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ON DRIVING CYCLE SIMULATION METHODS IN ORDER TO IMPROVE FUEL CONSUMPTION

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Abstract: *One of the most important debates in automotive industry it is the improvement of fuel consumption. If we want to evaluate the vehicle's fuel consumption, it is common to perform driving cycle simulations. However, we can prescribe the vehicle's speed to exactly follow a function of time (quasi-stationary analysis) and the transient behaviour of the system is not fully taken into account. The direction of cause and effect is unnatural. We have a so called "driver controlled model", characterised by a driver who tries to achieve the driving cycle speed with the help of a proper position for the accelerator. Transient analysis is required by such a model. So, in order to be capable to understand the need for more accurate simulations, the paper shows a comparison between these two methods of simulation.*

Keywords – vehicle, modelling, simulation, analysis, driving cycle, fuel consumption

1. INTRODUCTION

With the increased concern over urban air pollution due to motor vehicles, there is a need for models of vehicle fuel consumption and exhaust emissions. One common approach of modeling vehicle generated pollution is to take emissions data from standard drive cycle tests on a chassis dynamometer. Application of the data to a given situation is accomplished by a number of adjustment factors to allow for the different running modes, speeds, temperatures, fuels used etc., /1/, /4/, /5/.
In 1998 the European Automobile Manufacturers Association (ACEA) committed to the EU on behalf of its members to reduce the average CO_2 emissions from their new car sales in the EU to 140 g/km by 2008. This is a reduction of 25% over 1995 levels, and equivalent to a fuel consumption of 6.0 litres per 100 km for petrol cars and 5.3 litres for diesel cars. In 1999, the Japan Automobile Manufacturers Association (JAMA) and the Korean Automobile Manufacturers Association (KAMA) made similar commitments for their EU sales. The only difference is that their target year to achieve an average 140 g/km CO_2 figure is one year later, 2009. All three associations, in other words, were given a decade to comply (fig. 1).

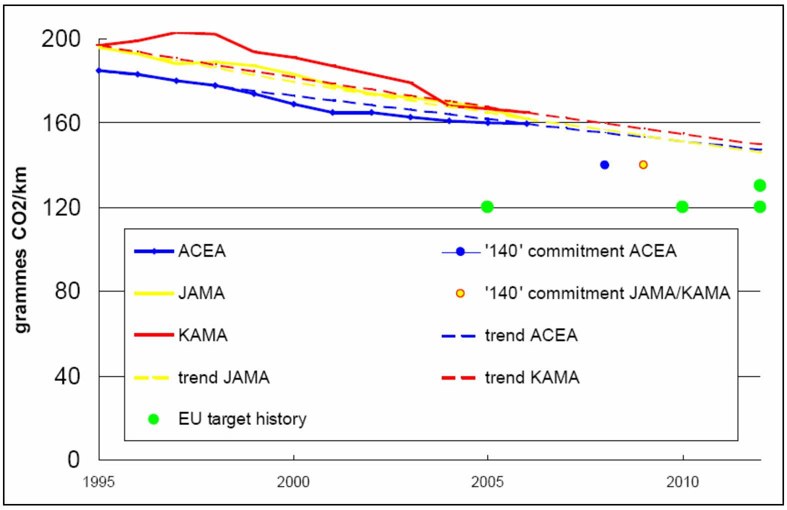

Fig. 1. Progress over time in the CO_2 commitment

As concerns fuel economy, it is a fundamental aspect in the development of a new car. The European Federation for Transport and Environment (T & E) presented a new directive which imposed a new standard regarding CO_2 emissions by 2012 (no more than 130 g CO_2/km), which is an indirect measure to control fuel consumption, /y/. T&E firmly believes that the EU should stick to achieving 120 g/km by 2012 through improved fuel efficiency of cars. Other measures should come on top of, rather than instead of, this measure. The target has been in place for 13 years now (since October 1994 when it was first put forward) so by 2012 the industry will have had 18 years of lead time to implement it. After a series of delays and weakening, the EU's credibility on climate change policy is at risk unless it maintains its longstanding commitment to the original target. Longer term targets are also necessary, not least to address the long - term challenge of climate change, but also to give long term certainty to the car industry. T&E has proposed a series of interim targets leading to 80 g/km by 2020, /5/.
This, in combination with requirements of short development period in the design process, calls for better simulation methods. As the chemical emissions (including CO_2 emission that gives the fuel consumption) are measured on certain driving cycle (e.g. the ECE Driving Cycle), it is important, in the design phase, to perform some driving cycle analysis.

2. SIGNIFICANT TERMS IN DYNAMIC SYSTEM MODELLING

In this paper it is about dynamic systems, in other words systems whose parameters vary in time. When trying to perform this driving cycle analysis, we use some models. The conceiving of a calculus model (understood as a system of equations) is the first step when trying to abstract the real phenomenon; thus the calculus model is nothing else but an idealization of the real physical phenomenon. Another way to the model definition is “everything needed to determine the solution”, /1/. The importance of a correct model results from the fact that a calculus process, no matter how sophisticated or precise would be, cannot substitute or avoid the drawbacks of a weak or wrong calculus model.
There are two types of dynamic systems modeling: quasi-stationary and transient. The conditions in a quasi-stationary analysis are only dependent on the present, not on the history. Conversely, in a model for transient analysis, the conditions at a certain time instant are dependent on the history, i.e. the previous states of the model. Transient analysis uses integration methods to follow a process in the time domain, while quasi-stationary analysis requires purely algebraic calculations at each time instant $t+\Delta t$ in a time instant sequence (1).
In a conventional quasi-stationary analysis of the driving cycle simulation, the vehicle speed is prescribed to follow a function of time exactly. Here, the transient behaviour of the system is not fully taken care of. The direction of cause and effect is unnatural. In the field of mechanics, such unnatural causality is sometimes referred to as inverse dynamics. The opposite is a driver controlled model, where an active driver model tries to achieve the driving cycle speed by choosing a proper accelerator pedal position (φ). Such a model requires a consistent transient analysis. A graphical representation of these two models is presented in figures 2 and 3. It is observed that the subsystems are connected with arrows. These arrows define the causality, i.e. the direction of cause and effect. In mathematical terms, it is the direction of the transport of variables between the subsystems of a system.

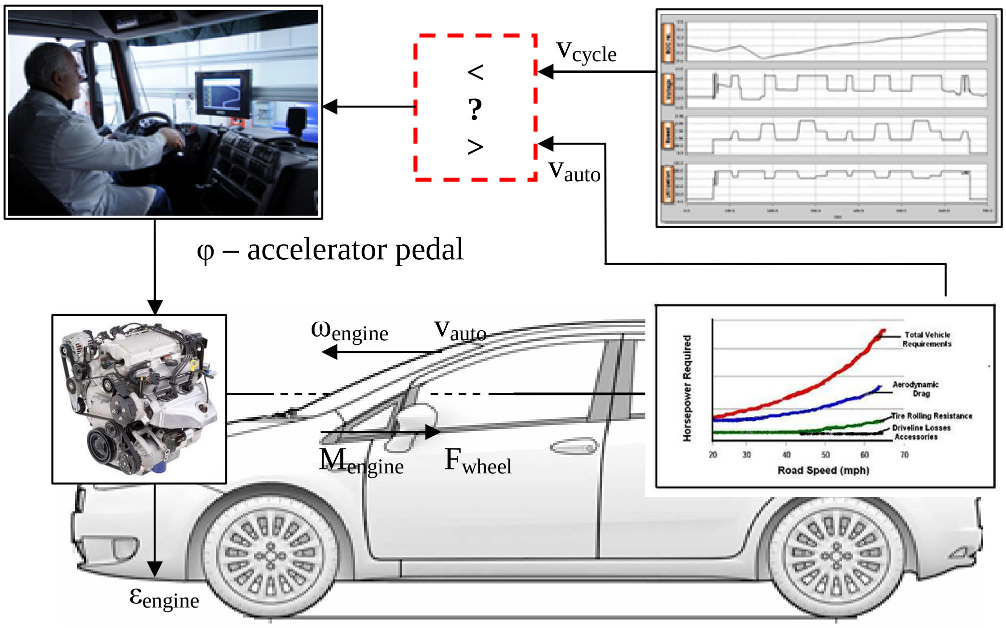

Fig.2. Driver controlled model

In the driver controlled model, the only signal from the driver is the accelerator pedal position. Obviously, one can also add other signals, such as: gear selection, brake pedal force etc., which have a natural causality, **from** the driver **to** the vehicle. The "driver" should here be understood as both human driver and control systems (e.g. Proportional Integrative Derivative controller – figure 4).

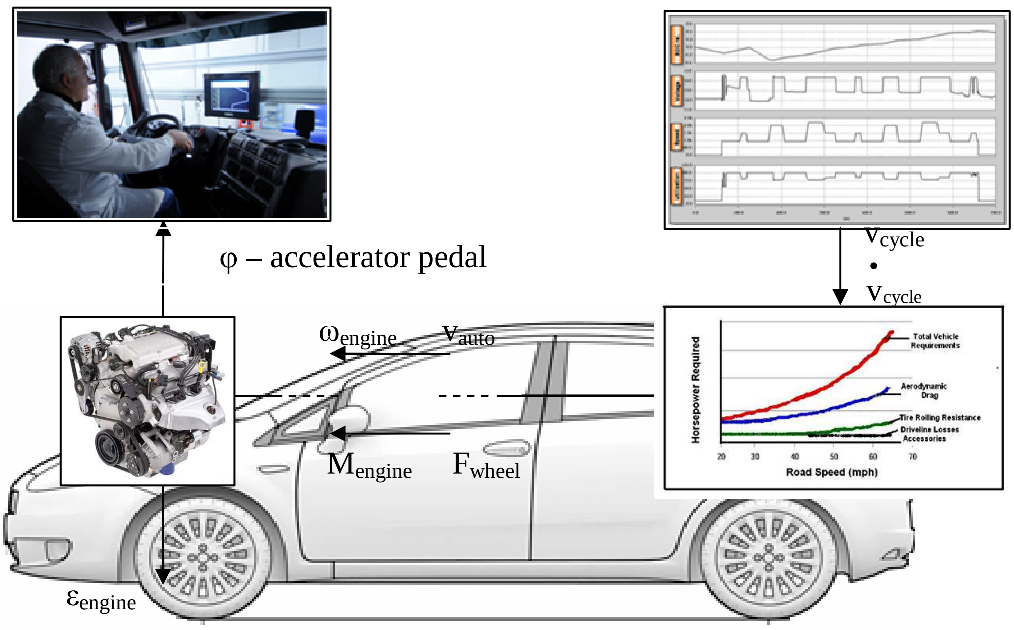

Fig.3. Conventional model

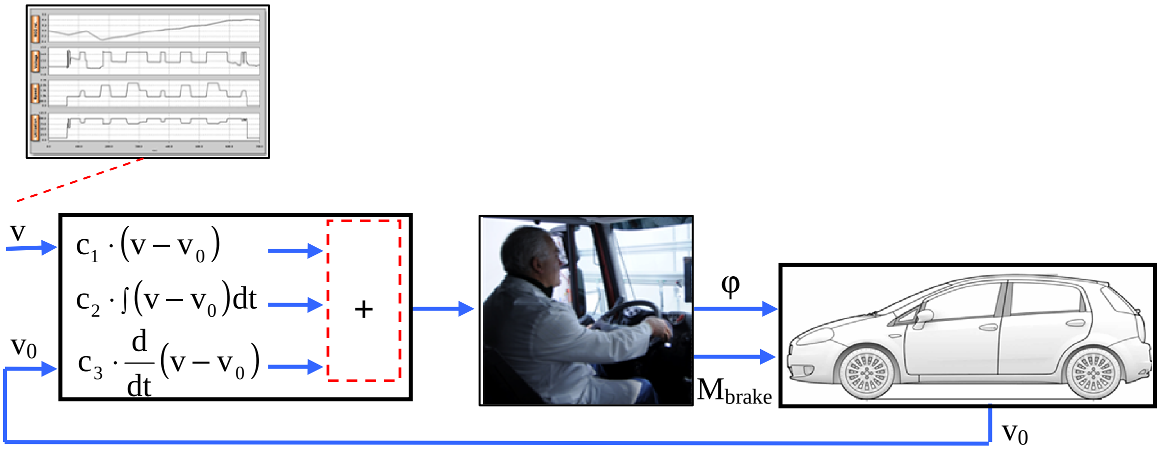

Fig.4. PID controller

Thus, the *driver* in the driver controlled model, acts as a vehicle speed regulator. Therefore, the driving cycle speed will not be followed exactly. Conversely, the *driver* in the *conventional model* has no control of the accelerator pedal position. It is what was previously stated as unnatural causality, typical for this kind of model.

3. DRIVING CYCLE ANALYSIS USING THE CONVENTIONAL MODEL

A mathematical description of this model is presented in the equations below. The quasi-stationary features of this model are defined by equations manipulations.

The subsystem **engine** is generally modelled starting from the engine steady state characteristics. In fact, it is about an engine torque vector as a function of accelerator pedal position, φ, between 0 and 1, and engine angular speed, ω:

$$M_{engine} = f_{engine}(\omega_{engine}, \varphi) \tag{1}$$

However, as concerns the turbocharged engine, it has to be said that it is difficult to evaluate the transient performances starting only from engine steady tests, because of the *turbo-lag* induced by the turbocharger (2, 3). Therefore, in order to find an appropriate engine map suitable for automotive simulation in transient mode, it has been proposed to build an engine torque map by using values resulted from intersection of the instant engine torque curve (the red curve, obtained on a flywheel engine test bench) with the engine torque curves obtained from steady state tests (the blue curves) – figure 5.

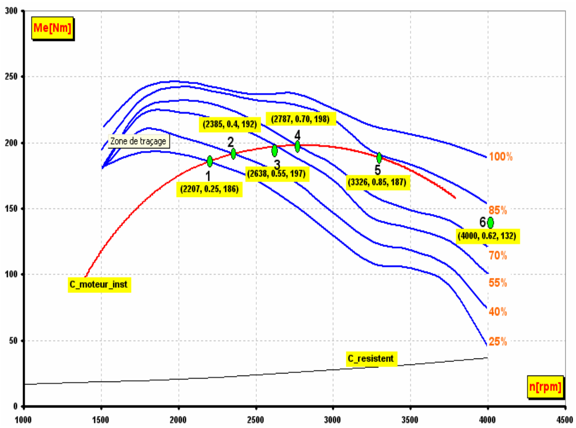

Fig.5. Building of an engine torque map

Subsystem	Relation	
engine flywheel	$M_{engine} - M_{flywheel} = J_{flywheel} \cdot \dot{\omega}_{flywheel}$	(2.1)
	$\omega_{engine} = \omega_{flywheel}$	(2.2)
gearbox - final drive	$M_{final\ shaft} = M_{flywheel} \cdot i_{cvk} \cdot i_0$	(3.1)
	$\omega_{flywheel} = \omega_{final\ shaft} \cdot i_{cvk} \cdot i_0$	(3.2)
wheel	$M_{wheel} - M_{brake} = F_{wheel} \cdot r_{wheel}$	(4.1)
	$M_{wheel} = M_{final\ shaft}$	(4.2)
	$v_{auto} = \omega_{wheel} \cdot r_{wheel}$	(4.3)
	$\omega_{wheel} = \omega_{final\ shaft}$	(4.4)
vehicle	$m_{auto} \cdot \dot{v}_{auto} = F_{wheel} - F_{wind} - \Sigma R$	(5.1)
	$v_{auto} = v_{driving\ cycle}$	(5.2)
	$\Sigma R = m_{auto} \cdot g \cdot f \cdot \cos\alpha + 0.5 \cdot \rho_{air} \cdot c_x \cdot A \cdot v^2_{auto} + m_{auto} \cdot g \cdot \sin\alpha$	(5.3)
driver	$i = f_{gear}(v_{auto}, F_{wheel})$	(6.1)
	$M_B = f_{brake}(F_{wheel})$	(6.2)
driving cycle	$v_{driving\ cycle} = f_{cycle}(t)$	(7.1)

Subsystem **engine flywheel** is modelled as a rotational mass inertia (relations 2.1 and 2.2). Subsystem **gearbox-final drive** is modelled as a loss free gear assembly, featuring i_{cvk} ratio discretely selectable and i_0 ratio (relations 3.1 and 3.2). Subsystem **wheel** is modelled as a driven wheel with eventually brake torque but without slip losses (relations 4.1 - 4.4). Subsystem **vehicle** is modelled as a translating mass inertia, facing rolling, aerodynamic, uphill and eventually wind resistances (relations 5.1 and 5.2). Subsystem **driver** is modelled only as a gear and brake torque selector (relations 6.1 and 6.2). Subsystem **driving cycle** is modelled as a time dependent vehicle speed (figure 1). Also, it is to be noted that the vehicle speed higher order time derivatives are defined by starting from the same $f_{cycle}(t)$ (relation 7.1).
Now summarizing the reasoning presented above, it can be said that vehicle speed and wheel force or torque are transmitted backwards, from the vehicle - through the transmission - to the engine. Thus, in order to achieve a certain prescribed vehicle speed, one can calculate with the aid of the equations above the necessary push on the accelerator pedal:

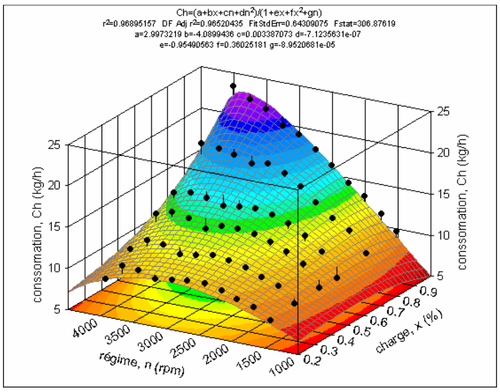

Fig.6. Engine fuel consumption map vs. engine speed and load

$$\varphi = f^{-1}_{engine}(\omega_{engine}, M_{engine}) \qquad (8)$$

Once having this unknown variable, the fuel consumption can be found if using an analytical function representing the fuel consumption versus speed and accelerator pedal push:

$$C_{h\_engine} = f_{engine}(\omega_{engine}, \varphi) \qquad (9)$$

An example of such a function – obtained for the same turbocharged diesel engine used to draw the curves from figure 5 – is presented in figure 6. As noticed, this function is obtained also by starting from engine steady-state tests, /2/.

4. CONCLUSIONS

An important advantage of this model is the possibility of comparison between different cars following a driving cycle in exactly the same way.
The driver controlled model is designed for advanced studies on the automotive power transfer and management control systems. Also, these studies demand real transient analysis evaluation for obtaining reliable results.

The conventional quasi-stationary model has the characteristic of not describing the phenomenon by a natural causality. Another difficulty consists in having an appropriate engine torque map. However, the model mentioned above is still to be used in order to have a much simpler approach

The dynamic systems simulation can be carried out, however, very efficiently with modern commercial dedicated software, such as MATLAB/Simulink™, AMESim™, ADVISOR™. Basically, they demand a model characterisation on an assignment blocks form. A special attention has to be offered in order to set the natural causality between the existing subsystems. If that is not possible, the algebraic loops take place and the simulation model will be compromised.

5. REFERENCES

(1) JACOBSON, B. – On vehicle driving cycle simulation, SAE Paper no 950031

(2) PODEVIN, P., DESCOMBES, G., TABACU, I., ZAHARIA, C., CLENCI, A. -Aspects concerning instant torque evaluation when accelerating a turbocharged diesel engine, CONAT International Congress on Automotive, Braşov, Romania, 2004, Paper code 1066

(3) TABACU, I., ZAHARIA, C., CLENCI, A. – Simularea regimurilor tranzitorii pe standul motor cu volant de inerție, Conferința Națională de Mecanica Solidelor, Pitești, Romania, 2002.

(4) D.J. WILLIAMS, D.Y.C. LEUNG - Modeling of motor vehicle fuel consumption and emissions using a power-based model, Environmental Monitoring and Assessment 65: 21–29, 2000, Kluwer Academic Publishers, Printed in the Netherlands.

(5) Reducing CO2 emissions from new cars, 2006 progress report on the car industry's voluntary commitment, September 2007