CALCULATION OF FUEL CONSUMPTION AND ENGINE-OUT EMISSIONS IN ECOGEST

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Abstract A numerical model capable of predicting fuel consumption and tailpipe emissions (HC, CO, NO_x and CO_2), from the knowledge of the motor vehicle dynamics and engine-out fuel consumption, exhaust gas temperature and emissions maps, has been developed. This model was introduced in a program called EcoGest. If the fuel consumption, exhaust gas temperature and emissions maps are not available from the OEMs, EcoGest uses an engine model to obtain it. Then, an exhaust system model is used to correct the engine-out emissions and calculate the real tailpipe emissions. In the present paper the engine model is presented. The model, called MotorIST, simulates the indicated cycle of a four stroke spark ignition engine. It uses a zero-dimensional modelization. The results obtained are both global results and values discretized during the compression and expansion strokes. MotorIST runs easily in PCs because it is fast and is not memory demanding. It is useful as a tool for preliminary design of engines and their auxiliary equipment, and for research and teaching work.

Keywords Vehicle simulation, spark ignition engines, engine-out fuel consumption, engine-out emissions

INTRODUCTION

The majority of commercial programs available to estimate emission factors of a certain fleet, such as Mobile 6 [1] or Copert III [2], are average speed based, suitable only for macro-scale applications such as national estimates. For each average speed there is in the database of the programs a corresponding driving cycle having the same average speed. Emission estimates are those of that cycle. The database cycles and corresponding emissions are obtained from chassis dynamometer tests [3]. But different cycles corresponding to different vehicle dynamics and driving behavior can have the same average speed, thus resulting different fuel consumption and tailpipe emissions for the same average speed. To overcome this limitation, the use of instantaneous emission models to estimate emission quantities or define emission factors is rapidly growing (modal emission models) [3]. Those programs, such as MODEM [4], assign a mean emission level to every pair of instantaneous speed and acceleration values. But, again, these emission matrices are derived from hundred vehicles tested in chassis

dynamometers. For the construction of the matrices vehicles are subdivided in three categories according to engine displacement: small cars (less than 1.4 liters), middle class cars (between 1.4 and 2.0 liters) and upper class cars (great than 2 liters) [3]. So, individual vehicle can differ very much from the respective category it is included in. These kinds of modal emissions models are more suitable for micro-scale applications were the driving behavior is to be taken into account.

A micro-scale model introduced in a program named EcoGest was developed [5]. This program allows to study a single vehicle in a real world situation, considering all the vehicle characteristics from exterior chassis through engine to exhaust after-treatment characteristics. Road topography and occupation of the vehicle are other important inputs of the model. EcoGest can be used to simulate real world driving cycles for all desired vehicles. EcoGest is then useful to build the database information on macro-scale studies and even in micro-scale studies thus avoiding the expensive and time-consuming chassis dynamometer tests or real world measurements. Furthermore, it allows detailed studies about the influence on fuel consumption and tailpipe emissions of driving behavior, route topography, number of passengers, time spent at idle, cruise velocity and exhaust system design. In the present paper the spark ignition engine model, used in the main program EcoGest, is presented.

ECOGEST

EcoGest is a Visual Basic program that solves the dynamic laws of vehicles for specific acceleration and deceleration curves of typical driving modes (slow, normal and fast) or for given driving cycles (speed against time) and speed gearbox selection. In order to estimate tailpipe emissions and fuel consumption of a vehicle on a second by second basis, EcoGest main inputs are [5]:

- driving cycle followed by the vehicle (speed against time) or typical acceleration/deceleration curves;
- route topography (inclinations of the road);
- characteristics of the vehicle (frontal area, weight, dimensions of the tires, drag coefficient, coefficient of rolling resistance, the rotational inertia coefficient, transmission relations, transmission efficiency, power (or torque) curve of the engine);
- number of passengers;
- weight of the luggage;
- gearbox management maps;

– maps of fuel consumption, exhaust gas temperature and emissions of the engine, as a function of load and engine speed.

If the fuel consumption, exhaust gas temperature and emissions maps are not available from the OEMs they have to be estimated. In order to build those maps, a numerical model, presented in this paper, is being further developed. EcoGest uses an exhaust system model to correct the engine-out emissions and calculate the tailpipe emissions of HC, CO, NO_x and CO_2 [6]. Combining the information obtained from the model of the engine-out emissions with this exhaust system model, the average and instantaneous fuel consumption and tailpipe emissions are estimated along any trip. Standard driving cycles such as FTP 75, US06, SC03, UDC, EUDC and NEDC are in the database of the program EcoGest. This allows to simulate those driving cycles and to analyze the vehicle energetic and environmental performance when following the cycles.

ENGINE NUMERICAL MODEL

The engine model, incorporated in EcoGest, is a FORTRAN code called MotorIST. An investigation team leaded by professor Mendes-Lopes developed MotorIST. Now it is being further developed in order to improve the emission simulation, the spark timing and the exhaust temperature models, consider the cold-start and extend the combustion model to other alternative fuels, such as natural gas, LPG and methanol. MotorIST, in its essence, simulates the indicated cycle of a four stroke spark ignition engine, covering the range of time between the closing of the intake valves all the way to the opening of the exhaust valves. It uses a pseudo-zero dimensional approach where average values in space are considered while a full discretization in time is adopted. However, combustion modeling involves a space discretization in three zones: unburned mixture, flame (reaction zone, limited by the trailing edge and leading edge of the flame) and burned mixture. As input values MotorIST requires a full characterization of the geometrical parameters that characterize the engine, namely number of cylinders, bore, stroke, compression ratio, valve dimensions and number, valve timing (opening and closing angles) among others. In addition, it is necessary to describe the atmospheric conditions, namely ambient pressure and temperature. Finally, fuel composition (carbon and hydrogen atoms, octane number) and information about the air fuel mixture (λ) is also required. With these input variables, for different engine speeds and throttle positions, and considering:

– the compression and expansions strokes as isentropic evolutions,

– heat transfer to the inner surface of the combustion chamber (cylinder walls, piston and cylinder head),
– the energy released during combustion (taking into account dissociation) and expansion (taking into account CO, H_2 and O_2 recombination due to temperature decrease),
– actualization of the mixture properties due to temperature and composition changes along the cycle,
– and mechanical losses,

MotorIST computes global and in-cylinder time-resolved results: indicated and effective work, power, torque, mean pressure, fuel consumption, as well as exhaust temperature, engine-out emissions, heat loss, …, for the former, and pressure, temperature, heat transfer coefficient, …, for the latter.

STRUCTURE

MotorIST follows the logical structure presented in Figure 1. It contains a body of main calculations and several sub-models included in individualized and independent subroutines.

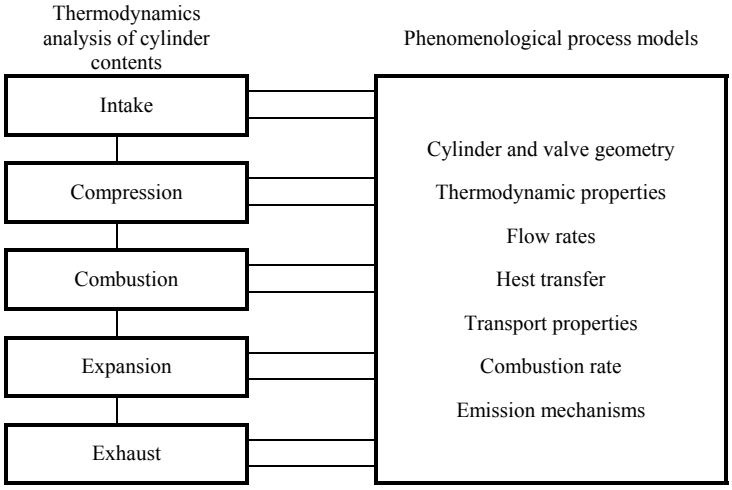

Figure 1. Logical structure of thermodynamic-based simulations of internal combustion engine operating cycle [7]

The main calculations are concerned with the evolution of temperature, pressure and work with the crank angle. The temperature calculations follow consecutive steps:

1. adiabatic compression or expansion [7]:

$$T = T'\left(V'/V\right)^{\gamma-1},\ V = \mathrm{f}(\theta) \tag{1}$$

2. heat release in combustion, Q_c, (with dissociation) or heat release in the recombination of the dissociated products (during expansion):

$$T = T + Q_c/\left(mc_v\right) \tag{2}$$

3. heat transfer to the inner cylinder surface (cylinder lateral walls, piston crown, engine head), Q_w :

$$T = T + Q_w/\left(mc_v\right) \tag{3}$$

4. gas properties actualization due to temperature changes and computation of composition changes due to combustion and recombination phenomena (occurring in step 2);
5. repetition of steps 1 to 3;
6. verification of the temperature T convergence and repetition from step 4 if necessary.

Pressure is obtained considering the ideal gas law [7]. Work is obtained by definition:

$$W = -\int p\,dV = -\frac{p+p'}{2}*\left(V - V'\right) \tag{4}$$

SUB-MODELS

The sub-models that assist the main calculations are volumetric efficiency, initial composition, heat transfer, spark timing, combustion, dissociation/recombination, knock, pollutants formation, blowdown, heat transfer during exhaust gas and mechanical losses.

Volumetric efficiency

This sub-model is based on the effect of different phenomena that affect the air flow rate as a function of speed (Figure 2). The goal of this model is to calculate the air quantity that enters the cylinders according to engine geometry, type of engine (normal, sport, variable admission, …), engine speed and throttle position. The strategy followed was to consider η_v as unity and add all the losses and gains due to quasi-static effects, charge heating, flow friction, choking, backflow, tuning and ram-effect. Each one of these losses and gains are computed from phenomenological models.

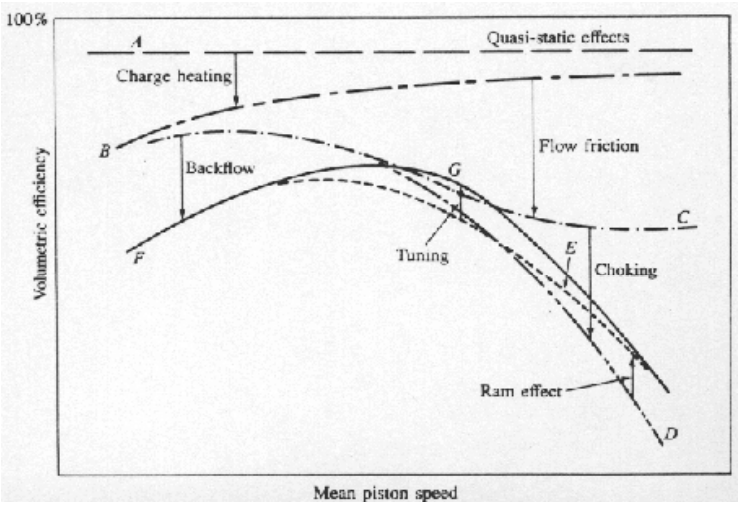

Figure 2. Effect on volumetric efficiency of different phenomena which affect the air flow rate as a function of speed. Solid line is final η_v versus speed curve [7]

Inicial composition

The initial gas composition in the cylinder is a mixture of the intake air, m_{ar}, admitted fuel, m_{fu}, and residual mass left over from the previous cycle, m_{res}. The total trapped mass m_T is then:

$$m_T = m_{ar} + m_{fu} + m_{res} \tag{5}$$

$$m_{ar} = \eta_v \times \frac{p_0 V}{\frac{R_o}{M_a} \times T_0} \times r_s \tag{6}$$

$$m_{fu} = \frac{m_a}{\lambda x} \tag{7}$$

$$m_{res} = V_{res} \times \frac{1.10\ p_0}{\frac{R_o}{M_{prod}} \times T_{res}} \tag{8}$$

m_T is used on reactants and products mass fractions calculations. Those are used in the computation of the mixture heat capacity. In Equation (8) a

10% overpressure was considered in the exhaust. The molar mass of products, M_{prod}, is calculated according to the λ value of the previous cycle.

Heat transfer

The heat transfer to/from the cylinder walls (lateral walls, piston crown and engine head), is calculated as follows, using the Nusselt correlation proposed by Annand [8]:

$$Q_w = \dot{Q}_w \Delta t \tag{9}$$

$$\dot{Q}_w = \sum_i A_i h_i \left(T_g - T_{wi} \right) \tag{10}$$

$$A_i = \mathrm{f}_i(\theta)\,,\ T_{wi} = \mathrm{g}_i(\theta) \tag{11}$$

$$h = \frac{\mathrm{Nu}k}{D} \tag{12}$$

$$\mathrm{Nu} = const.\,\mathrm{Re}^n \tag{13}$$

$$\mathrm{Re} = \frac{\overline{u}_e D}{\nu} \tag{14}$$

The *const* value and *n* values assume the values 0.35 to 0.80 (optional) and 0.70, respectively. *i* stands for lateral walls, piston crown or engine head.

Combustion

Duration and heat release

Combustion starts when the spark occurs and finishes when the trailing edge of the flame reaches the liner wall furtherest from the spark plug.

Spark timing was modeled according to an experimental map of a Fiat engine (2000 cm^3, 4 valve). It is function of engine speed and throttle position. But in order to get better results, the spark timing map for each modeled engine can be introduced in the program.

The leading edge flame front speed S_F is obtained from the turbulent flame speed S_t , and this one from the laminar flame speed S_u as follows [9], [10]:

$$\frac{S_F}{S_t} = \frac{\rho_u / \rho_b}{\left[\left(\rho_u / \rho_b\right) - 1\right] \dfrac{m_b}{m_u + m_b} + 1} \tag{15}$$

$$\frac{S_t}{S_u} = C \left(\frac{h_{ig} \bar{u}_e}{\nu} \right)^{\frac{1}{3}} \left(\frac{\bar{u}_e}{S_u} \right)^{\frac{1}{3}} \left(\frac{\rho_{ig}}{\rho_u} \right)^{\frac{1}{9}} \tag{16}$$

$$S_u\left(x_b\right) = S_u\left(x_b = 0\right)\left(1 - 2{,}06 x_b^{0{,}77}\right) \tag{17}$$

$$S_u\left(x_b = 0\right) = S_{u,0} \left(\frac{T_u}{T_0} \right)^{\alpha} \left(\frac{p}{p_0} \right)^{\beta} \tag{18}$$

$$S_{u,0} = B_m + B_\phi \left(\phi - \phi_m\right)^2 \tag{19}$$

The parameters α, β, ϕ, ϕ_m, B_m and B_ϕ can be found in the literature [11] and vary according to the considered fuel. The constant C was adjusted to give better model results.

The speed of the trailing edge of the flame was taken to be a fraction of S_F. This fraction is a function of throttle position and engine speed. The combustion heat release is computed knowing the burned mass fraction and the lower heating value for the considered fuel. The burned mass fraction was modeled according to a typical evolution (Figure 3).

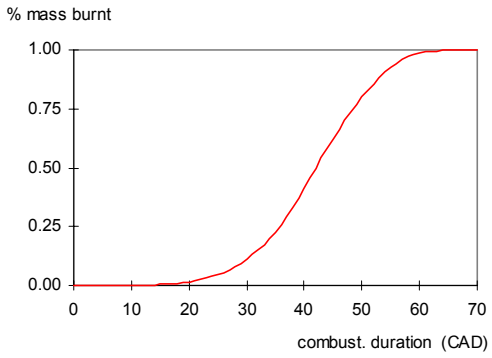

Figure 3. Typical evolution of the burned mass fraction with combustion duration illustrated for a combustion duration of 70 crank angle degrees

Dissociation/recombination

The calculation consists in getting the equilibrium composition of the combustion products considering the chemical species, N_2, H_2, CO_2, H_2O

and CO. The methods proposed by Goodger are applied [12]. The dissociation heat consumption is subtracted to the heat release from combustion, and recombination heat release is added.

Knock

Here it is adopted the so called *AnB* model, in which the auto-ignition lag is given by an Arrenhius function [7]:

$$\tau = A\ p^{-n} \exp\left(\frac{B}{T}\right) \tag{20}$$

For gasoline, Equation (20) is as follows [7]:

$$\tau = 1{,}869_x 10^{-2} \left(\frac{\mathrm{RON}}{100}\right)^{3{,}4017} p^{-1{,}7} \exp\left(\frac{3800}{T}\right) \tag{21}$$

In this equation τ is in s, p in kg.cm^{-2} and T in K. Knock occurs when, in crank angle basis:

$$\int_{\theta_0}^{\theta^*} \frac{d\theta}{\tau} \geq 6N \tag{22}$$

Pollutant emissions

Carbon monoxide

It is considered that the CO formation has three main contributions: incomplete combustion when the mixture is rich, incomplete combustion due to quenching and heterogeneous regions, and freezing of the chemical reaction of recombination. The first contribution is taken into account by simply considering the fuel combustion reaction. The second effect is taken into account by summing 0.1 % to the final CO concentration [7]. Finally, the third effect is obtained by considering the one step, reversible kinetic model [13]:

$$\frac{d[\mathrm{CO}]}{\mathrm{dt}} = k_b [\mathrm{CO_2}] - k_f [\mathrm{CO}][\mathrm{O_2}]^{1/2} \tag{23}$$

$$k_b = 2{,}3x10^6 \exp\left(-\frac{40}{R_0 T}\right) \tag{24}$$

$$k_f = k_b \, x \, K \tag{25}$$

R_0 is in kcal.mol^{-1}.K^{-1}, K is the dissociation constant. The pre-exponential factor was adjusted to agree better with model results and is know equal to 5x10^8.

Nitrogen oxides

It is only considered the thermal NO contribution to NO_x pollutant emissions. NO formation accounts only for kinetic mechanisms [7]:

$$\frac{d[\mathrm{NO}]}{dt} = \frac{2R_1\left\{1-\left([\mathrm{NO}]/[\mathrm{NO}]_e\right)^2\right\}}{1+\left([\mathrm{NO}]/[\mathrm{NO}]_e\right)R_1/\left(R_2+R_3\right)} \tag{26}$$

$$[\mathrm{NO}]_e = \left\{20{,}3\exp\left(-21650/T\right)[\mathrm{O}_2]_e[\mathrm{N}_2]_e\right\}^{1/2} \tag{27}$$

$$R_1 = 7{,}6x10^{13}\exp\left(-38000/T\right)[\mathrm{O}]_e[\mathrm{N}_2]_e \tag{28}$$

$$[\mathrm{O}]_e = 3{,}6x10^3\exp\left(-31090/T\right)\left\{[\mathrm{O}_2]_e/\left(R_0T\right)\right\}^{1/2} \tag{29}$$

The values of R_1 and $R_1/\left(R_2+R_3\right)$ can be found in the literature [7]. The combustion chamber is divided in three zones: unburned mixture, flame (reaction zone, limited by the trailing edge and leading edge of the flame) and burned mixture. The temperature in the flame and burned mixture is computed. Consequently, the NO concentration is calculated in these two zones.

Unburned hydrocarbons

It is considered that the HC formation has three main contributions: absorption/desorption from the oil layer, incomplete combustion due to quenching (in crevices and cylinder walls) and heterogeneous regions; and misfire or unfinished combustion. The first and third effects are not taken into account. The second effect is obtained by considering crevices (near the cylinder ring) and the cylinder walls. The quenching distance is computed by the Peclet number [14]:

$$d_{q2} = \frac{\mathrm{Pe}_2 \; k_f}{\rho_u \; S_u \; c_{p_f}} \tag{30}$$

$$\mathrm{Pe}_2 = 9{,}5\lambda\left(\frac{p}{3}\right)^{0{,}26\min\left(1,\lambda^2\right)} \tag{31}$$

$$\frac{\mathrm{Pe}_1}{\mathrm{Pe}_2} = \frac{d_{q1}}{d_{q2}} = 0{,}2 \tag{32}$$

Then the quenching volume is known and, consequently, the unburned HC mass is calculated. Then it is assumed that 85 % of those suffers a burnup process [7] that is kinetically controlled [11]:

$$\frac{d[\mathrm{HC}]}{\mathrm{dt}} = -6{,}7 \, x \, 10^{13} \; \exp\left(\frac{18\,735}{T}\right) x_{\mathrm{HC}} \; x_{\mathrm{O}_2} \left(\frac{p}{R_0 T}\right)^2 \tag{33}$$

Blowdown

The exhaust mass flow rate in the blowdown process is calculated according to the time duration and mass involved in the process. It is assumed that the time duration is two times the exhaust valve opening advance. Since blowdown is very rapid, the gas is assumed to expand isentropically from the pressure and temperature at the exhaust valve opening to a pressure equal to atmospheric p_0 plus an overpressure in the exhaust system. This overpressure is a function of engine speed and load, being typically 10% of p_0. The blowdown mass is the difference between the total mass and the one that stays inside the cylinder after blowdown (computed from the ideal gas law, the mentioned pressure, the calculated temperature after the isentropic expansion, and the cylinder volume at the end of blowdown - equal to the one at exhaust valve opening). The mass of forced exhaust gas is computed from the same temperature and pressure (both assumed not to change during the exhaust stroke) and the difference between the mentioned volume and the residual volume of the combustion chamber.

Heat transfer to the exhaust valve and manifold is considered from local Nusselt correlations that take into account the local geometry (valve head and stem, manifold) and assumed metal temperatures.

Mechanical losses

The origin of mechanical losses are considered to be i) friction and auxiliary equipment engagement losses and ii) pumping losses. The mean mechanical losses pressure is calculated as a function of engine speed and throttle position. The variation against engine speed is modeled by a second order polynomial function existing in the literature [15]. The variation against throttle position is modeled by a second order polynomial function of mean effective pressure. This induces an iterative process. The model parameters were fitted to experimental data, obtained from a monocylindrical Ford engine (673 cm^3) and four cylinders Fiat and Renault engines (2000 cm^3).

RESULTS

Comparison of model predictions with experimental data from Fiat, Volkswagen and Renault was performed. In general the predictions show good agreement with OEMs data (accuracy better than 20 %). Some of the obtained results are shown for maximum mean effective pressure *pme* as a function of engine speed (Figure 4) and volumetric efficiency (Figure 4), mechanical losses due to friction and engagement of auxiliary equipment (Figure 5), mechanical losses due to pumping (Figure 5), average total mechanical losses variation with load and speed and mechanical efficiency (Figure 6) and maximum mean effective pressure and specific fuel consumption (Figure 7). No comparisons were made for emission predictions.

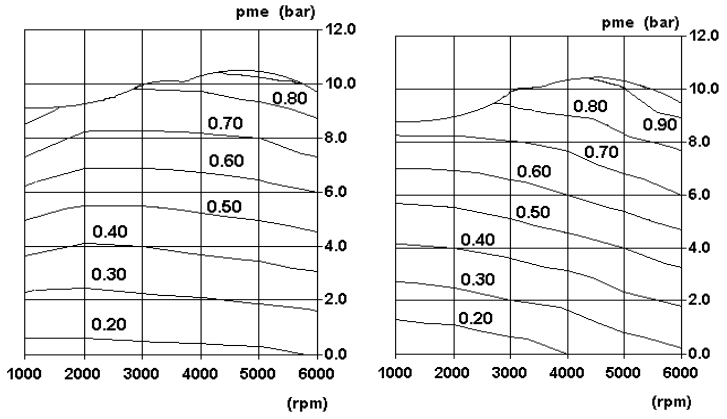

Figure 4. Maximum *pme* and volumetric efficiency-2000 cm^3 4 valve Volkswagen engine, λ =1. Volkswagen data on the left. Simulation predictions on the right

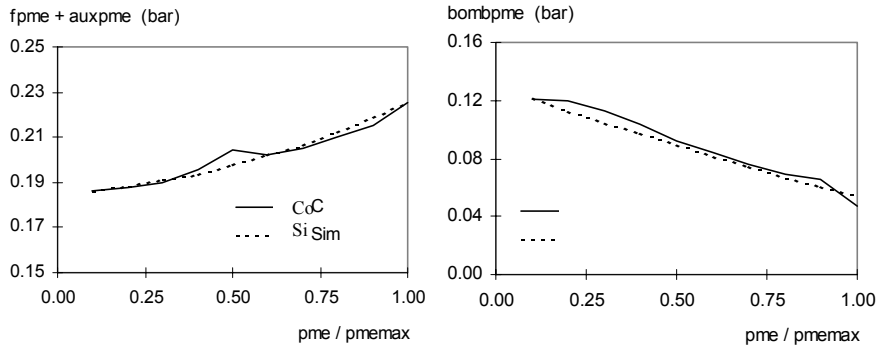

Figure 5. On the left variation of mechanical losses due to friction and auxiliaries engagement with load - 2000 cm^3 4 valve Fiat engine, λ =1. On the right variation of mechanical losses due to pumping with load - 2000 cm^3 4 valve Renault engine, λ =1

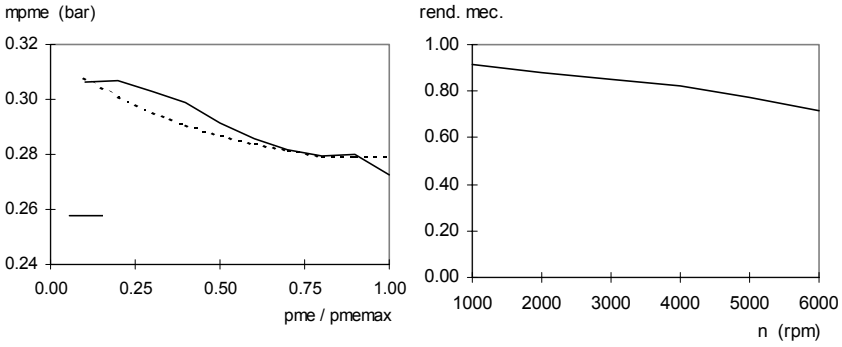

Figure 6. On the left variation of mean effective pressure of mechanical losses with load - 2000 cm^3 4 valve Renault engine, λ =1. Predicted variation of mechanical efficiency with engine speed on the right - 2000 cm^3 4 valve Renault engine, λ =1

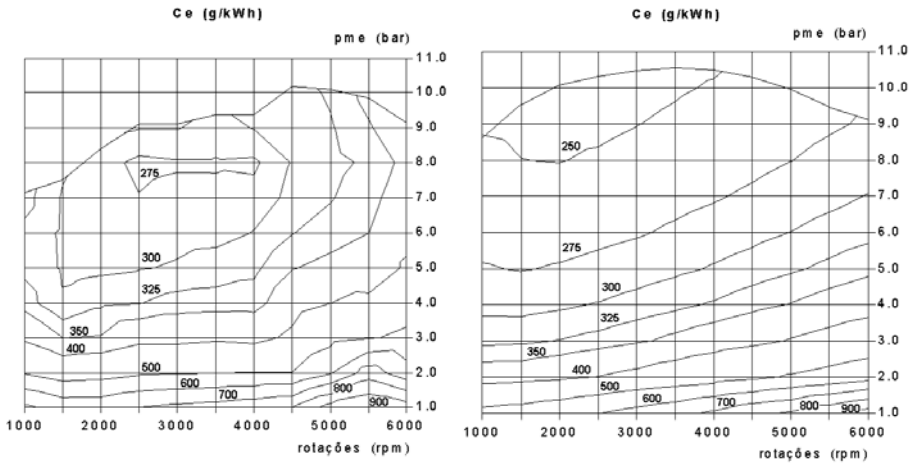

Figure 7. Maximum *pme* and specific fuel consumption - 2000 cm^3 4 valve Fiat engine, λ =1. Fiat data on the left. Simulation predictions on the right

CONCLUSIONS - FUTURE WORK

In the present paper the engine model developed for the micro-scale model EcoGest is presented. MotorIST, in its essence, simulates the indicated cycle of a four stroke spark ignition engine, covering the range of time between the closing of the intake valves all the way to the opening of the exhaust valves. It uses a pseudo-zero dimensional approach where average values in space are considered while a full discretization in time is adopted. As input values MotorIST requires a full characterization of the geometrical parameters that characterize the engine, namely number of cylinders, bore, stroke, compression ratio, valve dimensions and geometry, valve timing (opening and closing angles) among others. In addition, it is necessary to describe the atmospheric conditions, namely ambient pressure and temperature. Finally, fuel composition (carbon and hydrogen atoms, octane number) and information about the air fuel mixture (λ) is also required. With these input variables, and considering:

- the compression and expansions strokes as isentropic evolutions,
- heat transfer to the inner surface of the combustion chamber (cylinder walls, piston and cylinder head),

– the energy released during combustion (taking into account dissociation) and expansion (taking into account CO, H_2 and O_2 recombination due to temperature decrease),
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Comparison of model predictions with experimental data from Fiat, Volkswagen and Renault show good agreement (accuracy better than 20 %).

Experimental comparisons are needed to assess if the model is suitable for emissions analysis. The emission simulation models, the spark timing model and the exhaust temperature calculation should be improved. Cold-start should be added to the model. Extend the combustion model to other alternative fuels, such as natural gas, LPG and methanol is another goal to be achieved.

ACKNOWLEDGEMENTS

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NOMENCLATURE

A	area of the cylinder walls, m^2.
CO	carbon monoxide.
CO_2	carbon dioxide.
c_p	constant pressure heat capacity of gas mixture, $J.kg^{-1}.K^{-1}$.
c_v	constant volume heat capacity of gas mixture, $J.kg^{-1}.K^{-1}$.
c_{vf}	gas specific heat at flame conditions, $J.kg^{-1}.K^{-1}$.
D	cylinder diameter, m.
d_q	quenching distance, m.
H_2	Hydrogen.
H_2O	water.
HC	hydrocarbons.
h	interior heat transfer coefficient, $W.m^{-2}.K^{-1}$.
h_{ig}	mean distance between piston head and cylinder head, at ignition, m.
k	thermal conductivity of gas mixture, $W.m^{-1}.K^{-1}$.
K	dissociation constant, molar basis.

k_b	speed constant of inverse reaction.
k_f	speed constant of direct reaction or thermal conductivity of gas at flame conditions ($W.m^{-1}.K^{-1}$).
m_a	mass of air, kg.
M_a	molar mass of air, $kg.mol^{-1}$.
m_b	burned mass, kg.
m_{fu}	mass of fuel, kg.
M_{Prod}	molar mass of combustion products, $kg.mol^{-1}$.
m_{res}	mass of residual in-cylinder gas, kg.
m_T	total trapped mass, kg.
m_u	unburned mass, kg.
N	engine speed, rpm.
NO_x	oxides of nitrogen.
N_2	dioxide of nitrogen.
Nu	Nusselt number.
O, O_2	oxigen.
p	pressure of gas mixture, Pa.
p'	pressure of gas mixture, in the previous crank angle of discretization, Pa.
Pe	Peclet number.
p_0	ambient air pressure, Pa.
Q_c	heat release in combustion, J.
Q_W	convective heat transfer to/from cylinder walls, J.
$\dot{Q}_W$	convective heat transfer rate to/from cylinder walls, W.
pme	mean effective pressure, bar.
R_0	ideal gas constant, $8.3143\ kJ.kmol^{-1}.K^{-1}$.
Re	Reynolds number.
r_s	turbocharging ratio (ratio between air density out of the compressor and atmospheric air density), 1 stands for non turbocharged engines.
RON	research octane number.
S_F	flame front speed, $m.s^{-1}$.
S_u	laminar flame speed, $m.s^{-1}$.
$S_{u,0}$	laminar flame speed, at atmospheric conditions, $m.s^{-1}$.
S_t	turbulent flame speed, $m.s^{-1}$.
t	time, s.
T	temperature of gas mixture, K.
T'	temperature of gas mixture, in the previous crank angle of discritization, K.
T_g	temperature of gas mixture, K.
T_{res}	temperature of residual gas, K.
T_u	temperature of unburned mixture, K.
T_W	temperature of cylinder walls, K.
T_0	ambient air temperature, K.
$\overline{u}_e$	mean piston speed, $m.s^{-1}$.
V	volume of gas mixture, or cylinders capacity, m^3.
V'	volume of gas mixture, in the previous crank angle of discritization, m^3.
V_{res}	volume of residual gas, m^3.
W	work, J.
x	stoichiometric air fuel ratio.
x_b	burned gas diluent fraction.
x_{HC}	mole fraction of HC.
x_{O2}	mole fraction of O_2.

Greek symbols

Δt	time interval, s.
γ	ratio between c_p and c_v.
η_v	volumetric efficiency.
λ	coefficient of excess air.
ν	cinematic viscosity of gas mixture, $m^2.s^{-1}$.
θ	crank angle, º.
θ^*	crank angle for which it occurs knock, º.
θ_0	crank angle for which the chemical activity is still negligible, º.
ρ_b	density of burned gas, $kg.m^{-3}$.
ρ_{ig}	density of gas mixture at the moment of ignition, $kg.m^{-3}$.
ρ_u	density of unburned gas, $kg.m^{-3}$.
τ	autoignition lag, s.

Other symbols

[i]	concentration of species i.
$[i]_e$	equilibrium concentration of species i.

REFERENCES

1. US EPA *User's Guide to Mobile6.1 and Mobile6.2: Mobile Source Emission Factor Model*, Office of Transportation and Air Quality, United States Environmental Protection Agency, 2002.
2. Ntziachristos, L. and Samaras, Z. *COPERT III: Computer programme to calculate emissions from road transport - Methodology and emission factors (Version 2.1)*, European Environment Agency, 2000.
3. Sturm P. J., Hausberger S. Estimating Real World Emissions from Passenger Cars Use and Limitations of Instantaneous Emission Data. International Journal on Vehicle Design, 1999.
4. André, M., Pronello, C. Speed and Acceleration Impact on Pollutant emissions. SAE paper 961113.
5. Silva, C. M., Farias, T. L. and Mendes-Lopes, J. M. C. EcoGest - Numerical modelling of the dynamic, fuel consumption and tailpipe emissions of vehicles equipped with spark ignition engines. Proceedings of Urban Transport and the Environment 2002 - Eighth International Conference on Urban Transport and The Environment in the 21st Century; 13-15 March, 2002, Seville, Spain.
6. Silva, C. M., Farias, T. L. and Mendes-Lopes, J. M. C. Calculation of Tailpipe Emissions in EcoGest. Proceedings of the 7th International Conference on Energy for a Clean Environment; 7-10 July, 2003, Lisbon, Portugal.
7. Heywood, John B. *Internal Combustion Engine Fundamentals*, McGraw-Hill, 1988.
8. Annand, W. J. D. Heat transfer in the cylinders of reciprocating internal combustion engines. Proc. Inst. Mech. Engrs., vol. 177, n. 36, pp. 973-990, 1963
9. Tabaczynski, R J, Trinker, F H, & Shannon, B. A. S. Further refinement and validation of a turbulent flame propagation model for spark ignition engines. Combust. Flame, 39, pp. 111-121, 1980.

10. Keck, J. C. Turbulent flame structure and speed in spark-ignition engines, 19th Symp. (Intern.) on Combust., The Combustion Institute, pp. 1451-1466, 1982
11. Rhodes, D. B. and Keck, J. C. Laminar burning speed measurements of indolene-air diluent mixtures at high pressures and temperatures. SAE paper 850047.
12. Goodger, E. M. *Combustion calculations - theory, worked examples and problems*, The MacMillan Press Ltd, 1977.
13. Westbrook, C. K. and Dryer, F. L. Simplified reaction mechanisms for the oxidation of hydrocarbon fuels in flames. Combust. Sci. and Technology, vol. 27, pp. 31-43, 1981.
14. Lavoie, G. A. Correlations of combustion data for SI engines calculations - laminar flame speed, quench distance and global reaction rates. SAE paper 780229.
15. Barnes-Moss, H. W. A designer's viewpoint, in Passenger Car Engines. Conference Proceedings, pp. 133-147, Institution of Mechanical Engineers, London, 1975.