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Parametric study on the influence of driving cycles on fuel consumption for different powertrains

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Abstract
When assessing the performance of powertrain technology, one refers to the raw energy demand resulting from applying a predefined driving cycle. The choice of that cycle has huge implications on the final result, especially for hybrid systems, where the energy buffer's state depends not only on the current speed, but also on previous events. This paper quantifies that influence, by introducing characteristic cycle parameters such as average and top speed, and outlining their influence on fuel consumption.

Introduction
The ongoing debate on petroleum scarcity and global warming through anthropogenic CO2 emissions has given a new momentum to the development of alternative powertrain systems. Indeed, the transportation sector accounts for roughly one third of the worldwide secondary energy demand, mostly in form of oil based products such as gasoline [1], so efforts have to be made.
The strong dependence on oil is mainly the result of the unique properties of liquid hydrocarbon fuels. Their relatively high energy density and the ease of handling liquids make them an ideal energy carrier for automotive applications. Compared to current battery technology, the volume and mass requirements to guarantee long driving ranges at high performance are negligible. Also, refueling is simply a matter of pumping liquids from one recipient to another, taking mere minutes - as opposed to hours of nightly recharging.
On the other hand, the internal combustion engine is rather unsuited for propulsion applications. Its maximum torque output depends on the engine's rotation speed, which is limited by the combustion process and the system's thermal stability. A variable transmission is necessary to adapt the engine's rather narrow field of operation to the broad scope of torque-demands in an automobile. Moreover, the thermodynamic process efficiency (tank to wheel) depends heavily on the operation conditions. In general, higher loads yield better efficiency.
Electric engines are much better suited, since they provide an almost speed-independent maximum torque (up to maximum power) over a very wide range of rotational speeds, they can generate torque at standstill and the local efficiency is in generally much higher (not Carnot limited) and less operation point dependent.
However, current battery technologies do not reach the energy density of hydrocarbon fuels. So the purely battery electric vehicle cannot compete (under current conditions [2]). That's why there is a big interest in hybrid systems.
The presence of both a conventional and an electric powertrain allows decoupling the torque demand at the wheel from the energy conversion process in the internal combustion engine. Thus the electric system compensates for the missing resp. stores the excess energy when the engine is operated at a point of better efficiency, compared to the one that would usually be dictated by the torque demand and the wheels' rotational speed. Also the presence of electric torque converters enables recuperative braking, further increasing efficiency.
Unfortunately, hybrids are very complex systems. Each additional component requires special care, both for integrating it in the system and optimizing it on its own. Also, from a control engineering point of view, hybridization introduces a new degree of freedom: the choice of which energy source to use at a particular time. Much work has been done in both areas, and thus many approaches exist as of today.
However their goal is the same: reduce fuel consumption compared to the corresponding conventional vehicle. Usually, the consumption is determined by operating the vehicle on one of the many test

cycles that exist today. If vehicles have similar mass and performance capacities, then the mileage information obtained in a common cycle is comparable and gives a direct assessment as to the benefits of a tested powertrain. This holds true for a given cycle, but change the cycle and tables may be turned.

Methods and Materials

- THELMA

This study was motivated by ETHZ-LAV's participation in the CCEM project THELMA, "technology based electric mobility assessment". It is aim is to provide an integrated assessment of powertrain electrification on the Swiss transport sector and the impacts on both the Swiss electric grid and Switzerland as a whole. To that aim, a broad range of current and future vehicle designs, representative for the national fleet, are analyzed using drivetrain simulation and life cycle analysis.
Scenarios and traffic simulations are then used to determine the additional load on the electric grid. Its effects are then analyzed in power transmission simulations. The goal is to comprehensively assess the tradeoffs and sustainability implications of the increased use of light electric vehicles, as compared to other drivetrains and fuels.

ETHZ-LAV is cooperating with PSI-LEA on Work Package 2, "Vehicle Simulation and Powertrain Assessment". The main objective is the computation of energy "consumption" data for a large set of vehicles (in terms of class, size and powertrain technology) under a wide range of operation conditions. Concretely, the behavior of conventional, hybrid electric (including plug-in hybrids and range extenders), fuel cell and battery electric vehicles belonging to separate vehicle classes like light utility vehicles, S.U.V.'s, sedans, compacts, sports-cars and "city"-cars are compared.

- Calculation of fuel-consumption data

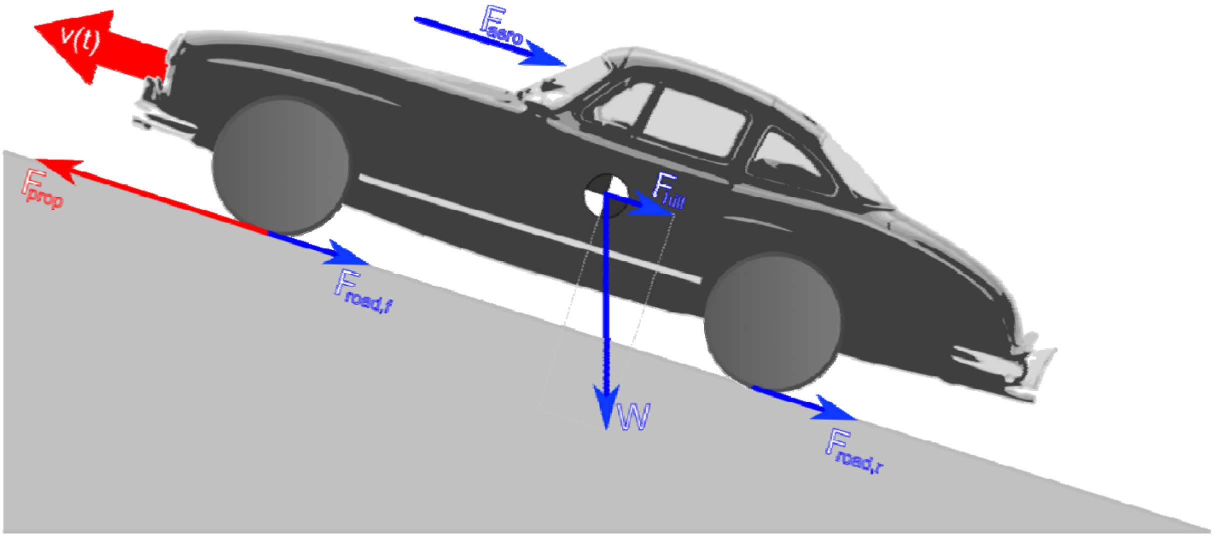

Figure 1: main forces acting on a vehicle going uphill at velocity $v(t)$

The fuel and electricity consumption data is obtained from simple backward (quasi-static) 1-D vehicle dynamics and powertrain models. In fact, the main forces acting on a vehicle (see

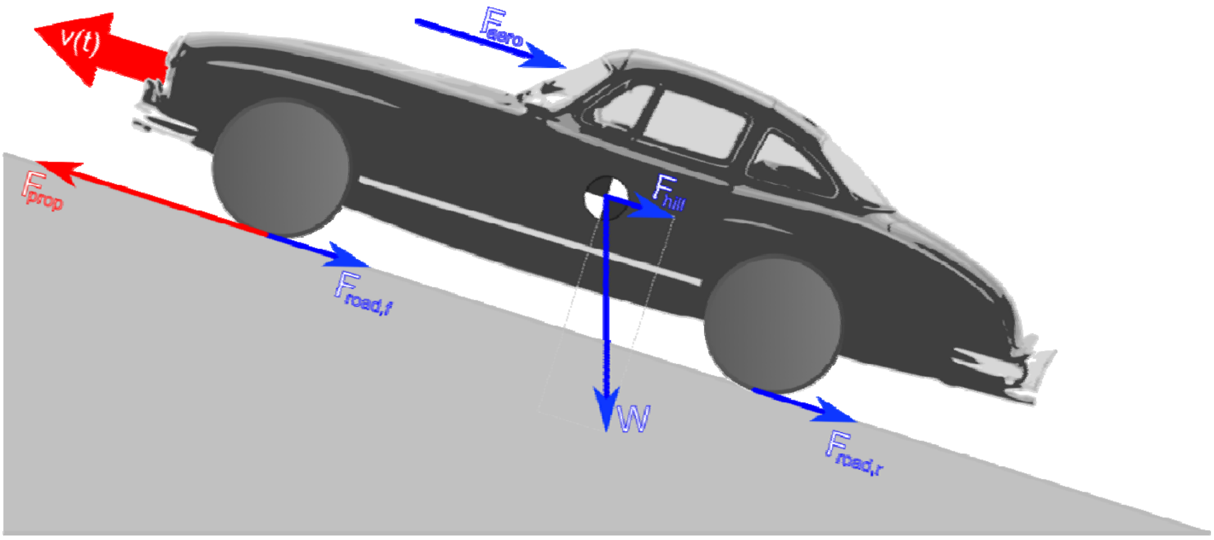

Figure 1) travelling at a constant speed v are [9]:

propulsion force	F_{prop}
hill climbing force	$F_{hill} = m_{car} \cdot g \cdot \sin\alpha$
aerodynamic drag	$F_{aero} = \frac{1}{2} \cdot \rho_{air} \cdot c_d \cdot A_r \cdot v^2$
Tire rolling resistance	$F_{road,r} + F_{road,f} = m_{car} \cdot g \cdot \cos\alpha \cdot c_r$

Newton's second law then yields an expression for the propulsion force demand of the dynamic system, assuming that the vehicle speed $v(t)$ is known in advance (backward simulation):

$$F_{prop}(t) = m_{car}\frac{dv(t)}{dt} + \frac{1}{2} \cdot \rho_{air} \cdot c_d \cdot A_r \cdot v^2 + m_{car} \cdot g \cdot (\cos\alpha \cdot c_r + \sin\alpha)$$

This information serves as basis for a powertrain simulation. In a conventional vehicle (see Figure 2), the wheels, the differential and gearbox define the engine's operation point.

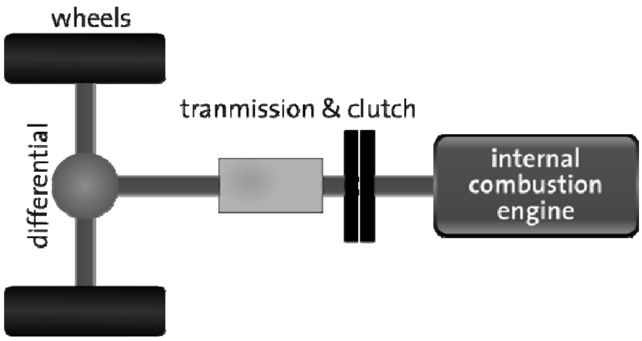

Figure 2: schematic representation of a conventional powertrain

$$\begin{cases} \omega_e = v(t) \cdot \dfrac{1}{r_{wheel}} \cdot \gamma_d \cdot \gamma_t(t) \\ T_e = F_{prop} \cdot r_{wheel} \cdot \dfrac{1}{\gamma_d} \cdot \dfrac{1}{\gamma_t(t)} \cdot \dfrac{1}{\eta_{mech}} \end{cases}$$

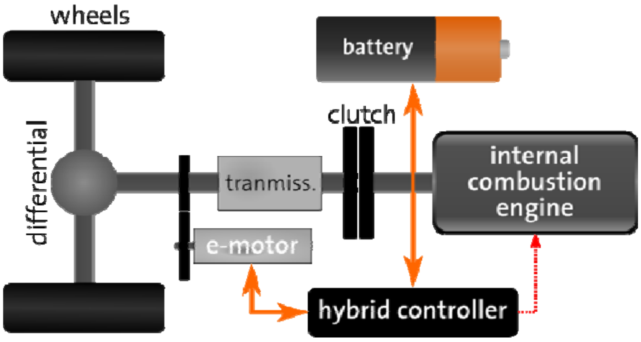

Figure 3: powertrain layout of a parallel hybrid

In the parallel hybrid configuration shown in Figure 3, an electric motor can add (or subtract) torque from the main shaft over a reduction gear. Thus the engine load can be shifted to higher or lower settings (within the boundaries of the electric system):

$$\begin{cases} \omega_e = v(t) \cdot \dfrac{1}{r_{wheel}} \cdot \gamma_d \cdot \gamma_t(t) \\ T_e = \left(F_{prop} \cdot r_{wheel} \cdot \dfrac{1}{\gamma_d} - T_{e-motor} \cdot \dfrac{1}{\gamma_e}\right) \cdot \dfrac{1}{\gamma_t(t)} \cdot \dfrac{1}{\eta_{mech}} \end{cases}$$

In the simulations, dynamic programming is used to find the optimal control input (motor torque) and gearbox setting to reduce fuel consumption to a minimum. The employed algorithm follows the methodology of [6] very closely. The electric engine and ICE engine are represented using Willans' approach [7], derived from efficiency maps. The battery model is based on [8], using the parameters for Li-ION batteries.

- **Driving Cycles**

The above simulation methodology can be applied to any number of vehicles, as long as certain parameters such as mass, aerodynamic properties, etc. are known. Then, as stated earlier, the main influence on fuel consumption is the speed profile (i.e. $v(t)$). But since the aim of THELMA is to assess the Swiss transportation system, those profiles have to be carefully picked from the many official driving cycles available in order to reflect Swiss driving conditions. Also, one of THELMA's research goals is to assess the driver's influence with respect driver aggressiveness. That's why a method of characterizing cycles is required.

A simple approach is to look at the average and maximum speeds. Assuming only partial (no mixed) cycles are looked at, the maximum speed is considered a proxy of the speed limit, which in turn is an indicator of the road type (urban, rural, motorway). The average velocity on the other hand is a result of traffic congestion; the further away it is from the maximum (closer to 0), the more viscous the traffic flow (the average velocity is supposed to never exceed the legal limitation).

Those are the only two parameters that will be looked at in this preliminary study. However in literature, drive cycle characterization is mentioned in the context of tail pipe emission forecasting. [10] uses the discrete probability distribution, measuring the chance of the vehicle accelerating at given intensity ($^{m}/_{s^2}$ or g) when at a known speed. The aim is to predict consumption and emission data by extrapolating measured data from known cycles to unknown cycles. Figure 4 shows the discrete probability function in a representation, inspired by [10].

In this representation, recognizing the traffic conditions is not obvious; but contrary to the simple v_{avg} - v_{max} approach, the dynamics of the cycles are captured. Further study must show, how those quantities are linked and if it is possible to integrate the concept of driver aggressiveness.

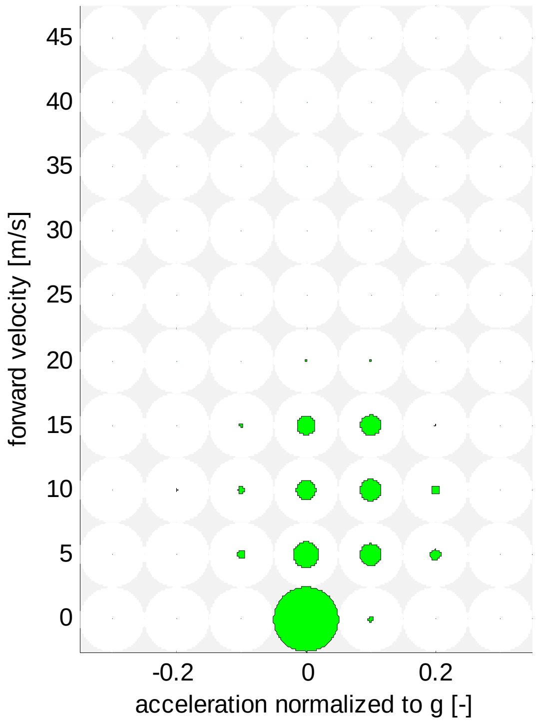

Figure 4: example of drive cycle characterization using the kinetic parameters described in [10] (here on ARTEMIS urban). Note that the circles represent the probability of having some acceleration at a known speed, relative to the most likely condition – in this case no acceleration at standstill.

Results and Discussion

Figure 5 shows how several widely used driving cycles would rank in the proposed classification scheme. It becomes clear that the maximum speed alone does not reflect the type of road. There are strong regional differences.

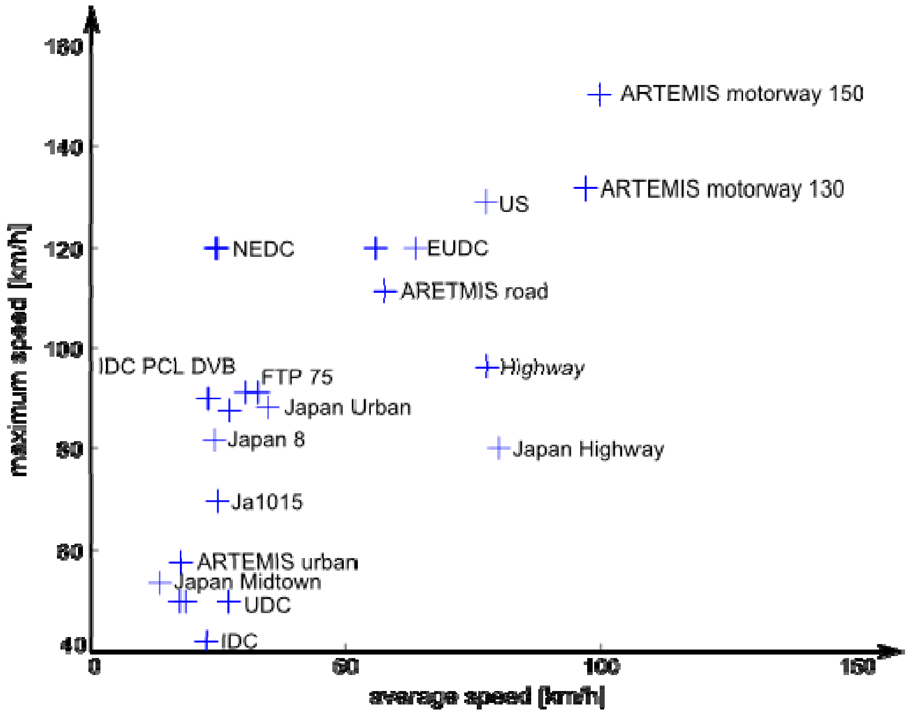

Figure 5: spread in a maximum-speed / average-speed classification scheme

In this preliminary study, the ARTEMIS [5] and the cycles from [4] are combined to cover a broad range of average and top speeds. Examples:

cycle	v_{avg} [km/h]	v_{max} [km/h]
ARTEMIS urban (scaled)	11.0	22.9
ARTEMIS free flow urban (scaled)	16.6	30.0
Constant 30 km/h	30.0	30.0
ARTEMIS urban dense + flowing stable	17.5	48.9
ARTEMIS pre-road	30.5	49.8
Constant 50	50.0	50.0
ARTEMIS free flow	27.8	57.7

Using the vehicle simulation explained above, fuel consumption data was computed. As can be seen from Figure 6, the different cycles in a series (having the same maximum speed) follow a more or less smoothly decaying curve. Now the fact that fuel consumption drops with increasing average speed was to be expected as in conventional powertrains, since high loads generally give better thermodynamic efficiency. But the fact that the points aren't scattered wildly suggests that the initial assumption (i.e. v_{max} is a proxy for the kind of road) is admissible.
At the same time it underlines the need for additional parameters: for instance, the motorway energy demand between 60-90 km/h is much higher than the one on a rural road for the same average speed. This is because the motorway cycle must have at least one peak at 120 km/h, and is therefore much more "aggressive" in terms of acceleration. For the rural cycle, there are two options: either there truly is some congestion reducing the overall traffic flow, or there may just as well be stop signs along the way, forcing the only vehicle on the road to stop in regular intervals; a fact that cannot captured by this simple model.

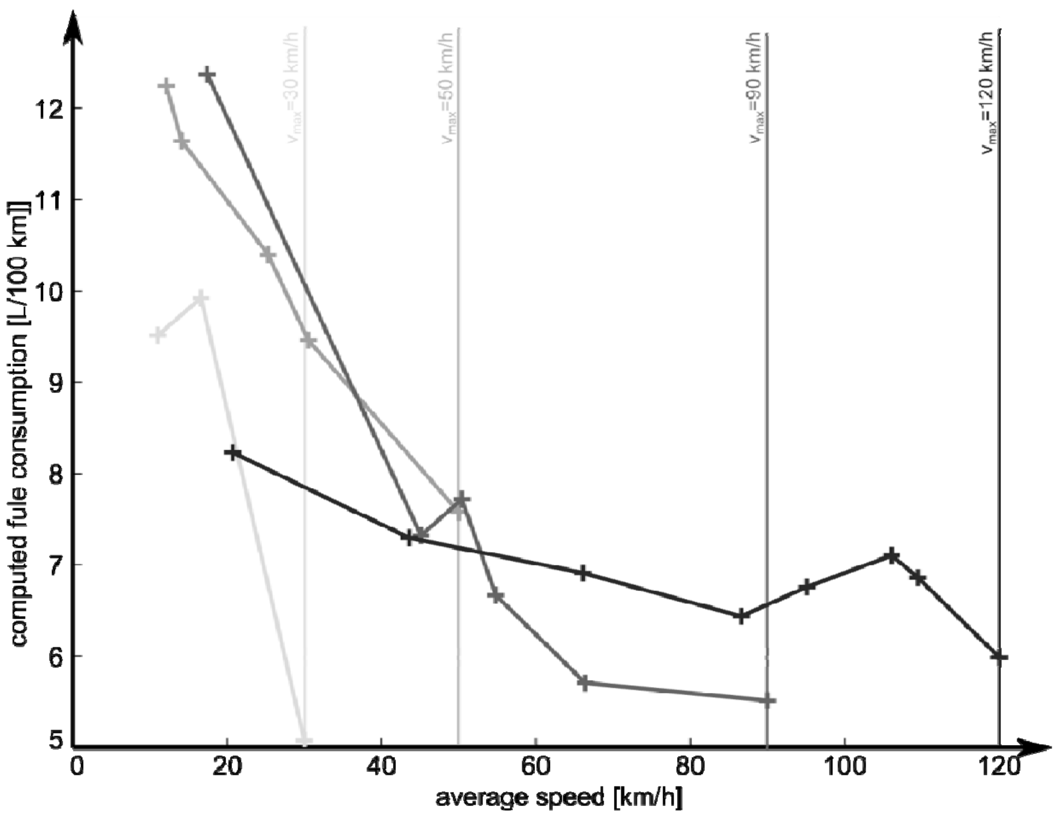

Figure 6: fuel consumption for a 1500 kg, 80kW conventional vehicle

Outlook

The aforementioned “kinematic parameters” described in [10] will be used to conduct a more detailed analysis. This will allow to capture the dynamics in much better detail. The results from this study will serve as a basis to relate the matrix plots to traffic conditions.

Then, the study will have to be extended to larger sets of vehicles (weight, chassis, tires etc.) and hybrid powertrains.

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Table of symbols

Symbol	Unit	Descritpion
F_{prop}	$[N]$	Longitudinal vehicle propulsion force
F_{hill}	$[N]$	Hill climbing (or “descending”) force
$F_{road,r}$	$[N]$	Tire rolling (or road) resistance at the front wheels
$F_{road,r}$	$[N]$	Tire rolling (or road) resistance at the rear wheels
m_{car}	$[kg]$	Total mass of the vehicle
g	$[{}^{kg}/_{m \cdot s^2}]$	Earth acceleration
α	$[rad]$	Road inclination
ρ_{atr}	$[{}^{kg}/_{m^3}]$	Density of the ambient air
c_d	$[-]$	Aerodynamic drag coefficient
A_r	$[m^2]$	Reference area for aerodynamic drag calculation
v	$[{}^{m}/_{s}]$	Vehicle’s forward velocity
c_r	$[-]$	Tire rolling resistance → friction coefficient
ω_e	$[{}^{rad}/_{s}]$	rotation speed of the internal combustion engine
T_e	$[Nm]$	Load of the Internal combustion engine
r_{wheel}	$[m]$	Wheel radius
γ_d	$[-]$	Fixed gear ratio of the differential
γ_t	$[-]$	Variable transmission ratio of the gear box
η_{mech}	$[-]$	Combined efficiency of gears, differential, …
γ_e	$[-]$	Fixed ratio of the electric motor’s reduction gear (hybrid only)