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Development of an overhead camshaft system adapted to a SAE Supermileage single-person vehicle in a fuel economy perspective

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ABSTRACT
This article presents a comparative study between two camshafts systems adapted to the single cylinder engine of a Supermileage vehicle in a fuel economy perspective. One system is from a Honda AF70E engine and the other is a new design. The new camshaft system was improved for fuel economy by developing a new camshaft that enhances volumetric efficiency while reducing friction losses. The comparison was made by measuring the efficiency of the engine in the speed range where the engine was used by the Supermileage vehicle and a calculation was made to show which of the configuration is best for the vehicle.

KEYWORDS
Supermileage, Camshaft, Volumetric efficiency, Valve timing, Valve lift

1 INTRODUCTION

With environmental standards constantly increasing and public consciousness of the importance of sustainable development, improvement in energy efficiency is, and must be, at the center of interest of research and development of any product or application [1]. This is particularly true with technologies relying on fossil fuels, such as the energy and automotive industries. Many challenges ought to be met in the coming years and innovative solutions such as: CO_2 carbon capture, reduction in energy demand, smart grid technologies and alternative green energy sources (wind, hydro, solar) are now being implemented or tested [2], [3]. To raise awareness on these social and economic issues, education can play a major role and nowadays, many colleges and universities worldwide offer courses on sustainable development and energy efficiency to engineering students [4]. The Department of Mechanical Engineering at Université Laval is moving in this direction by offering students the opportunity to design a fuel-efficient vehicle built to the standards set by the SAE Supermileage rulebook. [5]. The main objective of this competition is to design a single-person vehicle that consumes as little fuel as possible. The vehicle from Université Laval used a Briggs and Stratton base (mandatory as per SAE Supermileage competition rules) paired with a AF70E Honda engine head. The Honda engine head, with its standard camshaft system, has been used in the past with great success by Laval's team, leading to a North-American record breaking performance of 1610 km/L in 2016 [6]. To improve the fuel efficiency of the vehicle, it was postulated that a custom camshaft system designed for fuel efficiency and retrofitted on the engine head could increase the kilometers per liter rating of the vehicle. This paper presents the mechanical improvements and results obtained as well as the methodology behind its design and testing.

2 ENGINE CONFIGURATION AND FUEL EFFICIENCY STRATEGY

The single cylinder Briggs and Stratton base paired with the Honda engine head was used with a custom-built crankshaft, connecting rod and cylinder. These parts have been re-designed to achieve different engine parameters than those of the original engine. The engine parameters are shown in Table 1. The use of isooctane fuel is required at SAE Supermileage competition and since the octane rate of isooctane is 100, a geometric compression ratio up to 16:1 can be reached before knocking occurs. In the proposed design, a geometric compression ratio of 13:1 was used.

Table 1 : Parameters of the modified Briggs and Stratton engine better suited for a Supermileage vehicle.

Engine parameter	Value
Stroke	60.2 mm
Bore	38 mm
Connecting rod length	112 mm
Displacement	68 cc
Geometric compression ratio	13:1

A run-kill strategy was adopted during the competition which involves using the engine only to accelerate the vehicle. During the accelerations, the engine is kept in a wide-open throttle (WOT) configuration such that the engine is used at full load. This race strategy was used since the drag of the vehicle was too low to allow constant speed with the engine used at WOT. Moreover, it is well known that an engine operated at WOT is more efficient [7]. The engine was coupled to an adjustable speed centrifugal clutch. The system was adjusted so that the torque that the clutch can transmit was higher than the output torque of the engine at the minimum speed of the vehicle on track. The clutch was then fully engaged and there was no slippage in the device. The clutch was connected to the rear wheel with a ratio of 13:125. The accelerations of the vehicle were carried out from 15 to 31 km/h. This allowed to respect the minimum average speed of 24 km/h imposed by the SAE Supermileage competition [5]. Because of the gearing ratio, during an acceleration, the engine speed went from 1500 to 3100 revolutions per minute (RPM).

3 THEORETICAL CONCEPT

3.1 Efficiency of an internal combustion engine

The efficiency of an internal combustion engine is directly related to its capacity to compress the air-fuel mixture [8], [9]. The geometric compression ratio (GCR) is an indicator that characterizes this capability to compress the air-fuel mixture. When calculated based on geometric parameters, the geometric compression ratio (C_G) is defined by Equation 1. This indicator is the ratio of the volume of the combustion chamber when the piston is at bottom dead center (BDC), V_{BDC}, on the volume of the combustion chamber when the piston is at top dead center (TDC), V_{TDC}.

$$C_G = V_{BDC} / V_{TDC} \tag{1}$$

However, even if the GCR is high, it is not sufficient to guarantee a good compression of the intake gases due to the dynamic effects of the flow especially when the piston is moving at high speeds. The volumetric efficiency (η_v) is a better indicator of the capability to compress the air-fuel mixture. It is defined as the ratio of the mass of intake gases entering the combustion chamber (m_{in}) to the maximum mass that can be supplied at atmospheric pressure and ambient temperature ($m_{in\ max}$) as expressed by Equation 2 [9].

$$\eta_v = m_{in} / m_{in\ max} \tag{2}$$

The increased volumetric efficiency will cause an increase in the mass of air-fuel mixture inducted during the intake stroke. It will also result in an increase of the peak compressed pressure and mean effective pressure of the cycle with the associated increase in efficiency.

3.2 Parameters of the camshaft affecting volumetric efficiency

The camshaft is the device that controls the opening of the intake and exhaust valves. The shape of the cams determines more precisely the timing of the valves' openings and closings. (referred to as the timing) and the lift profile of each valve (referred to as the lift). Due to the dynamic effects of the intake and exhaust flows, the valves lift profile influences the amount of mixture entering and exiting the combustion chamber. The two parameters of the camshaft that modify the dynamics of the flow entering and exiting the combustion chamber are the timing and the lift achieved by the valves [10], [11].

3.2.1 Valve timing

The timing of the valves represents the instants when each valve opens and closes. These instants are described with the angular position of the crankshaft relative to the extreme positions of the piston, namely TDC and BDC. As shown by Heisler and Heywood, the greatest influence of the timing is on the behavior of the VE as a function of engine speed [9], [12].

When the openings and closings are close to the dead centers, a good VE at low engine speed is favored [9], [12], [13]. At low engine speed, the combustion cycle occurs relatively slowly. The inertia of the inlet flow is small and the time to allow the mixture to enter the combustion chamber is relatively long. This means that when the piston is at the BDC, the combustion chamber is almost full of air-fuel mixture. The intake valve closes soon after the BDC. This leads to no or little fresh mixture discharged into the intake manifold while the piston is moving up [10]. This allows the compression of a maximum amount of air-fuel mixture and results in a maximum combustion efficiency. However, at high engine speed, when the timings are close to the dead centers, the VE is reduced because the time to allow the air-fuel mixture in the combustion chamber is reduced. The intake valve closes shortly after BDC during the compression stroke and the mass of mixture that has entered into the chamber is not as high as it could have been. This results in less gas to compress, leading to a lower efficiency.

When the timings are shifted away from the dead centers, VE at high engine speed can be favored [9], [12]. At high engine speed, there is less time to bring the air-fuel mixture into the combustion chamber. At the BDC after the intake stroke, the flow has not been accelerated enough to completely fill the combustion chamber, but the flow has acquired momentum. Hence, even if the piston begins to go up at the beginning of the compression stroke, air continues to enter because the valve is still open and because of the kinetic energy acquired by the flow. Following this logic, by closing the intake valve later after the BDC, better VE can be achieved at high engine speeds. However, when operated at low engine speeds, the combustion chamber is full of mixture when the piston is at the BDC. As the piston begins to rise, the mixture is moved back into the intake manifold. This means that at low engine speeds, the mass of air-fuel compressed is not optimal so the VE is reduced and the actual engine efficiency is lower. Therefore, by opting for timings that are close to the dead centers, the VE is higher at low engine speeds while it is lower at high speeds. Conversely, when the intake valve opening is advanced from TDC and closing retarded after BDC, VE is improved at high engine speeds, but reduced at low speeds.

3.2.2 Valve lift

The lift represents the measurement of the valve opening where a lift value of zero is associated with a closed valve. The gases entering or exiting the combustion chamber must pass around these valves. Valves have a throttling effect in the manifolds because they force a sudden narrowing of the curtain area. This narrowing in the manifolds around the valves forces gases to reach higher velocities. Since the pressure drop increases with the square of the speed, there are a significant pressure drops around the valves [9]. Also, the engine speed defines the flow rate entering the combustion chamber. Hence, the pressure drops around the valves are highly related to the valve lift and to the engine speed.

VE is related to the pressure drops because the latter are the cause of a decrease in pressure into the combustion chamber as mentioned by Osman [7]. This decrease in pressure results in a decrease in the capability to compress the air-fuel mixture. Hence, decreasing pressure drops allows better VE to be achieved. According to Heywood [9], the following conclusion can be drawn: the more the valve lift increases, the more the VE is increased, but the higher the lift, the less an augmentation in lift is significant on the gain in VE. This is due to the fact that the more the valve opens, the less the increase in curtain area increases. This conclusion can be expanded to assume that there is next to no gain in extending valve lift to a point where the curtain area exceeds the cross sectional area of the port/valve seat itself, as the valve seat then becomes the limiting area.

In this paper, the work focused on improving the efficiency of the engine by improving its VE and its camshaft mechanical system. It was postulated that modifications of the camshaft (new timings and lift values for the intake and exhaust valves) would improve the VE.

3.2.3 Combustion cycle

In this article, the combustion cycle of an internal combustion engine is referenced in terms of the crankshaft angle as shown in Figure 1. In this figure, a typical valve lift profile is shown to present the evolution of the valve opening in the combustion cycle.

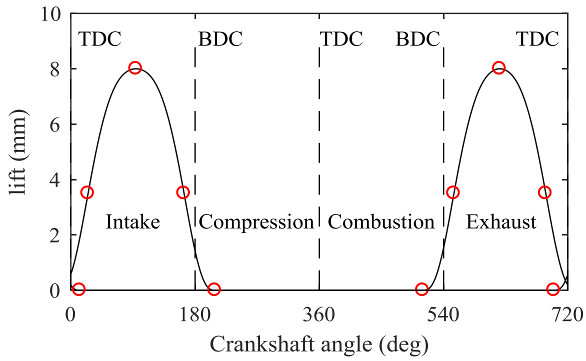

Figure 1 : Representation of the combustion cycle referenced with the crankshaft angle. The valve lift profiles are illustrated to demonstrate the evolution of the valves' lift into the combustion cycle. Red dots represent the control points that were used to generate the functional valve lift profiles.

4 CAMSHAFT SYSTEM DESIGN

4.1 Analysis of the original Honda camshaft

The cam profiles of the original Honda camshaft used with the AF70E head were measured with a Coordinate Measuring Machine (CMM) with a density of 1.5 points/deg. The valve lifts were obtained by performing a motion simulation with the entire assembly. The timings and valve lift characteristics of the original Honda assembly are shown in Table 2.

Table 2 : Original timing and lift of the AF70E engine head.

	Timings (°)		Maximum valve lift (mm)
	opening	closing	
Intake	24 before TDC	44 after BDC	5.8
Exhaust	41 before BDC	36 after TDC	5.6

By looking at Heywood's figure on VE as a function of engine speed for a camshaft with different timings [9], it is clear that the timings of the Honda camshaft favors maximum efficiency at high speeds (about 4000 RPM). It was therefore postulated that there was room for improvement of the VE at low engine speed.

In competition, because of the run-kill strategy and the specific vehicle configuration, the engine was run from 1500 to 3100 RPM. Under these conditions, what was sought by the design of the camshaft was a good VE over this range of engine speeds to improve the overall engine efficiency during an acceleration.

4.2 Timing design

To improve VE over the desired engine speed range, timings were selected according to Heywood's figure of the VE as a function of engine speed for a camshaft with different timings [9]. The proposed timings for the new camshaft are shown in Table 3. As it is expected, timings close to dead centers allow high VE to be achieved at low speed.

Table 3 : Proposed timings for the new camshaft.

	Timings (°)	
	opening	closing
Intake	15 before TDC	20 after BDC
Exhaust	30 before BDC	10 after TDC

By looking the timings in the new design (Table 3) we can see that the intake valve closes earlier than with the original design (Table 2). The exhaust valve in the new design (Table 3) is also opened latter than with the original design (Table 2), this allows the power stroke to be increased and more work to be produced by the combustion. There is also less overlap (time when the intake and exhaust valves are open at the same time) with the new design. On the original profile there was an overlap of 60° compared to 25° overlap with the new design. This reduces the amount of unburned mixture going straight into the exhaust at low engine speeds.

4.3 Valve lift design

Since the valve lift influences the pressure drops, the selection of the lift was done by modeling an index of pressure drop as a function of the lift and engine speed. The performance index represents the average of the flow velocity around the valves during the stroke (the intake stroke for example).

The average velocity around the valves during a stroke (v_{avg}) was approximated by calculating the instantaneous maximum theoretical flow velocity (v_i) around the valve (VE=100%) assuming no losses in the intake port. Calculations were made at each 0.5° for the entire stroke and dividing by the number of calculated instantaneous mean flow velocities (n), as shown in Equation 3, where v_1 is the mean flow velocity when the valve opens and v_n is the flow velocity when the valve closes. The opening and closing timings were those presented in Table 3.

$$v_{avg} = \frac{\sum_{i=1}^{n} v_i}{n} \quad (3)$$

The instantaneous mean flow velocity was obtained by dividing the instantaneous flow rate by the instantaneous curtain area. The flow rate was determined by assuming conservation of mass. It was possible to calculate the instantaneous speed of the piston using the geometric dimensions of the engine (crankshaft radius, piston diameter and rod length). Knowing the piston diameter, it was then possible to obtain the incoming flow rate as a function of the speed of the engine for each position of the crankshaft (Q_i). The calculation of Q_i is presented in Appendix A. Heywood presents a model to calculate the curtain area around the valve as a function of the valve lift (A_i) [9]. By knowing the instantaneous flow rate and the instantaneous curtain area, it was possible to calculate the instantaneous mean flow velocity around the valves for each crankshaft angle in function of the engine speed (r) and valve lift (l) with Equation 4:

$$v_i\,(r,l) \;=\; Q_i \,/\, A_i\,. \tag{4}$$

The average of the flow velocity around the intake valve during the intake stroke, calculated with Equation 3, is shown in Figure 2 as a function of the engine speed and the maximum valve lift for a given valve profile. This figure shows that, as the engine speed increases, the average flow velocity increases. Also, the more the maximum valve lift increases, the less an additional increase in valve lift has an influence on the reduction of average flow velocity.

According to Figure 2, a relation can be made between the pressure drop and the VE. As described previously, VE and the pressure drops behave oppositely under conditions dictated by valve lift and engine speed as described by Heywood [9]. This supports the assumption that the impact of the lift and engine speed on the reduction of VE is caused by an increase in the pressure drops in the system. Therefore, the advantage of increasing valve lift reduces as the VE approaches 100%.

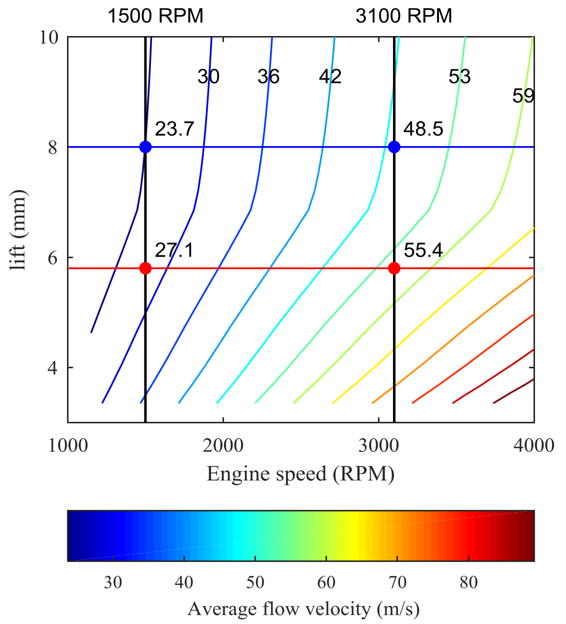

Figure 2 : Average flow velocity around the intake valve during the intake stroke as a function of engine speed and the maximum valve lift. The blue horizontal line shows the lift on the new improved design and the red horizontal line shows the lift of the original Honda system.

Following this analysis, a maximum lift of 8 mm was selected for both valves (intake and exhaust). The modeling showed that an increase from 5.6 mm and 5.8 mm to 8 mm allowed a significant reduction of the average flow velocity, which is directly related to the pressure drop (see red and blue line, FIG 2). Also, as shown in Figure 2, the potential improvements are much less significant for a maximum lift higher than 8 mm. Finally, since the valve spring used can withstand an 8 mm maximum lift, it was decided to limit the valve maximum lift to 8 mm.

4.4 Rocker design

The system used with the original Honda head was a simple overhead camshaft. With this configuration, rocker arms were used to transmit the movement of the camshaft to the valves. The original rockers ratio was 1.5. The system designed by Honda is shown in the Figure 3-a. The original rocker arms were designed so that the cam/rocker interface was made between two fixed surfaces. It was estimated that the friction losses at these interfaces represented about 6% of the work done at the crankshaft. The losses were calculated by comparing the energy lost at the cam/rocker interfaces and the maximum energy done by the engine on a cycle. Details about the calculation are presented ~~added~~ in Appendix B.

To minimize losses at the interface between cams and rockers arm, the new design integrated a deep groove ball bearing as followers (see blue bearing in Figure 3-b). By using roller contacts, almost all of the friction losses at the cam/rocker interface were eliminated as it can be shown by [14]. Details about the calculation of friction losses are presented in Appendix B. The rollers have no effect on the shape of the efficiency curve as a function of the engine speed. It only increases overall efficiency throughout the entire speed range of the engine as friction losses are known to be independent of velocity. New rockers ratio was 1.2.

Moreover, since the lubrication was not optimal in the engine, bushings were used at the pivots points of the rocker arms (yellow parts shown in Figure 3-b). This also contributed to minimize friction losses. The points of contact between the rocker arms and the valve were kept unchanged (in black). These contacts were made between two pieces of hardened steel and there was little relative displacement between the two components.

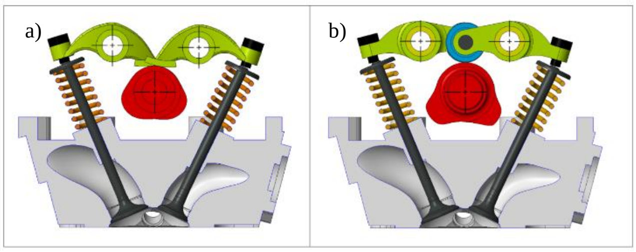

Figure 3 : CAD representation of the camshaft system. a) Original Honda design. b) Improved design with new valve timing, lift profile and new rocker design.

4.5 Cam profile generation

Based on the determination of the parameters (timings and lift), the valve lift profiles were generated according to the state-of-the-art principles described by Norton [15]. To generate a profile, 5 points were imposed and polynomials splines of fifth-order were drawn between the points by a custom-made program. An example is shown in Figure 1. The red dots represent the spline control points and the lines are the lift profiles of the valves that were generated. The first and last points were fixed at zero and the central point was placed at the maximum lift (8 mm). Knowing the valve lift profile and the rocker arms parameters, the cam profiles were generated following the method shown by Norton. Lastly, to ensure that the camshaft was functional, the following elements were also checked:

- The spring force applied was sufficient to avoid any loss of contact between the cam and the rocker arm.
- The angle of contact between the rocker arm and the valve was always less than 30° (principle stated by Norton [15]).

- There was no interference between a valve and the piston.

5 EXPERIMENTAL SETUP

5.1 Controlled parameters

The experimental tests aimed at comparing the two camshaft assemblies (original Honda and the new improved design) from the standpoint of energy efficiency. It was decided to characterize the efficiency of the engine as a function of the engine speed because different VE behaviors were expected for each camshaft system.

The index used to measure efficiency was the specific torque. It is determined by dividing the average output torque (T) produced at the crankshaft by the amount of gasoline injected (m_{inj}) at each engine cycle as shown in Equation 5.

The specific torque is proportional to the efficiency since the average output engine torque multiplied by 4π represents the output energy of the engine on a cycle and the amount of gasoline injected multiplied by the calorific value represents the input energy on a cycle. The ratio of output to input energy is known to be the definition of efficiency.

$$ST = T / m_{inj} \tag{5}$$

The following engine parameters were controlled: the amount of fuel injected at each combustion cycle and the moment at which the spark occurred. The latter was called the spark timing because the moment when the spark was controlled is best described using the crankshaft angle. The timing is described in crank angle Before TDC (BTDC). For both camshaft systems, the engine parameters to achieve the maximum efficiency were expected to be different since the amount of air admitted per cycle was not the same.

5.2 Torque measurement

Torque measurement was made with a homemade test bench adapted to the needs of a small Supermileage-type engine. On this test bench, a magnetic brake was used to keep wheel speed constant with the engine running. A load cell (OMEGA LC101-25) measured the braking force applied. Since the speed was constant, the torque applied by the magnetic brake on the wheel directly represented the input torque applied by the engine. The torque at the engine was measured with an accuracy of ± 0.05 Nm. This accuracy was obtained from an uncertainty analysis of the measurement data.

5.3 Measurement of the mass of gasoline injected

The gasoline injection was electronically controlled by a microcontroller with a custom-made program that was intended for small engine control (NXP KIT33812ECUEVME). The injector used was the original Honda AF70E engine part (Honda 16450-GGL-J01). The injection time (t, in µs) was controlled in the engine control program and this value was assumed to be proportional to the quantity of fuel injected (m_{inj}, in µg). An experimental setup was made to establish the relationship between this injection time and the mass of gasoline injected with an electronic scale. This relationship was established prior to the test campaign on the engine. The relation is shown in Equation 6. The precision on the mass of gasoline injected was known with an accuracy of 1%.

$$m_{inj} = 0.4669\, t - 198.1 \tag{6}$$

5.4 Other physical quantities measured

The efficiency of an engine is known to depend on the ambient air and engine temperature. For example, the volumetric efficiency changes depending on the temperature and air inlet pressure [9]. In order to carry out valid comparative tests, the two camshaft assemblies were compared under the same operating conditions. The tests were carried out in a room at a temperature of 22°C and a controlled cooling system was installed on the engine to maintain its temperature at 100 °C.

5.5 Measurement methodology

The specific torque was characterized as a function of the engine speed for the two camshaft systems by proceeding as follows:

1. For a fixed engine speed and spark timing, the mass of fuel injected was varied by increments of 100 µg for the entire domain where the engine was running. The specific torque was noted with a sample of 10 measurements for each increment of fuel mass injected.
2. The spark timing was varied by increments of 5°. We returned to step 1 until all the spark timing values where the engine was running were analyzed, then we went to step 3.
3. The engine speed was incremented by 200 RPM. We return to step 1 until all the speed values where the engine was running were tested, then we went to step 4.
4. The camshaft setup was changed and the experiment was restarted (step 1). When both camshaft systems were tested, the test campaign was completed.

6 EXPERIMENTAL RESULTS

6.1 Results

The test campaign enabled the mapping of the engine parameters for the entire engine speed range for both camshaft systems. An example of a mapping is shown in Figure 4. In this figure, the crosses represent the measurement points. The point cloud obtained was imported into a MatLab data processing script to obtain the iso-contours.

In this figure, it is very interesting to note the relation between the quantity of fuel injected and the spark timing. On the black line shown in this figure, the specific torque varies very little. This means that for several different engine parameters, it is possible to measure a quasi-constant specific torque. It should be noted that by increasing the quantity of fuel injected, it is necessary to delay the spark (increase the controlled spark timing) to maintain the specific torque quasi-constant.

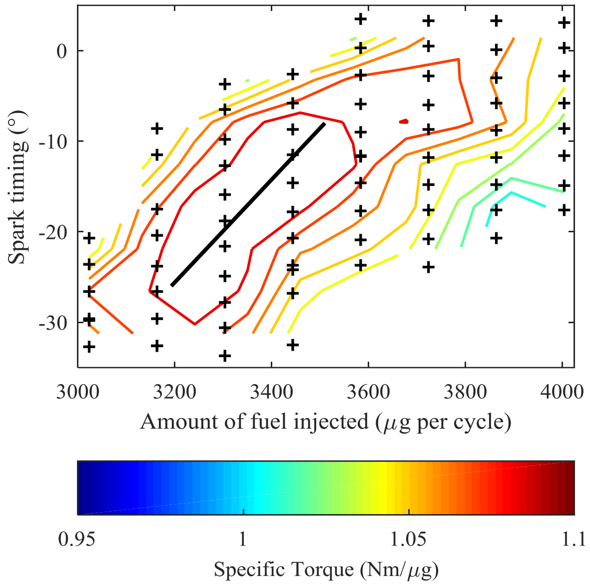

Figure 4 : Example of a mapping of the specific torque as a function of the quantity of fuel injected and the spark timing (BTDC); This figure is for the new camshaft obtained for a speed of 2500 RPM.

For each engine speed and each camshaft design, the value of the quantity of fuel and the spark timing that yielded the highest specific torque were recorded. These values were used in Figure 5 to show the optimal spark timing as a function of engine speed (Figure 5-a), the optimal quantity of fuel as a function of engine speed (Figure 5-b), and the maximum specific torque as a function of the engine speed (Figure 5-d). The engine torque at the optimal efficiency parameters is also shown in Figure 5-c.

For both systems, as the engine speed increased, the spark timing tended to occur earlier in the cycle. This is what was expected as the flame propagation velocity remained almost unchanged, but the speed of the piston increased as the engine speed increased. Thus, the spark timing must have occurred earlier to obtain the maximum pressure in the combustion chamber at the TDC.

For each of the two systems, there was a sudden change in the spark timing at a certain speed. In the case of the original camshaft system, at 2700 and 2900 RPM, there was a drop and there was a sudden increase in the case of the improved camshaft design after 3300 RPM. These sudden changes may seem strange. However, by observing Figure 5-b, we note the opposite phenomenon for each of the two camshaft systems at the exact same speed. Indeed, at 2700 and 2900 RPM there was a decrease in the amount of fuel injected for the original camshaft and there was an increase in the new system beyond 3300 RPM. These variations can be explained by the relation developed in Figure 4. It was shown that for several combinations of spark timing and quantity of gasoline injected, the specific torque may remain almost constant.

Figure 5-b shows that, in general, the optimum amount of fuel to be injected for the new camshaft system appears to be slightly lower (apart from the discrepancies just discussed). This is an observation that is in disagreement with what was expected. Indeed, by improving the VE with the new camshaft profiles, it was expected that slightly more fuel be needed with the new system since the air quantity admitted in the combustion chamber was increased. On the other hand, as observed in Figure 5-a, the spark timings of the new system were always lower. According to the relationship developed in Figure 4, this would explain why the quantity of optimum fuel to be injected was always lower with the new system. The change in VE between both camshaft systems cannot be judged by the amount of fuel injected. Note that to properly characterizes the VE as a function of the engine speed, the inlet flow should have been measured with a flow meter. This was not done in this work, but could potentially be interesting for future studies.

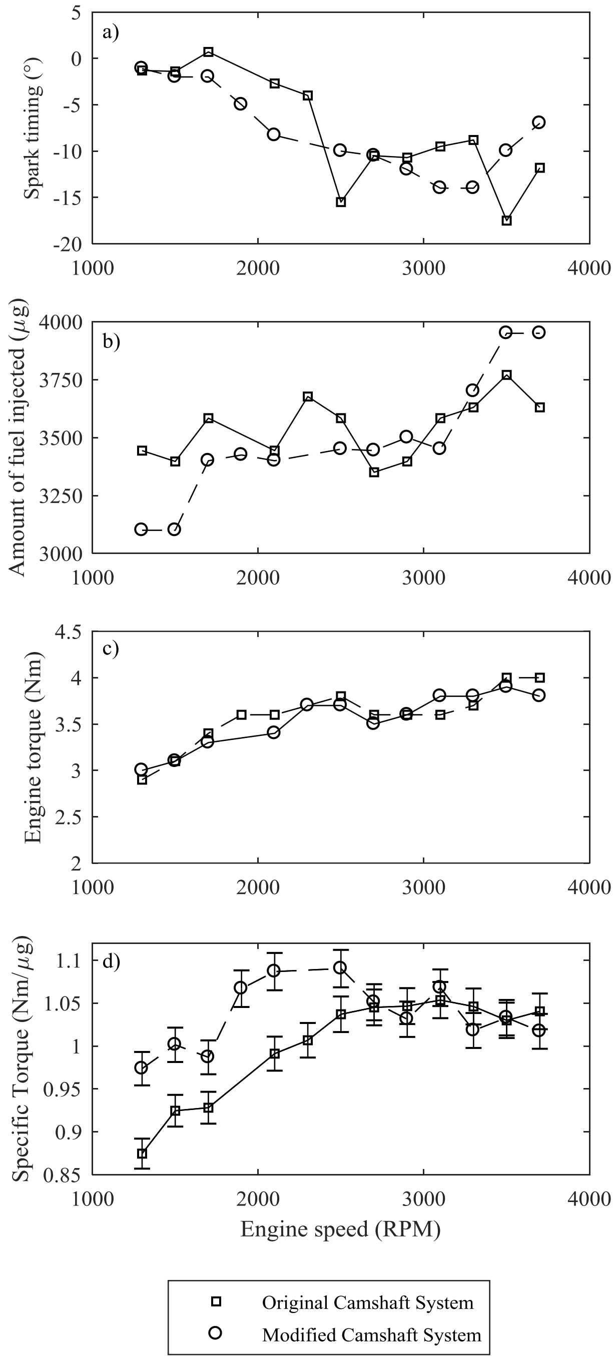

Figure 5 : Results of experimental tests. a) Spark timing as a function of engine speed that allow best specific torque. b) Amount of fuel to inject that allows best specific torque. c) Engine torque at the best specific torque. d) Best specific torque.

The optimum specific torque measurement, which is directly related to the efficiency, as a function of the engine speed for each of the camshafts are shown in Figure 5-d. For the original design, the maximum efficiency was reached around 3000 RPM. For the new camshaft system, the maximum efficiency is around 2000 - 2500 RPM. Also, the maximum efficiency achieved with the new system is superior to the original Honda camshaft system. The efficiency at low speed was improved by about 10%. While, at relatively high engine speeds (beyond 3000 RPM), the new system becomes less efficient than the original design, we reached our goal of increasing the efficiency at lower speeds.

6.2 Results analysis

To conclude that the new camshaft system improved for fuel efficiency was better suited to the Supermileage vehicle, it needed to effectively consume less fuel during an acceleration. With the results of the dynamometer tests, an approximation of the quantity of fuel to be injected for an acceleration carried out on the track was calculated. Bounds of 200 RPM were made between measurements. The amount of fuel injected (q_r in micrograms) to jump between two engine speeds where measurements were taken (from 1900 to 2100 for example) was estimated with Equation 7:

$$q_r(i) = m_{in}(i) \times n_{inj}\ (i) \times \Delta_t(i), \quad (7)$$

Where *i* is the RPM range, m_{in} represents the experimental optimal value of fuel to inject for an engine speed range [mg] (see FIG 5-b), n_{inj} represents the number of injection per second [nb/s], and Δ_t represents the time spent at this engine speed range [s]. Δ_t was calculated with Equation 8:

$$\Delta_t = m \times \Delta_v / F_r, \quad (8)$$

where Δ_v is the speed variation of the vehicle [m/s], which is proportional to engine speed variation, *m* is the mass of the vehicle [kg], and F_r the force exerted on the wheel to propel the vehicle [N]. The latter can be evaluated with the torque at the engine crankshaft and with the transmission ratio. The wheel radius is 0.25 m and the mass of the vehicle is 105 kg.

The total amount of fuel injected during an acceleration ($q_{r\ tot}$) was estimated with Equation 9:

$$q_{r\ tot} = \sum_{i=\ RPM\ init}^{RPM\ end} m_{inj}\ (i) \times n_{inj}(i) \times \Delta_t(i). \quad (9)$$

It must be noted that this model does not consider the amount of gasoline needed to run the engine up to 1500 RPM. On the other hand, since the efficiency at low engine speeds was improved with the new camshaft system, it was expected that this amount of gasoline would be lower than with the original system. Therefore, this approach underestimates the efficiency improvement. Also, the first acceleration was not modeled. Initially, the vehicle starts from rest. Again, since the efficiency at low engine speeds was improved, it was expected that with the new camshaft design, this acceleration would consume even less gasoline than with the original camshaft system.

The quantities of fuel injected for an acceleration were calculated for a speed variation of the vehicle from 15 to 31 km/h as it is representative of typical accelerations carried out on track during competitions. The calculation for each of the camshaft systems gave an amount of 204 milligrams (new camshaft system) and 216 milligrams (original camshaft system) for this acceleration. The relative gain in efficiency was estimated to be close to 5.5%. The improved efficiency at low engine speed with the modified camshaft system is better suited for the Supermileage vehicle as the amount of fuel to inject was lower for each acceleration. This ultimately allowed the vehicle to consume less fuel during a test on track. Indeed, at the 2017 edition of the SAE Supermileage, where the new modified camshaft design was used for the first time, a fuel consumption of 1749 km/L was recorded surpassing the 1610 km/L mark recorded in 2016. The use of the new modified camshaft was the main change on the vehicle between the 2017 and the 2016 editions. The overall improvement in efficiency based on the results recorded in competition is of 8.6%

7 CONCLUSION

This paper presents a new camshaft system especially adapted to the engine of a Supermileage vehicle. The goal of the redesign was to reduce the overall fuel consumption per km.

The design of the new system was based on improving volumetric efficiency, mainly at low engine speeds, as well as the reduction of friction losses in the system. The volumetric efficiency was improved by modifying the timings and the lift of the valves (intake and exhaust) through the design of a new camshaft. The friction losses were reduced by using a roller contact between the camshaft and the rocker arm and by using bushings at the pivots of the rocker arms

With these modifications, the dynamometer tests and an approximation calculation predicted an improvement in the on-track consumption of the order of 5.5%. Ultimately, at the 2017 SAE Supermileage competition, four fuel economy runs were recorded. The first one with a fuel consumption of 1381 km/L, the second one with a consumption of 1720 km/L, the third one with a consumption of 1749 km/L (new North-American record), and a last one with a consumption of 1735 km/L. The first attempt must be discarded due to a problem with the Supermileage's pilot. The best fuel consumption recorded represents an improvement of 8.6% compared to the previous record established in 2016 when the Honda system was used. The dynamometer and track tests thus showed that the new camshaft design is better suited for the engine of a Supermileage vehicle.

8 REFERENCES

[1] "Framework for the UN DESD international implementation scheme; 2006." [Online]. Available: http://www.unesco.org/ulis/cgi-bin/ulis.pl?catno=148650&set=0059ED04EE_1_252&gp=1&lin=1&ll=1. [Accessed: 22-Oct-2017].

[2] S. L′Orange Seigo, S. Dohle, and M. Siegrist, "Public perception of carbon capture and storage (CCS): A review," *Renew. Sustain. Energy Rev.*, vol. 38, no. Supplement C, pp. 848–863, Oct. 2014.

[3] M. L. Tuballa and M. L. Abundo, "A review of the development of Smart Grid technologies," *Renew. Sustain. Energy Rev.*, vol. 59, no. Supplement C, pp. 710–725, Jun. 2016.

[4] B. Karatzoglou, "An in-depth literature review of the evolving roles and contributions of universities to Education for Sustainable Development," *J. Clean. Prod.*, vol. 49, no. Supplement C, pp. 44–53, Jun. 2013.

[5] "Rules - SAE Supermileage - SAE Collegiate Design Series - Students - SAE International." [Online]. Available: http://students.sae.org/cds/supermileage/rules/. [Accessed: 22-Oct-2017].
[6] "Results - SAE Supermileage - SAE Collegiate Design Series - Students - SAE International." [Online]. Available: http://students.sae.org/cds/supermileage/results/. [Accessed: 22-Oct-2017].
[7] O. A. Kutlar, H. Arslan, and A. T. Calik, "Methods to improve efficiency of four stroke, spark ignition engines at part load," *Energy Convers. Manag.*, vol. 46, no. 20, pp. 3202–3220, Dec. 2005.
[8] T. G. Leone *et al.*, "The Effect of Compression Ratio, Fuel Octane Rating, and Ethanol Content on Spark-Ignition Engine Efficiency," *Environ. Sci. Technol.*, vol. 49, no. 18, pp. 10778–10789, Sep. 2015.
[9] J. Heywood, *Internal Combustion Engine Fundamentals*. McGraw-Hill Education, 1988.
[10] A. H. Kakaee, B. Mashadi, and M. Ghajar, "A novel volumetric efficiency model for spark ignition engines equipped with variable valve timing and variable valve lift Part 1: model development," *Proc. Inst. Mech. Eng. Part J. Automob. Eng.*, vol. 231, no. 2, pp. 175–191, Feb. 2017.
[11] M. Ghajar, A. H. Kakaee, and B. Mashadi, "Semi-empirical modeling of volumetric efficiency in engines equipped with variable valve timing system," *J. Cent. South Univ.*, vol. 23, no. 12, pp. 3132–3142, Dec. 2016.
[12] H. Heisler, *Advanced Engine Technology*. E. Arnold, 1995.
[13] E. Sher and T. Bar-Kohany, "Optimization of variable valve timing for maximizing performance of an unthrottled SI engine—a theoretical study," *Energy*, vol. 27, no. 8, pp. 757–775, Aug. 2002.
[14] D. C. Sun and R. C. Rosenberg, "An Experimental Study of Automotive Cam-Lifter Interface Friction," *E Trans.*, vol. 30, no. 2, pp. 167–176, Jan. 1987.
[15] R. L. Norton, *Cam Design and Manufacturing Handbook*. Industrial Press Inc., 2009.

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APPENDIX A

The incoming flow rate as a function of the speed of the engine for each position of the crankshaft (Q_i) is estimated with Equation 10:

$$Q_i = \frac{\pi}{4} d^2 \times \left(r \sin\alpha + \frac{r^2 \sin\alpha \cos\alpha}{l\sqrt{1-(r/l)^2 \sin^2\alpha)}} \right) \times \frac{2\pi}{60} RPM \,, \tag{10}$$

where d is the piston diameter, r is the crankshaft radius, l the rod length, α is the crank angle (0 deg is referenced a TDC), and RPM is the engine speed en RPM.

APPENDIX B

Energy lost at cam/rocker interface with flat surface (E_f) is estimated with Equation 11:

$$E_f = 2 \sum \mu\, F_c * \Delta\omega\, r_c \,, \tag{11}$$

where μ represents the friction coefficient (steel to steel, with lubrication), F_c is the contact force (known with spring stiffness and cam profiles), $\Delta\omega$ is the angular variation between two calculations (here 0.5 deg), and r_c is the instantaneous radius of the cam. The summation is done for one camshaft rotations. The factor 2 is for the two cam/rocker interface (intake and exhaust). With the flat surface cam/rocker interface, energy lost is about 3.2 J for both interface during one engine cycle.

Energy lost at cam/rocker interface with roller contact (E_r) is estimated with Equation 12:

$$E_r = 2 \sum M * \frac{r_c}{r_b} \Delta\omega \,, \tag{12}$$

where M is the frictional moment of the bearing as specified by the manufacturer, r_c is the instantaneous radius of the cam, r_b is the radius of the bearing, and $\Delta\omega$ is the angular variation between two calculations (here 0.5 deg). The factor 2 is for the two cam/rocker interface (intake and exhaust). Note that M depends of the cam profile and the spring stiffness, which are known. With the roller contact interface, energy lost is about 0.1 J for both interfaces during one engine cycle.

The energy done by the engine is estimated by multiplying the maximum average engine torque (4 Nm, output torque measured at the crankshaft) by 4π (an engine cycle is two rotations) so the energy done by the engine on one cycle is 50.3 J.