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# Simple method of estimating consumption of internal combustion engine for hybrid application

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**Abstract**—This article presents two simple methods to evaluate the fuel consumption of internal combustion engines. Furthermore, it points out the link between piston volume, engine size and fuel consumption as it gives the possibility to visualize the chosen working points inside an internal combustion engine operation points map. This leads to the fact that this approach is a very helpful tool for the design of hybrid systems including internal combustion engines. The two presented methods, one very simple, one more elaborated, are interesting as first approach of internal combustion engine sizing and to evaluate the consumption of internal combustion engine. The results for six pure internal combustion engines on two different driving cycles have been compared to literature data showing extremely satisfying accuracy. An application on a series hybrid vehicle underlines the potential of fuel savings due to hybrid architecture.

**Index Terms**—Internal combustion engine, fuel consumption, hybrid vehicle, modeling

## I. INTRODUCTION

The limitation of resources and the climate change impose a change in the mode of personal transport. One promising solution is the use of a combination of different energy sources - so called hybrid systems [1]–[3] - in order to increase the overall system efficiency. Internal combustion engines (ICE) are highly developed systems that have been the foundation of personal transport throughout the last century [4]. Even though the electrification has increased inside the automotive industry [5], the use of internal combustion engines will continue. Therefore, it is interesting to be able to dispose of a simple approach to predesign an ICE and to estimate the fuel consumption of this engine. This works for the different operating points used as well as entire driving cycles. There is a considerable number of highly detailed and accurate models of internal combustion engine available [6]–[10]. But for a first system evaluation, a simple and easy to use method seems more appropriate.

In most previous works, authors utilize the specific fuel consumption with variable engine speed [11]–[14] to estimate fuel consumption and consequently adopted the envelope of the minimum line as operating points of engine of such system. This choice is also due to the fact that the engine is the prime mover and the on board energy storage system (ESS) gives supplementary energy while needed [11]–[15]. This work focuses on estimating the consumption on a plug-in hybrid series architecture. Because tendency is now growing to construct a vehicle equipped with a bigger capacity of

ESS (rechargeable on electrical outlet) [16]. This makes it possible to utilize the ICE as secondary energy source to extend the all electric autonomy range, whilst opening the possibility to impose engine operational points in region with high efficiency.

There are different approaches toward the control of series hybrid vehicles available. There are two methods of rule based strategies: the first method is based on fuzzy rules and the second method is based on deterministic rules. In article [17], [18], fuzzy logic rules of four input parameters are applied to control torque and engine speed within its high efficiency region. The fuzzy logic rules are robust and suitable to control a nonlinear time-variant system [16], but it is case sensitive and not easy to implement [11]. On the other hand, deterministic rules are easier to implement using lookup table [11]–[14], [16], [19]. But, a strategy of only two parameters will result in a higher number of ICE startups, thus increasing fuel consumption [12], [13]. Further possible consequences are important fluctuations of torque and engine speed during hybrid driving [11], [14], [19]. The second large class of control strategies for hybrid vehicles is based on optimization control strategies to find the most adapted ICE operation points [20]. However, the strategy is optimized for a particular driving cycle and cannot easily be used for a real-time application. Moreover, a dynamic programming demand represents a significant burden on calculation, thus require an important memory to run the algorithm permanently.

In this paper, a three variables deterministic rule of boolean function is introduced to decide on the use of the engine; it is a heuristic method and utilizes the minimum and maximum thresholds of the battery state of charge (SOC) and a small memory of previous ICE operation to determine its decision. No information on future or past route profile is needed as in [12], [13]. It only needs the present vehicle power demand which can be easily determined.

In the next section, the methods of evaluating the fuel consumption are introduced and validated against literature values for six ICE vehicles using two different driving cycles. A consumption map for one representative engine is developed thereafter highlighting the working points of ICE vehicles for two different driving cycles as well as working points for a series hybrid system. The same model will be used to estimate the most adapted ICE for different applications. The article

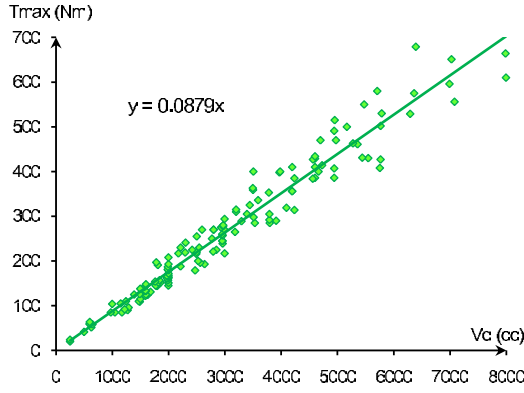


Figure 1. Maximum torque of gasoline engines

closes with conclusions and perspectives.

## II. MODELING OF EFFICIENCY AND CONSUMPTION OF INTERNAL COMBUSTION ENGINE

In this section, the methods to estimate fuel consumption are introduced.

### A. General approach

For the first approach, the maximum torque line is defined with a simple quadratic equation 1.  $T_{max}$  is in Nm and  $N$  is in rpm. If the real maximum torque is not known, as a first step we can use the linear equation 2, obtained from statistics shown in figure 1, to estimate the maximal torque.  $V_d$  is in  $\text{cm}^3$  and the result is as presented in figure 2.

$$T = (-4.2 \cdot 10^{-6} N^2 + 0.032 N + 105) \frac{T_{max}}{165} \quad (1)$$

$$T_{max} = 0.0879 V_d \quad (2)$$

Fuel consumption for internal combustion engine can be evaluated from the calculation of the brake power,  $P_b$ , which is the usable power delivered by the engine. This brake power (equation 3) is the result of the difference between the gross indicated power,  $P_{ig}$ , and the power losses,  $P_f$ , due to friction losses from the pistons, bearings and other mechanical components.

$$P_b = P_{ig} - P_f \quad (3)$$

The gross indicated power is usually calculated based on the evolution of the in-cylinder pressure during a cycle. Since this type of calculation is based on the pressure, therefore the force applied on the piston during the cycle, it takes into account all the phenomenon occurring during the cycle (combustion, heat exchange with the wall, pumping losses,...). However, this type of measurement is hardly available over the full engine operating range, and some assumption have to be made in order to evaluate the power available. It can be calculated by the fuel indicated efficiency,  $\eta_{f,i}$ , which compares the gross indicated power with the power delivered by the combustion of the fuel. This power is calculated based on the mass flow

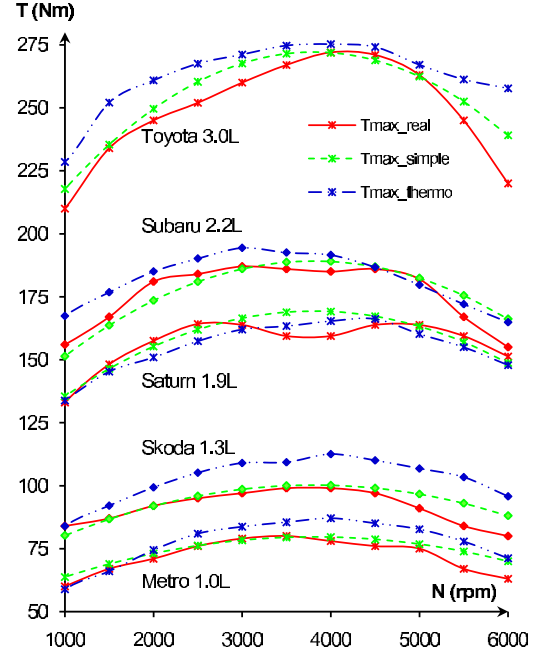


Figure 2. Maximum torque line of different engines

of the fuel,  $\dot{m}_f$  in  $\text{kg s}^{-1}$ , and its lower heating value, LHV in  $\text{J kg}^{-1}$  (Gasoline  $44 \text{ MJ kg}^{-1}$  and Diesel  $42.5 \text{ MJ kg}^{-1}$ ). A first evaluation of the fuel indicated efficiency considers a value of 40%. This evaluation considers energy losses through two main pathways: i) 30% of the available energy of the fuel is lost in the hot exhaust gases and ii) 30% is lost due to heat transfers to the wall, the remaining 40% is the considered fuel indicated efficiency. This value is in agreement with commonly measured fuel indicated efficiencies [6] but should be re-evaluated in order to take into account the type of engine (gasoline or Diesel, two or four strokes) and the engine operating conditions (idle, part load, full load). The fuel indicated efficiency assumes a perfect combustion process during which all the energy of the fuel is delivered. This might be real for low operating speed but the efficiency of the combustion process decreases with increasing operating speed. This is due to the reduced time for the combustion to occur and part of the fuel remain unburned at the end of the combustion process. This unburned gas comes from flame quenching due to heat transfer with the wall and quenching in the crevices around the piston head. A simple way to evaluate the combustion efficiency,  $\eta_c$ , is to consider an inverted evolution with the engine speed (equation 4).

$$\eta_c = \eta_{c0} - \frac{A}{B + N} \quad (4)$$

In this equation,  $\eta_{c0}$  is assumed to be 0.98 and the values of  $A$  and  $B$  are 300 and 2000 respectively.  $N$  is the engine speed in rpm. Therefore, the gross indicated power is calculated thanks to the equation 5.

$$P_{ig} = \eta_{f,i} \left( \eta_{c0} - \frac{A}{B + N} \right) \dot{m}_f \text{LHV} \quad (5)$$

To calculate the brake power,  $P_b$ , we have to estimate the friction power,  $P_f$ , or the friction mean effective pressure ( $FMEP$ ). The friction losses are the sum of different phenomena, such as piston friction, which can be independent of speed or proportional to speed. In the simplest case we consider it as a linear equation in function of engine velocity as in equation 6 [6]. The constant of friction  $f$  is assumed to be 100 kPa and the factor  $f_p$  is 20.  $N$  is the velocity of engine in rpm.

$$FMEP = f + f_p N \quad (6)$$

Then, the friction power is calculated from the friction mean effective pressure by multiplying FMEP by the cylinder volume,  $V_d$ , displaced per second (equation 7).

$$P_f = (f + f_p N) \frac{V_d N}{R_c 60} \quad (7)$$

$R_c$  is 1 for 2 stroke engine and 2 for 4 stroke engine. Thus, the fuel mass flow rate can be easily evaluated from the brake power requirements thanks to the resulting equation 8 as a function of the engine speed and its displacement volume.

$$\dot{m}_f = \frac{P_e + (f + f_p N) \frac{V_d N}{R_c 60}}{\eta_{f,i} (\eta_{c0} - \frac{A}{B+N}) LHV} \quad (8)$$

### B. Refined approach

The refined approach consists in a zero-dimensional single-zone thermodynamic model of the internal combustion engine. The aim of this model is to evaluate the influence of the engine speed and load on the fuel indicated efficiency.  $\eta_{f,i}$  is defined as the ratio between the work transfer to the piston thanks to the in-cylinder pressure and the theoretical energy available in the fuel (equation 9). Therefore, the evolution of the pressure inside the cylinder during the engine cycle is needed.

$$\eta_{f,i} = \frac{\int -P dV}{\dot{m}_{fuel} LHV} \quad (9)$$

The cylinder is considered as an open system where pressure,  $P$ , temperature,  $T$ , and mixture composition are homogeneous. The gas is supposed to be a constant mixture of fuel and air. The model is governed by the conservation of mass and the conservation of energy. The cylinder can only exchange mass through the intake and exhaust valves (equation 10).

$$\frac{dm}{dt} = \frac{dm_{int}}{dt} + \frac{dm_{exh}}{dt} \quad (10)$$

where  $m$  is the mass inside the cylinder and  $dm_{int}/dt$  (respectively  $dm_{exh}/dt$ ) is the mass flow rate through the intake (respectively exhaust) valve.

The first law of thermodynamics is applied to the open system which exchanges heat through the wall ( $Q_{wall}$ ) and receives heat thanks to a combustion model ( $Q_{comb}$ ). The work transfer through the piston due to the change of in-cylinder volume,  $V$ , is equal to  $-P dV/dt$  (equation 11).

$$\frac{dm u}{dt} = \frac{dW}{dt} + \frac{dQ}{dt} = -P \frac{dV}{dt} + \frac{dQ_{wall}}{dt} + \frac{dQ_{comb}}{dt} \quad (11)$$

where  $u$  is the internal energy.

Since, the unknowns are the pressure, the temperature and the mass inside the cylinder a third equation is needed which is the ideal gas law in its derived form (equation 12).

$$P \frac{dV}{dt} + V \frac{dP}{dt} = \frac{dm}{dt} r T + m \frac{dr}{dt} T + m r \frac{dT}{dt} \quad (12)$$

where  $r$  is the ideal gas constant ( $r = R/M$ ). Since the mixture composition is assumed to be constant  $dr/dt = 0$ .

Therefore, some models are needed in order to calculate the mass flow rate through the valve, the heat exchange with the wall and the heat release from the combustion process. The mass flow rate is calculated with the assumption of the quasi-steady adiabatic and compressible flow. The flow is generated due to the difference in pressure between the upstream ( $us$ ) and the downstream ( $ds$ ) if  $P_{us} \geq P_{ds}$ . Thus, mass flow rate depends on the upstream conditions (pressure,  $P_{us}$ , temperature,  $T_{us}$ ) and the downstream pressure,  $P_{ds}$  (equation 13).

$$\frac{dm}{dt} = AC_d P_{us} \sqrt{\frac{2\gamma}{(\gamma-1)rT_{us}} (X^{\frac{2}{\gamma}} - X^{\frac{\gamma+1}{\gamma}})} \quad (13)$$

where

$$X = \frac{P_{ds}}{P_{us}} \leq \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \quad (14)$$

The heat exchange with the wall is considered to be forced convection (equation 15) and the heat transfert coefficient is calculated thanks to the correlation defined by Han et al. [21].

$$\frac{dQ_{wall}}{dt} = hS(T - T_{wall}) \quad (15)$$

Finally, the heat release rate,  $dQ_{comb}/dt$ , is calculated thanks to the empirical burning law defined by Vibe [22] which is commonly used [6] in internal combustion engines thermodynamic model. This law (equation 16) considers an exponential evolution of the heat release from the combustion, starting at  $\theta_i$  crank angle degree before the top-dead-center with a combustion duration of  $\Delta\theta$ .

$$Q_{comb} = \dot{m}_{fuel} LHV (1 - e^{-a(\frac{\theta - \theta_i}{\Delta\theta})^{n+1}}) \quad (16)$$

### III. VALIDATION OF THE METHOD

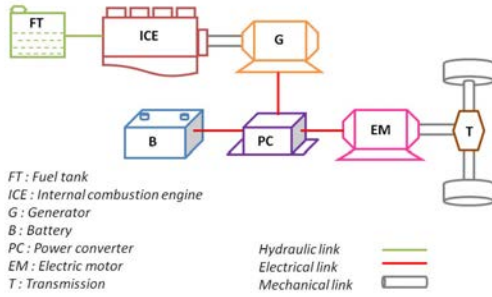
The presented approach is validated for different internal combustion engines of different sizes and with regard to different driving cycles. The results are presented in table I. It can be seen, that the simple approach gives a first idea of fuel consumption. The accuracy for extra urban driving as in EUDC is rather promising underestimating the fuel consumption by not more than 10%. For urban driving as in ECE R15 the accuracy is not so good with maximum underestimation of 25%. The thermodynamic approach largely improves the fuel consumption predictions for urban driving with maximum error as low as 3.7%, whilst showing an overestimation of fuel consumption of up to 13% for extra urban driving. The discrepancies for the urban cycles mainly come from the evaluation of the fuel indicated efficiency. We choose a constant value for all the operating conditions of the

Table I  
VALIDATION OF THE MODEL IN L/100km

		Toyota Camry 2.4	Alfa Romeo 159 2.2 JTS	BMW 320i	Renault Clio 1.6 16V (2009)	Honda Jazz 1.5 VTEC (2008)	Suzuki Alto 1.0 (2009)
ECE R15	Constructor	10.7	13	10.8	9.0	8.4	5.5
	Thermo	11.10	12.60	11.15	9.19	8.67	5.33
	Simple	8.34	10.89	9.39	7.36	6.93	4.12
	Error <sub>thm</sub> (%)	3.7	-3.1	3.2	2.1	3.2	-3.1
	Error <sub>sim</sub> (%)	-22.1	-16.2	-13.1	-18.2	-17.5	-25.1
EUDC	Constructor	7.1	7.3	6.2	6.1	5.5	3.8
	Thermo	7.86	7.97	6.91	6.71	6.12	4.28
	Simple	6.38	7.04	5.77	5.69	5.19	3.67
	Error <sub>thm</sub> (%)	10.7	9.2	11.5	10.0	11.3	12.6
	Error <sub>sim</sub> (%)	-10.1	-3.6	-6.9	-6.7	-5.6	-3.4



(a) Sketch



(b) Architecture

Figure 3. Presentation of the Series hybrid car Noao

engine whereas  $\eta_{f,i}$  should vary as a function of engine load and speed as it has been done in the thermodynamic model. The discrepancies of the thermodynamic model are explained by the fact that the presented approach is a 0-dimensional model, which does not take into account three dimensional phenomena such as boundary layer or acoustics inside the intake pipes [6].

#### IV. APPLICATION OF METHODS ON SERIES HYBRID VEHICLE

The presented approaches are applied to a series hybrid configuration using a 1000 cm<sup>3</sup> gasoline engine, with the architecture shown in figure 3(b). It is inspired by the series hybrid competition car called *Noao* (figure 3(a)), which has been produced in 2011 by different companies installed around the Magny-Cours race track in France. The vehicle has the parameters presented in table II.

A system control using heuristic method has been designed and simulated, leading to the operating points indicated in figure 4(a) and figure 4(b). This method uses the Boolean

Table II  
VEHICLE PARAMETERS

Mass kg	Front area m <sup>2</sup>	$C_x$ value [-]	rolling resistance [-]	wheel diameter m
1200	2	0.35	0.012	0.62

functions to determine the engine on-off command. The control considers three variables to make decision, the previous state of engine, the  $SOC_{min}$ , and  $SOC_{max}$ . Note that the engine will start whenever the battery state of charge attains its minimum threshold and will only stop if it reach the maximum threshold. In addition, a combination of a piecewise linear equation 17 in function of power demand during recharge can increase the power given by engine while needed.  $P_{trans}$  in equation 18 is the fraction of difference between  $P_{opt}$  and  $P_{Tmax}$ . This function is adjusted to make the engine work between its optimal point and its maximum torque point precisely via lookup table. It is possible to utilize this control strategy for a real-time optimization because it can yield minimum consumption for urban cycle and at the same time can deliver required power if needed for extra urban cycle.

$$P_{ICE} = \begin{cases} P_{opt} & IF & P_{dmd} < P_{opt} \\ P_{trans} & IF & P_{opt} \leq P_{dmd} \leq P_{Tmax} \\ P_{Tmax} & IF & P_{dmd} > P_{Tmax} \end{cases} \quad (17)$$

$$P_{trans} = \left( \frac{P_{dmd} - P_{opt}}{P_{Tmax} - P_{opt}} \right) P_{dmd} + P_{opt} \quad (18)$$

After that, the fuel consumption of the ICE and hybrid solution are evaluated (table III). To evaluate the overall efficiency, we use the equivalent consumption as indication of improvement of this hybrid system. This parameter includes the mass of fuel to recover the same amount of battery energy used during imposed driving cycles. We can find that the fuel consumption of the hybrid solution estimates lower equivalent fuel consumption for ECE R15 cycle, with an improvement of consumption around 30% compared to the solution using the architecture having only ICE. In EUDC cycle, the quantity of fuel has increased by 10%. This is due to the series hybrid architecture, the ICE power is not connected directly to the transmission, thus introducing supplementary losses.



Table III  
COMPARISON OF THE MODELS ON SERIES HYBRID ARCHITECTURE

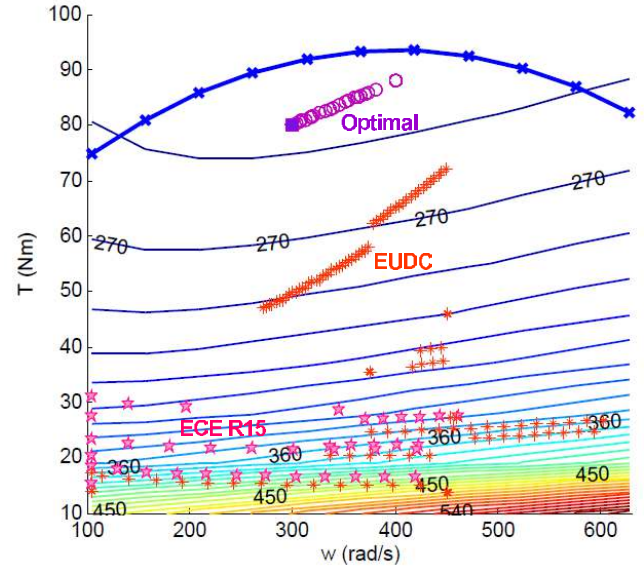
Cycle	Model	Engine only [L/100km]	Equivalent consumption Series hybrid [L/100km]	[%]
ECE R15	Thermo	7.47	4.93	-34
	Simple	6.25	4.54	-27
EUDC	Thermo	5.95	6.50	+9
	Simple	5.44	6.00	+10

The specific fuel consumption maps of the both methods, including working points of ICE and hybrid architecture for both driving cycles are presented in figure 4. It can be seen, that the difference of the two iso-consumption maps is most important in regions of low torque application. This is the same region, that is mainly used for the urban driving cycle, thus explaining the refinement in the results of urban driving using the thermodynamic model. Furthermore, it can be seen that the consumption around the working points for extra urban driving only changes slightly with higher consumption for the thermodynamic model, underlining the observations in table I. But the most interesting conclusion for the iso-consumption map is the fact that for the hybrid architecture, the working point are in the region of optimal consumption thus explaining the considerable gain in fuel consumption using a series hybrid architecture.

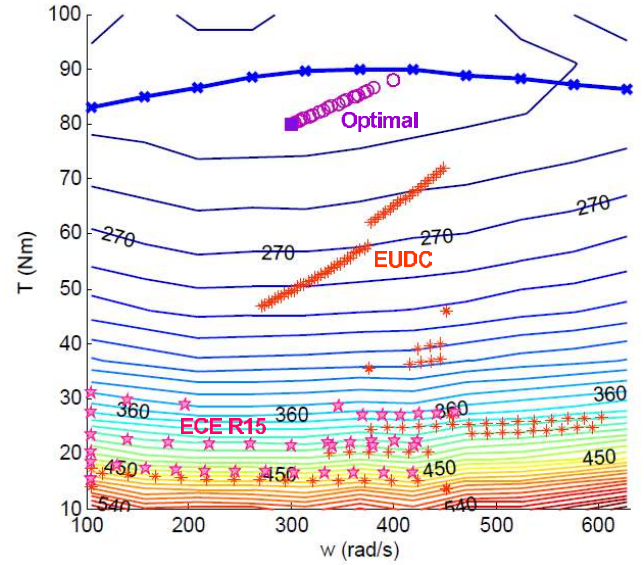
Investigations of a large number of ICE have shown that not only the maximum torque can be approximated using the displacement (figure 1), but also the maximum power is related to the engine displacement (figure 5(b)) and the engine weight in function of the maximum power (figure 5(b)). This knowledge enables the predimensioning of an internal combustion engine for hybrid applications. For hybrid applications the maximum engine power is known by system analysis. Based on the maximum displacement, maximum torque and corresponding engine speed can be evaluated. The engine weight can be deduced, providing additional information for the system. Non the less the presented methods provide a possibility to evaluate and visualize fuel consumption for different driving cycles.

## V. CONCLUSION AND PERSPECTIVE

This work presents two methods to estimate the fuel consumption of internal combustion engine. The simple approach, depending only on the cylinder volume, power requirements and rotation speed, can be used for very first estimations of operating points and consumption of an ICE. The more elaborated approach allows the refinement of the estimation using a one dimensional model. Both modeling approaches have been verified against literature data and show good accuracy for six different ICEs and two different driving cycles, with a maximum derivation of -25%. Thereafter, the approach is used to evaluate the ICE choice for an series hybrid vehicle, including considerations of maximum ICE power and parameters like displacement and engine weight. In the following the presented method will be used as tool to



(a) Simple model



(b) Thermo model

Figure 4. Comparison of operational points between a system with engine only and series hybrid architecture

design ICE for hybrid applications linking maximum power, displacement, engine weight and consumption depending on working points attained during different driving cycles.

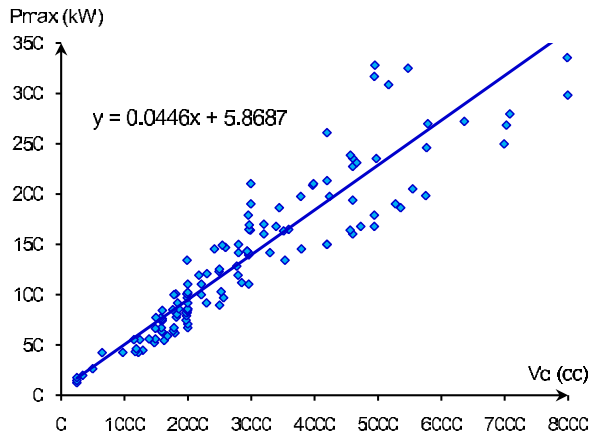
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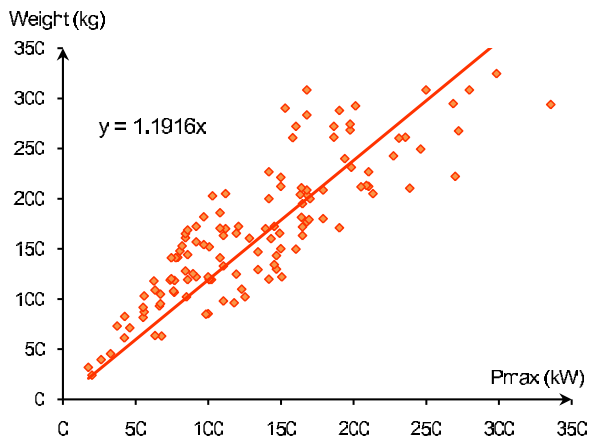
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(a) Maximum power in function of engine displacement



(b) Engine weight in function of maximum power

Figure 5. Approximated engine characteristics

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