%. In the first case, the maximum temperature and pressure inside the cylinder does not serve as a basis to justify the reduction in NOx emissions since they rose from 2610 K to 2633 K and from 74.8 bar to 77.4 bar. One justification would be the smaller amount of air admitted, but the difference in mass between the original and optimized cases is so small that perhaps it would not have as much impact on the variation of the NOx concentration. Perhaps the NOx formation kinetics could better explain this variation, but this was not analysed in this study. As to the hydrocarbons emitted, the lower temperatures observed at the beginning and at the end of the cycle because of the dilution of the fresh mixture with greater mass of flared gases acted in the increase of the hydrocarbon concentration, in spite of the higher maximum temperature inside the cylinder during the whole cycle. Since the variation in concentration was very small, it is difficult to be sure what the predominant factor was for the changes that occurred.

Table 3: Properties and parameters of operation for the turbocharged and aspirated engine with the original cam profile and the aspirated engine with the optimized cam profile at 3500 rpm and 10 bar.

	Turbocharged	Naturally Aspirated Engine	Naturally Aspirated Engine Optimized	Percent Increase/Decrease	Index
BSFC [g/kWh]	293.10	276.74	270.70	2.2%	Decrease
Torque [N-m]	127.19	127.19	127.31	0.1%	Increase
Volumetric Efficiency	85.37%	80.61%	78.92%	2.1%	Decrease
Hydrocarbon Emissions (ppm)	115.48	111.23	113.14	1.7%	Increase
NOx within the cylinder at the time of opening of the exhaust valve (ppm)	0.07968	0.36537	0.16607	54.5%	Decrease
Percentage of mass burned at the start of combustion (EGR + Residual)	5.4%	4.5%	5.7%	26.7%	Increase

Table 4: Angular positions of the valve events before and after the optimization for all operation conditions studied.

		Intake valve opening	Intake valve closing	Exhaust valve opening	Exhaust valve closing
Original curves		322.0	608.0	98.0	392.0
Optimized curves	3500 rpm and 10 bar	287.4	573.3	119.5	397.4
	3000 rpm and 6 bar	263.3	532.6	105.7	434.0
	2500 rpm and 10 bar	289.5	568.2	127.2	395.8
	2500 rpm and 8 bar	272.2	553.1	122.7	395.6
	2000 rpm and 8 bar	272.5	540.3	116.8	400.0
	2000 rpm and 4 bar	251.2	518.4	110.2	396.3
	2000 rpm and 2 bar	247.8	501.8	103.3	385.8
	1500 rpm and 2 bar	262.2	517.9	122.6	371.7

Note: The delays are highlighted in red and the antecipation are in blue in relation to BDC (180°, 540°) and TDC (360°)

Tables 4 and 5 summarize all the results for the cases that were presented in Table 1. The brake specific fuel consumption was reduced in all operation conditions. Table 4 shows the crankshaft angles of the optimized intake and exhaust valve events. The colors blue and red are used to indicate anticipation or delays, respectively, regarding the dead centers of the engine. Table 5 shows the parameters of the engine analysed in the study. The percentage reductions relating to the operation with the original cam profiles are highlighted in red and with a negative sign and the increases are in blue.

Table 5: Percent increase/decrease of the engine parameters in relation to the operation with the original cam profiles.

	3500 rpm and 10 bar	3000 rpm and 6 bar	2500 rpm and 10 bar	2500 rpm and 8 bar	2000 rpm and 8 bar	2000 rpm and 4 bar	2000 rpm and 2 bar	1500 rpm and 2 bar
BSFC [g/kWh]	-2.2%	-5.4%	-2.1%	-4.4%	-4.2%	-11.9%	-21.6%	-22.5%
Torque [N-m]	0.1%	5.5%	0.2%	3.5%	0.0%	25.0%	44.8%	-5.8%
Volumetric Efficiency	-2.1%	-0.2%	-2.0%	-1.1%	-4.0%	9.5%	13.4%	-21.7%
Hydrocarbon Emissions (ppm)	1.7%	-3.0%	-3.4%	-1.6%	-2.8%	-7.5%	-6.0%	-15.8%
NOx within the cylinder at the opening timing of the exhaust valve (ppm)	-54.5%	218%	42.8%	33.4%	-11.6%	-5.5%	-19.2%	52.7%
Percentage of burnt mass at the start of combustion (EGR+Residual)	26.7%	120%	24.7%	84.1%	61.1%	77.6%	82.2%	68.1%

4. CONCLUSIONS

With the results generated from the methodology developed, it was concluded that, under the conditions of partial load operation (urban operating conditions), optimizing the valve event led to a maximum improvement of the specific fuel consumption of 22.5% (operation at 1500 RPM and 2 bar) when compared to the operation of the aspirated motor with the original cam profile of its camshaft. This gain was mainly explained by the increase in the valve overlap period, which led to an increase in the amount of gases burned inside the cylinder and an increase in the pressure in the intake valve port, thereby reducing the work of pumping gases into and out of the cylinder.

As a complement to undertaking the consumption analysis, parameters such as torque, NOx emissions, hydro- carbon emissions and volumetric efficiency were collected. The torque decreased only when operating at 1500 RPM and 2 bar, when the volumetric efficiency fell by about 22%. The oscillations in the NOx emissions were proportional to the maximum temperature obtained inside the cylinder, the amount of air admitted, the period available for oxidizing the fuel elements and the pressure. The emissions of hydrocarbons were inversely proportional to the maximum temperature obtained inside the cylinder, the amount of air admitted and the period available for oxidizing the fuel elements. However, other factors not included in the analysis, such as the kinetics of element formation and the turbulence level may have influenced the final composition of the combustion gases. The volumetric efficiency ranged proportionally to the amount of air admitted inside the cylinders, which was controlled by the internal recirculation of the burned gases.

The results obtained show the importance of applying a variable valve actuation system in a spark ignition engine in order to obtain a better specific fuel consumption, higher torque and power output when the engine is operated at full load.

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