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Optimization of the Fuel Efficiency of the M3165 Internal Combustion Engine in Transitory Operation

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Abstract
This paper describes the methodology used to optimise the fuel conversion efficiency of the M3165 internal combustion engine in transitory operation for application in a high energetic efficiency vehicle for competing in the European Shell Eco-marathon 2010. The M3165 engine is a four-stroke spark ignition engine working according to the Miller-Atkinson cycle. Before performing experimental engine testing a computer program simulating the engine operation was used to perform a sensitivity analysis and identify the operating and design variables with more influence in the fuel efficiency of the engine. To measure the performance and fuel efficiency of the engine in transient operation a methodology was developed. The methodology used takes into account the inertia of the engine, of the transmission system and of the vehicle. Once the measurement method was established operating parameters of the engine were optimized experimentally in the range of application of the engine. These include the injection duration, ignition timing, cylinder head temperature and oil temperature. The injection duration and ignition timing were optimized for each engine rotation speed in the operating range of the engine.

1. Introduction
The application of an engine to drive a vehicle competing in the Shell Eco-marathon implies that its operation is transient. In this competition due to the high energetic efficiency of the vehicles these are operated in a start-stop mode. A typical lap has between 2 km and 4 km, and the engine is turned on between 1 and 5 times in a lap and each time the engine is turned on is between 2 s and 10 s. The best strategy is chosen by each team according to the characteristics of their vehicle, the topology of the circuit and the atmospheric conditions present during the competition. Most of the vehicles use a transmission system with a single transmission ratio. During the time the engine is on, its operation is transitory with the engine speed increasing as the vehicle speed increases. To obtain a good performance the engine must be optimized for fuel efficiency in the same transitory conditions as those encountered during the competition.

2. Optimization Strategy
The Shell Eco-marathon Europe 2010 competition was held in the EuroSpeedway Lausitz racing circuit. The distance covered in the race corresponds to eight laps to the circuit minus the distance between the starting line and the finishing line. The perimeter of the circuit is 3200 m and the distance between the starting line and the finishing line is 115 m [1]. The total distance covered in the race is then given by Eq. (1)

$$d_{trial} = \sum_{k=1}^{8} d_{lap,k} = 7 \times 3200\,\mathrm{m} + (3200 - 115)\,\mathrm{m} = 25485\,\mathrm{m} \tag{1}$$

The performance of the vehicle in the competition is expressed in km/L and is calculated dividing the distance covered by the vehicle in a trial by the volume of fuel at a reference temperature equal to 15ºC correspondent to the mass of fuel consumed in the trial and is given by Eq. (2).

$$P[\mathrm{km/L}] = \frac{d_{trial}[\mathrm{m}]\,\rho_f(T_0)[\mathrm{kg/m^3}]}{m_{f,trial}[\mathrm{kg}] \times 10^6} \tag{2}$$

The mass of fuel consumed in a trial is the sum of the mass of fuel consumed in all times the engine is turned on. For the strategy used in the race there are two types of connections of the engine: There is only one connection of the first type which is the first connection in a trial where the vehicle accelerates from rest to 28.9 km/h; The other connections in a trial are of the second type and are the normal connections during the race where the vehicle accelerates from 24.4 km/h to 38.0 km/h. In laps 1 to 7 are made 3 normal connections per lap and in lap 8 are made two normal connections. In the first connection the engine is turned on and in the first phase of this connection the engine works at an engine speed slightly above that where the centrifugal clutch starts to transmit torque while the cup of the centrifugal clutch rotates at a speed proportional to the vehicle speed. This causes an important slipping in the clutch until the vehicle attains a speed were the cup of the centrifugal clutch attains the same rotational speed of the engine. At this vehicle speed the centrifugal clutch locks and the slipping of the clutch becomes minimal and then the vehicle accelerates strongly until it attains 28.9 km/h which corresponds to 3430 rpm engine speed, for the transmission ratio used in the vehicle. In the first phase of the first connection of the engine the specific fuel consumption of the set engine-clutch is very high due mainly to the clutch slipping. In the second phase of the first engine connection the specific fuel consumption of the set engine-clutch decreases significantly due to the minimal or absence of clutch slipping. Due to the architecture of the transmission system of the transmission, where a one-way clutch exists in the rear wheel, a normal engine start is composed of two phases. In the first phase, after the engine is started by the electric starter, it accelerates its own inertia and the inertia of the transmission system from the rotation speed the electric starter puts the engine running until the engine rotation speed that corresponds to the vehicle speed when the engine is turned on, in this case 2900 rpm. At this speed the one-way clutch in the rear wheel locks and the second phase starts where the engine must accelerate all inertia of the vehicle from 24.4 to 38.0 km/h, which corresponds to an acceleration of the engine from 2900 to 4520 rpm, for the transmission ratio used in the vehicle.

The mass of fuel consumed in a trial is given by Eq. (3) where E_1 is the mechanical energy produced at the output shaft of the centrifugal clutch of the engine in the first engine connection when it drives the vehicle from rest to 30 km/h and bsfc_1 is the engine brake specific fuel consumption for the same engine connection. $E_{2,1}$ is the mechanical energy produced at the output shaft of the centrifugal clutch of the engine in the first phase of a normal engine connection when it drives the transmission system of the vehicle with the output shaft of the centrifugal clutch of the engine accelerating from 0 to 2900 rpm, speed at which the transmission system pulley mounted in the rear wheel of the vehicle attains the same speed of the rear wheel of the vehicle, the one-way clutch in the rear wheel locks and the engine starts to accelerate the all vehicle and $\text{bsfc}_{2,1}$ is the engine brake specific fuel consumption for the first phase of a normal engine connection. $E_{2,2}$ is the mechanical energy produced at the output shaft of the centrifugal clutch of the engine in the second phase of a normal engine connection when it drives the all vehicle from 24.4 km/h to 38.0 km/h and the speed of the output shaft of the centrifugal clutch changes from 2900 rpm to 4520 rpm while the one-way clutch in the rear wheel is locked and $\text{bsfc}_{2,2}$ is the engine brake specific fuel consumption for the second phase of a normal engine connection.

$$m_{f,trial}[\text{kg}] = \frac{E_1[\text{J}]\,\text{bsfc}_1[\text{g/kW}\cdot\text{h}]}{3.6\text{E}9} + \sum_{j=1}^{23} \frac{E_{2,1}[\text{J}]\,\text{bsfc}_{2,1}[\text{g/kW}\cdot\text{h}]}{3.6\text{E}9} + \frac{E_{2,2}[\text{J}]\,\text{bsfc}_{2,2}[\text{g/kW}\cdot\text{h}]}{3.6\text{E}9} \tag{3}$$

Eq. (2) indicates that a minimum mass of fuel consumed in a trial results in a maximum vehicle performance. Eq. (3) shows that to obtain a minimum mass of fuel consumed in a trial the sum of the products of the energy produced at the output shaft of the centrifugal clutch of the engine by the engine brake specific fuel consumption in each engine connection and phase must be minimized. The object of the present study of optimization is to find the engine operating parameters that minimise this sum for given values of E_1, $E_{2,1}$, and $E_{2,2}$ which result from the characteristics of the vehicle, topology of the circuit, driving strategy and the atmospheric conditions during the competition. Table 1 presents the energies produced by the engine in the different engine connections and phases when driving the Eco Veículo XC20i in the EuroSpeedway Lausitz, during the Shell Eco-marathon Europe 2010 event, for ambient temperature and track temperature equal to 15 ºC.

Table 1: Energies produced at the output shaft of the engine centrifugal clutch in different engine connections and phases when driving the Eco Veículo XC20i in the EuroSpeedway Lausitz, during the Shell Eco-marathon Europe 2010 event, for ambient temperature and track temperature equal to 15 ºC.

E_1 /J	$E_{2,1}$ /J	$E_{2,2}$ /J
3178	24.31	3251

3. **Engine Characteristics**

Table 2 summarizes the main engine characteristics. This engine was designed in-house. The engine is a one cylinder with 31.65 cm^3 displacement volume, is a four-stroke spark ignition engine and operates according to the Miller-Atkinson cycle. It features a hemispheric combustion chamber with one intake valve and one exhaust valve, double overhead cam (DOHC) and two spark plugs. It features electronic fuel injection in the intake manifold and electronic ignition. The programmable electronic control unit for fuel injection and ignition is a Haltech E6K.

Table 2: Main engine characteristics

Bore /mm	33.0
Stroke /mm	37.0
Number of cylinders	1
Displacement volume /cm^3	31.65
Compression ratio	15.0:1
Combustion chamber	Hemispheric
Number of intake valves	1
Number of exhaust valves	1
Intake valve head diameter /mm	12.32
Exhaust valve head diameter /mm	12.14
Intake valve seat width /mm	0.83
Exhaust valve seat width /mm	0.65
Intake valve seat angle /deg	45
Exhaust valve seat angle /deg	45
Intake valve maximum lift /mm	3.35
Exhaust valve maximum lift /mm	2.95
IVO /ºBTDC	10
IVC /ºABDC	75
EVO /ºBBDC	44
EVC /ºATDC	0
Number of spark plugs	2
Spark plug	NGK CR8HIX
Lubricating system	Splash lubrication
Lubricating oil	Shell Helix Ultra Extra 5W30, ACEA C2, C3 (A3, B4)
Lubricating oil volume	7.5 mL above level of lower point of crankshaft

To optimize the fuel efficiency of the engine in transitory mode it was necessary to apply to the engine a load with a moment of inertia equivalent to the inertia of the vehicle and with an aerodynamic torque equivalent to the aerodynamic drag force applied on the vehicle, in order to have a transitory behaviour mainly of engine rotational speed and engine temperatures similar to the one felt by the engine when driving the vehicle in the Shell Eco -marathon Europe 2010 competition. Table 3 summarizes the important characteristics of the vehicle necessary to calculate this moment of inertia equivalent to the inertia of the vehicle and this aerodynamic torque equivalent to the aerodynamic drag force of the vehicle.

Table 3: Characteristics of the vehicle driven by the engine

Vehicle mass, m_v /kg	88.0
Diameter of the wheel, D_w /m	0.474
Moment of inertia of left front wheel, I_{w1} /kg.m^2	6.10E-2
Moment of inertia of right front wheel, I_{w2} /kg.m^2	6.10E-2
Moment of inertia of rear wheel, I_{w3} /kg.m^2	6.08E-2
Moment of inertia of driven wheel of second stage, I_4 /kg.m^2	4.835E-4
Moment of inertia of the intermediate axle, I_{ia} /kg.m^2	6.516E-4
Mass of transmission belt of second stage, $m_{b,34}$ /kg	55.1E-3
Radius of driver wheel of second stage, r_3 /m	23.873E-3
Transmission ratio of the second stage, i_{34}	2.400
Moment of inertia of driver wheel of 1st stage and its fixation, I_1 /kg.m^2	4.683E-4
Mass of transmission belt of first stage, $m_{b,12}$ /kg	14.4E-3
Radius of driver wheel of the first stage, r_1 /m	16.711E-3
Transmission ratio, i_{tr}	10.629
Transmission system efficiency, η_{tr}	0.97
Vehicle drag coefficient, C_d	0.118
Vehicle frontal area, A_f /m^2	0.260
Air density, ρar /(kg/m^3)	1.225

The moment of inertia of the engine load mounted at the output shaft of the engine equivalent to the inertia of the vehicle is given by Eq. (4). For the characteristics of the vehicle given in Table 3 I_{eq} = 4.730E-2 kg.m^2. Ideally this should be equal to the sum of I_w and I_{wf} in Eq. (8).

$$I_{eq} = \frac{\left(m_v \left(\frac{D_w}{2} \right)^2 + (I_{w1} + I_{w2} + I_{w3}) + (I_{ia} + m_{b,34} r_3^2) i_{34}^2 + (I_1 + m_{b,12} r_1^2) i_{tr}^2 + I_4 \right)}{i_{tr}^2 \eta_{tr}} \quad (4)$$

The constant of proportionality between the aerodynamic torque of the load mounted on the engine output shaft and the square angular speed of the engine output shaft which generates an aerodynamic torque equivalent to the aerodynamic drag force applied on the vehicle is given by Eq. (5). For the characteristics of the vehicle given in Table 3 K_{eq} = 2.148E-7 N.m.s^2/rad^2.

$$K_{eq} = \frac{C_d A_f \frac{1}{2} \rho_{ar} \left(\frac{D_w}{2} \right)^3}{i_{tr}^3 \eta_{tr}} \quad (5)$$

4. Experimental Setup

The experimental setup used consists in the engine with a centrifugal clutch mounted in its crankshaft. The power output shaft of the engine is the output shaft of this centrifugal clutch. A wheel with a known moment of inertia is mounted in the output shaft of the centrifugal clutch. In the output shaft of the centrifugal clutch is mounted a magnet, a Hall Effect sensor is mounted in the proximity of the magnet to detect the passage of the magnet. This Hall Effect sensor is connected to a data acquisition system to measure and record the evolution of the rotational speed of the output shaft of the centrifugal clutch with time. Unless otherwise stated in this paper when we refer to the engine speed we refer to the rotational speed of the output shaft of the centrifugal clutch. A pipette is connected to the input of the fuel injector of the engine to measure the volume of fuel consumed in each test. This pipette is an ISO class A pipette with a capacity of 1.0 mL and a resolution of 0.01 mL. A thermocouple type K is mounted on the engine crankcase to measure its temperature. This thermocouple is a washer type and the internal diameter of the washer is 6 mm. This temperature is considered equal to the temperature of the lubricating oil in the crankcase because the crankcase is in aluminium alloy and is thermally insulated in the exterior wall by a cover of rigid polyurethane foam with 12 mm thickness. A thermocouple type K is mounted on the engine cylinder head to measure its temperature. This thermocouple has a MgO insulated junction inside a AISI 310 stainless steel shield with 1.5 mm external diameter.

The torque developed by the engine in the sum of two components and is given by Eq. (6): the aerodynamic torque of the inertia wheel, inertia wheel fixation to the centrifugal clutch cup and centrifugal clutch cup, T_{aero}; the torque to accelerate the inertia wheel, inertia wheel fixation to the centrifugal clutch cup and centrifugal clutch cup, $T_{inertia}$.

$$T = T_{aero} + T_{inertia} \quad (6)$$

The aerodynamic torque of the inertia wheel, inertia wheel fixation to the centrifugal clutch cup and centrifugal clutch cup is given by Eq. (7) where ω is the angular speed of the inertia wheel in rad/s and is given by Eq. (8) where n is the angular speed of the inertia wheel in revolutions per minute, rpm. Constants k_1 and k_2 were determined experimentally for an air density of 1.200 kg/m^3, performing deceleration tests of the inertia wheel, inertia wheel fixation to the centrifugal clutch cup and centrifugal clutch cup that were previously put rotating decoupled of the engine crankshaft at a rotation speed of approximately 2700 rpm, using an electric motor. The values of constants k_1 and k_2 determined experimentally for the two inertia wheels and its fixations, used in this work, are presented in Table 4.

$$T_{aero} = \frac{\rho_a}{1.200} \left(k_2 \omega^2 + k_1 \omega \right) \quad (7)$$

$$\omega = 2\pi \frac{n}{60} \quad (8)$$

The torque to accelerate the inertia wheel, inertia wheel fixation to the centrifugal clutch cup and centrifugal clutch cup is given by Eq. (9) where γ is the angular acceleration of the inertia wheel, I_w is the moment of inertia of the inertia wheel and I_{wf} is the sum of the moment of inertia of the inertia wheel fixation to the centrifugal clutch and of the moment of inertia of the centrifugal clutch cup.

$$T_{inertia} = (I_w + I_{wf})\gamma \tag{9}$$

The angular acceleration of the inertia wheel γ in (rad/s^2) is given by Eq. (10) where ω is the angular speed of the inertia wheel in rad/s and t is the time in seconds.

$$\gamma = \frac{d\omega}{dt} \tag{10}$$

The mechanical energy produced by the engine during the test at the output shaft of the centrifugal clutch, E, in J, is given by Eq. (11) where T is the torque available at the output shaft of the centrifugal clutch in N.m and θ is the rotation angle of the output shaft of the centrifugal clutch in rad .

$$E = \int_0^t T d\theta \tag{11}$$

$d\theta$ is given by Eq. (12).

$$d\theta = \omega\, dt \tag{12}$$

The engine brake specific fuel consumption, bsfc, in g/kW·h, in a test is given by Eq. (13) where ρ_f is the fuel density in kg/m^3 and V_f is the volume of fuel consumed in the test in mL.

$$\text{bsfc} = \frac{3600\rho_f V_f}{E} \tag{13}$$

The fuel used in the tests was unleaded gasoline 95 I.O. R.M. The density of the fuel was measured experimentally and its value was corrected for T_0 = 15 ºC. The value obtained was 740.62±1.86 kg/m^3 for a confidence level of 95 %. The density of the fuel in kg/m^3 at any temperature of the fuel, T_f, in ºC, is given by Eq. (14) taken from [2].

$$\rho_f(T_f) = \frac{\rho_f(T_0)}{\exp(\alpha(T_f - T_0))} \tag{14}$$

The value of α in Eq.(14) is given by Eq. (15) taken from [2] and the value of α_T is given by Eq. (16) taken from [1] where for gasoline K_0 = 346.4228 and K_1 = 0.4388.

$$\alpha = \alpha_T + 0.8\alpha_T{}^2(T_f - T_0) \tag{15}$$

$$\alpha_T = \frac{K_0 + K_1\rho_f(T_0)}{(\rho_f(T_0))^2} \tag{16}$$

In decelerations tests that occur after the engine is turned off at the end of a normal acceleration test the engine stays connected to the inertia wheel because the centrifugal clutch is locked until the engine speed is below the engine rotational speed where the clutch decouples which is close to 1350 rpm with the clutch used. This allows the friction torque of the engine to be measured as a function of engine speed from the engine speed the engine is turned off to the engine speed where the clutch decouples. The average friction torque in a deceleration test between engine rotation speeds n_2 and n_1 is given by Eq. (17).

$$\overline{T}_f = \frac{\int_{n_1}^{n_2} T d\theta}{\int_{n_1}^{n_2} d\theta} \tag{17}$$

Table 4: Moment of inertia and aerodynamic characteristics of the inertia wheels and inertia wheels fixation to the engine

Inertia wheel and its fixation	I_w /kg.m^2	I_{wf} /kg.m^2	k_1 /(N.m.s/rad)	k_2 /(N.m.s^2/rad^2)
Wheel with inertia equivalent to the complete vehicle	4.829E-2	4.967E-4	4.586E-5	1.520E-6
Wheel with inertia equivalent to the transmission system	2.129E-5	4.907E-4	3.350E-5	0

The brake fuel conversion efficiency of the engine, $\eta_{f,b}$, in a test is given by Eq. (18) taken from [3] where bsfc is the engine brake specific fuel consumption in the test as given by Eq. (13), in g/kW·h, and Q_{HV} is the lower heating value of the fuel at constant pressure in MJ/kg. Since Q_{HV} is constant for a given fuel the engine brake fuel conversion efficiency is inversely proportional to the engine brake specific fuel consumption.

$$\eta_{f,b} = \frac{3600}{\mathrm{bsfc}[\mathrm{g/kW \cdot h}]\, Q_{HV}[\mathrm{MJ/kg}]} \tag{18}$$

5. Experimental Procedure

To obtain the optimum performance of the vehicle two configurations were used for testing. In the first configuration a wheel was mounted in the engine output shaft with an inertia equivalent to the inertia of the transmission system. In the second configuration a wheel was mounted in the engine output shaft with an inertia equivalent to the inertia of the all vehicle. With the first configuration engine tests were run from 0 to 3000 rpm with different engine settings to simulate the first phase of a normal engine start. With the second configuration where run tests from 0 to 3500 rpm to simulate the first start of the engine in the competition when the vehicle starts from rest. With this configuration were also run tests from 2410 rpm to 4400 rpm and tests from 2410 to 3000 rpm to simulate the second phase of a normal engine start. In these tests the simulation of the second phase of a normal engine start is obtained by subtracting the fuel consumption and the energy production of the test from 2410 rpm to 3000 rpm to that of the tests from 2410 rpm and 4400 rpm.

6. Experimental Results

6.1. Effect of cylinder head temperature

Fig. 1 shows the evolution of the engine brake specific fuel consumption in a transitory start from rest to 3000 rpm with the initial cylinder head temperature (CHTi), when the engine accelerates a rotating mass with a moment of inertia equivalent to the moment of inertia of the transmission system. It can be seen that the engine brake specific fuel consumption is reduced more than two times from 4955 g/kW·h at a CHTi of 60 ºC to a 2099 g/kW·h at a CHT of 105 ºC. The optimum CHTi for this parameter is 105 ºC. Fig. 1 shows that operating with a initial cylinder head temperature between 100 and 105 ºC allows low brake specific fuel consumption in this phase and that it is of utmost importance to operate between 90 and 110 ºC.

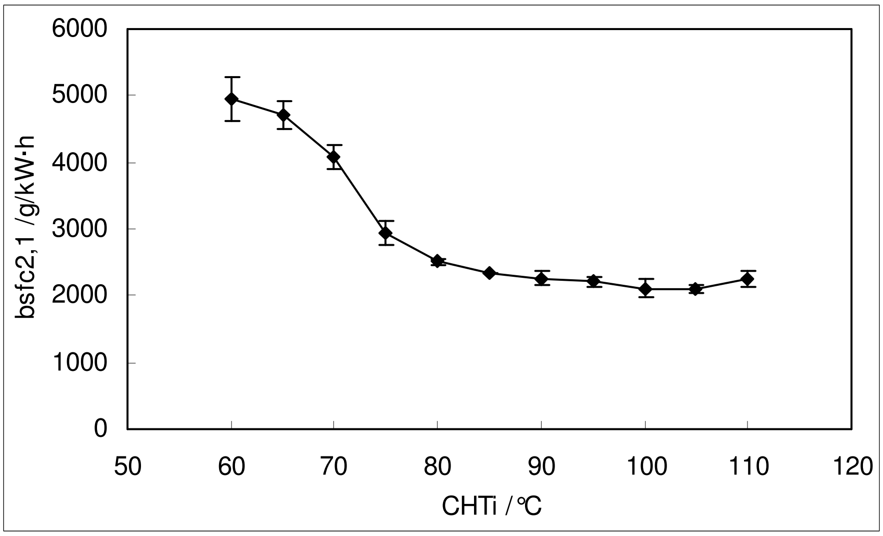

Figure 1: Evolution of the engine brake specific fuel consumption in a transitory start from rest to 3000 rpm with the initial cylinder head temperature (CHTi), when the engine accelerates a rotating mass with a moment of inertia equivalent to the moment of inertia of the transmission system.

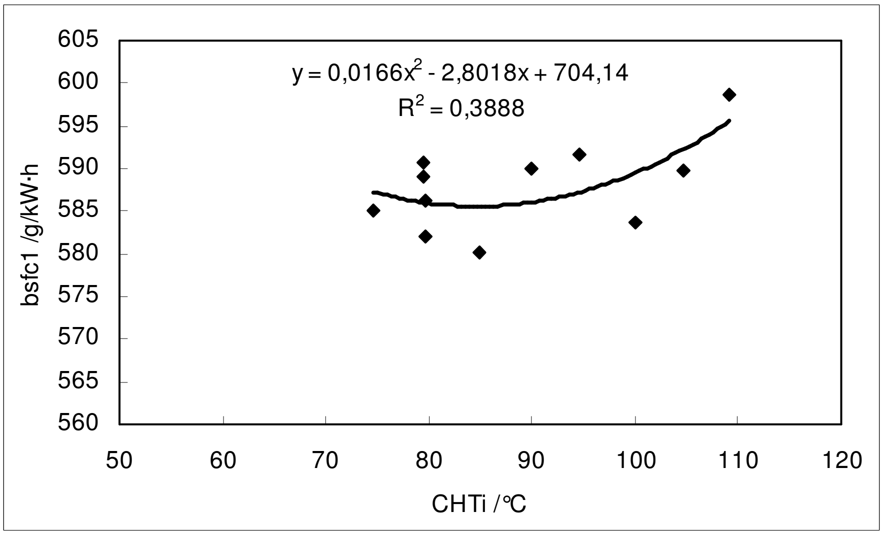

Figure 2: Evolution of the engine brake specific fuel consumption in a transitory start from rest to 3500 rpm with the initial cylinder head temperature (CHTi), with the engine accelerating a rotating mass with a moment of inertia equivalent to the moment of inertia of the complete vehicle. Lubricant oil temperature (OT) is in the range 74-80 ºC.

Fig. 2 shows the evolution of the engine brake specific fuel consumption in a transitory start from rest to 3500 rpm with the initial cylinder head temperature (CHTi), with the engine accelerating a rotating mass with a moment of inertia equivalent to the moment of inertia of the complete vehicle. This figure indicates that the CHTi has a small effect in the engine brake specific fuel consumption in this mode of operation and that the optimum CHTi is 85 ºC.

6.2. Effect of lubricating oil temperature

Fig. 3 shows the evolution of engine brake specific fuel consumption in transitory accelerations from 3000 rpm to 4400 rpm, with the engine accelerating a rotating mass with a moment of inertia equivalent to the moment of inertia of the total vehicle, with lubricant oil temperature (OT) when CHTi is in the range 105-107 ºC. This was obtained subtracting the energy produced by the output shaft of the centrifugal clutch of the engine and the volume of fuel obtained in transitory starts from 2410 to 4400 rpm and transitory starts from 2410 to 3000 rpm and applying Eq. (13). It can be seen that engine brake specific fuel consumption is reduced from an OT of 69 ºC to an OT of 87 ºC. Fig. 3 indicates that the optimum OT is larger than 87 ºC.

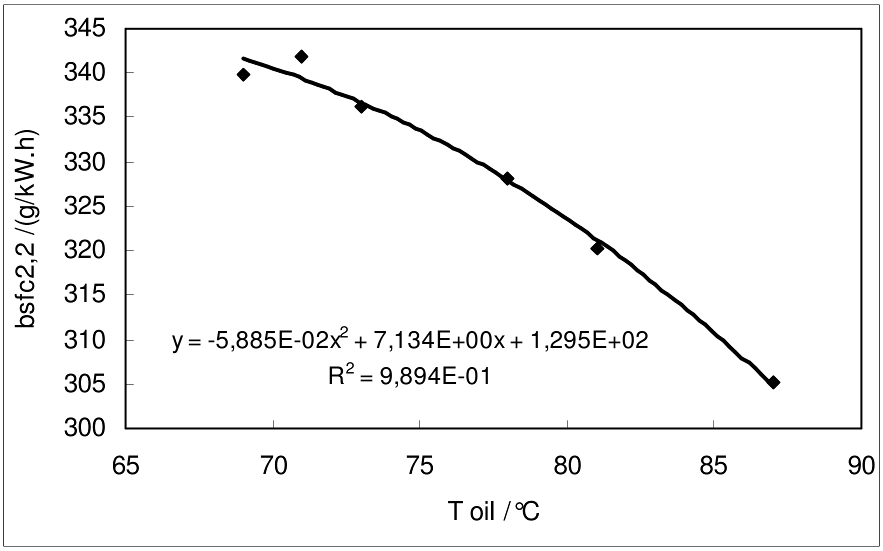

Figure 3: Evolution of the engine brake specific fuel consumption in a transitory acceleration from 3000 rpm to 4400 rpm, with the engine accelerating a rotating mass with a moment of inertia equivalent to the mass of the total vehicle, with the lubricant oil temperature. CHTi is in the range 105-107 ºC.

Fig. 4 shows the evolution of the average friction torque with the lubricating oil temperature in deceleration of the engine from 4400 rpm to 1500 rpm. Fig. 4 indicates that the optimum OT is above 87 ºC. Fig. 4 also shows that the reduction of the average friction moment when the OT increases between 69 ºC and 87 ºC is less that 0.01 N·m. Given the fact that the average torque of the engine is 1.60 N·m this indicates that there is not too much to gain with respect to average engine friction torque to operate with the OT above 80 ºC.

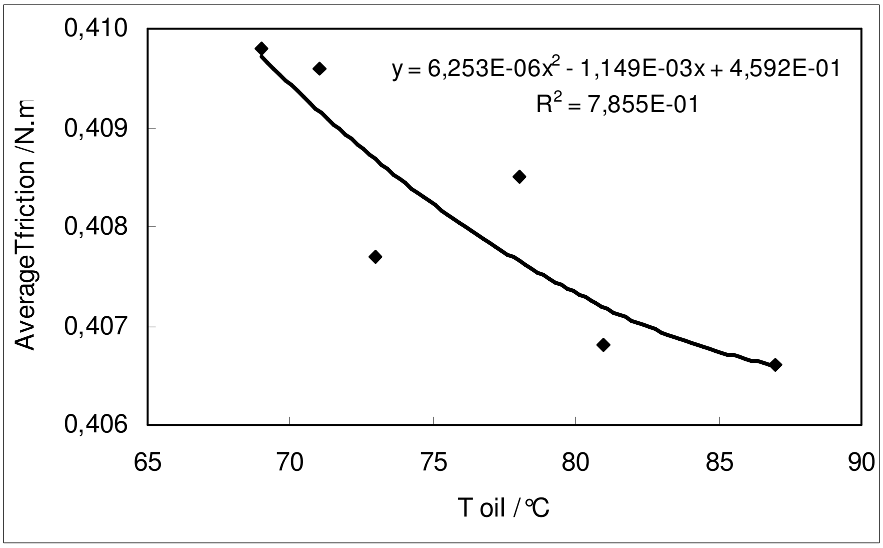

Figure 4: Evolution of the average friction torque with the lubricating oil temperature in deceleration of the engine from 4400 rpm to 1500 rpm.

Fig. 5 shows the evolution of engine brake specific fuel consumption in transitory accelerations from 3000 rpm to 4400 rpm, with the engine accelerating a rotating mass with a moment of inertia equivalent to the moment of inertia of the total vehicle, with the initial cylinder head temperature when OT is in the range 72-76 ºC. This was obtained subtracting the energy produced by the output shaft of the centrifugal clutch of the engine and the volume of fuel obtained in transitory starts from 2410 to 4400 rpm and transitory starts from 2410 to 3000 rpm and applying Eq. (13). It can be seen that engine brake specific fuel consumption is reduced from a CHTi of 118 ºC to an OT of 80 ºC. Fig. 5 indicates that the optimum CHTi is lower than 80 ºC.

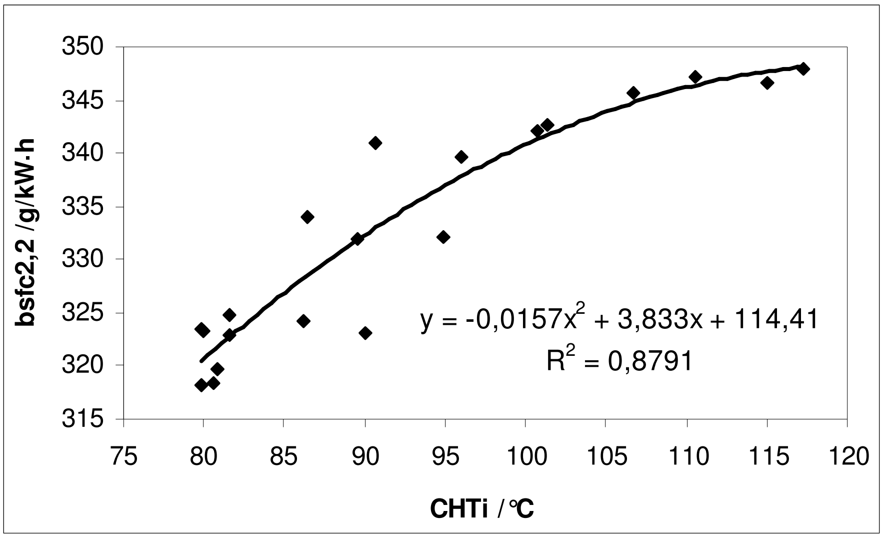

Figure 5: Evolution of the engine brake specific fuel consumption in a transitory acceleration from 3000 rpm to 4400 rpm, with the engine accelerating a rotating mass with a moment of inertia equivalent to the mass of the total vehicle, with the initial cylinder head temperature. OT is in the range 72-76 ºC.

6.3. Effect of ignition timing and of the duration of fuel injection

At each engine rotational speed the ignition timing and the have a strong influence on the engine brake specific fuel consumption. Fig. 6 shows the evolution of the engine brake specific fuel consumption with the ignition timing and with the duration of fuel injection settings at 4500 rpm when the engine accelerates a rotating mass with a moment of inertia equivalent to the total mass of the vehicle from 2410 to 4400 rpm. It can be seen from the observation of Fig. 6 that the optimum settings are TI = 11 deg BTDC and 10.512 ms duration of fuel injection. The *y*-error bars represent plus and minus one standard deviation of the three measurements made for each point. A similar behaviour was observed through experimental measurements for the evolution of the engine brake specific fuel consumption with the ignition timing and with the duration of fuel injection settings at other engine speeds when the engine accelerates a rotating mass with a moment of inertia equivalent to the total mass of the vehicle from 2410 to 4400 rpm. The optimum settings for the ignition timing and the duration of the fuel injection were retained for 2500, 3000, 3500, 4000 and 4500 rpm and are presented in Table 5.

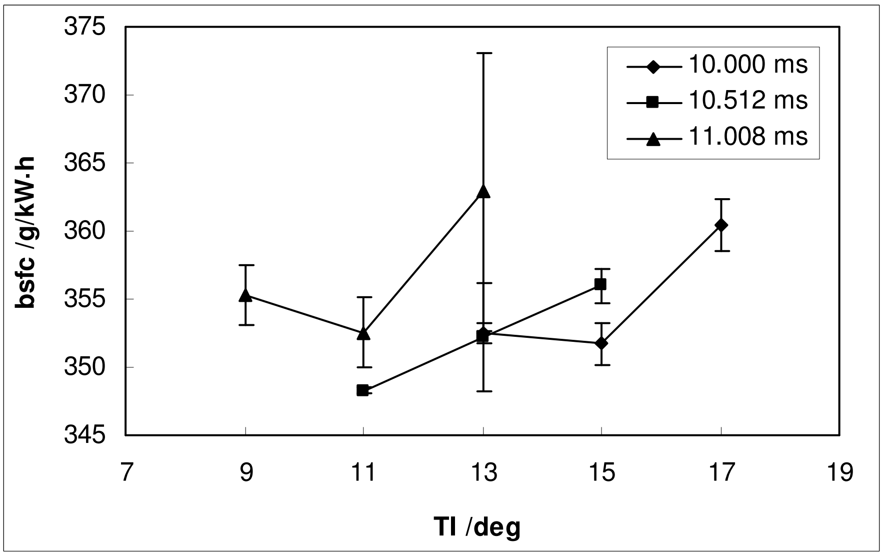

Figure 6: Evolution of the engine brake specific fuel consumption with the ignition timing and with the duration of fuel injection settings at 4500 rpm when the engine accelerates a rotating mass with a moment of inertia equivalent to the total mass of the vehicle from 2410 to 4400 rpm.

Table 5: Optimum settings for the ignition timing and the duration of the fuel injection obtained from experimental measurements.

MAP/kPa	95-98	98-101	95-98	98-101	95-98	98-101	95-98	98-101	95-98	98-101
n /rpm	2500	2500	3000	3000	3500	3500	4000	4000	4500	4500
tinj /ms	8.496	10.000	8.496	10.512	8.496	11.008	8.496	10.000	8.496	10.512
TI /deg	9	9	10	10	10	10	11	11	11	11

7. Discussion of Results

Table 6 presents the estimated performance of the Eco Veículo XC20i in the Shell Eco-marathon Europe 2010 and the real performance. The optimum CHTi seems to be below 80 ºC. Fig. 3 shows that the optimum oil temperature is above 87 ºC. It is almost impossible to operate during all the duration of the competition with a temperature in the cylinder head below the temperature of the lubricating oil in the carter because the heat is generated in the cylinder head and most of the heat losses occur in the carter. At present time a difference of about 25 ºC exists between the temperature of the cylinder head and the temperature of the carter for the engine operating in the conditions of the competition, due to the design of the engine. This study shows that the design of the engine should be changed to promote the heat transfer from the cylinder head to the carter in order to get these two temperatures as close as possible. The difference between the estimated and the real performance for the conditions most probably found in the competition is 176.6 km/L, which is a difference of 8.01 % relative to the real performance. This can be attributed to several factors including: the average speed in the real competition being larger than the minimum average speed allowed, used to estimate the real performance; the existence of wind in the real competition; the track surface presenting some degree of degradation

Table 6: Estimated performance of the Eco Veículo XC20i in the Shell Eco-marathon Europe 2010 for an OT = 75 °C and several CHTi at 30.00 km/h average speed and real performance at 30.60 km/h average speed.

	Estimated	Estimated	Estimated	Estimated	Estimated	Estimated	Real
CHTi /°C	80	85	90	95	100	105	~100
OT /°C	75	75	75	75	75	75	~75
E_1 /J	3178	3178	3178	3178	3178	3178	
$E_{2,1}$ /J	24.31	24.31	24.31	24.31	24.31	24.31	
$E_{2,2}$ /J	3251	3251	3251	3251	3251	3251	
bsfc1 /g/kW·h	586.2	585.9	586.4	587.8	590.0	593.0	
bsfc2,1 /g/kW·h	2514	2332	2259	2208	2113	2099	
bsfc2,2 /g/kW·h	320.6	326.8	332.2	336.9	340.7	343.8	
Performance /km/L	2495	2462	2430	2402	2381	2362	2204.4

8. Conclusions

A method was developed to optimize the fuel efficiency of the M3165 internal combustion engine in transitory operation similar to the one encountered when driving the Eco Veículo XC20i in the European Shell Eco-marathon Europe 2010. This method was used and allowed to identify and optimize several operating variables of the engine. These variables are the injection duration, ignition timing, cylinder head temperature and oil temperature. The optimum initial cylinder head temperature at each engine start is less than 80 °C. The optimum lubricating oil temperature is larger than 87 °C. The estimated performance of the vehicle is close to the real performance but optimistic by 8.01 %.

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