Dossier

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A Phenomenological Heat Transfer Model of SI Engines – Application to the Simulation of a Full-Hybrid Vehicle

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Résumé — Un modèle phénoménologique de transfert thermique au sein de moteurs à allumage commandé — Application à la simulation d'un véhicule full-hybride — Un véhicule hybride électrique permet des économies de carburant non négligeables grâce à une meilleure utilisation du Moteur à Combustion Interne (MCI) sur des points de régime-charge à meilleurs rendements. Cependant ce fonctionnement particulier a un impact sur le comportement thermique du MCI, en particulier, pendant sa période de chauffe après un départ à froid. En effet, le moteur peut être arrêté lorsque le véhicule est à l'arrêt (système *Stop&Start*) ainsi que pendant les phases de propulsion en électrique pur (possible à vitesse faible si l'état de charge de la batterie est suffisamment élevé) entraînant un manque de source de chaleur et donc un ralentissement du réchauffement du moteur. De plus, l'utilisation du MCI avec des charges plus importantes lors de la recharge des batteries provoque une augmentation de la puissance thermique disponible issue de la combustion. Les stratégies de contrôle d'un véhicule hybride (répartition d'énergie entre les deux types de propulsion : thermique et électrique) ont un effet important sur sa consommation finale. Ainsi, la simulation phénoménologique 0D de véhicules hybrides est utile afin d'évaluer l'efficacité de ces stratégies. Cependant, il existe un nombre très limité d'études où la période de chauffe du moteur thermique est prise en compte dans ce type de simulations. Un outil de simulation utilisant le logiciel Amesim a été développé afin de simuler la montée en température du MCI utilisé dans un système de propulsion hybride parallèle. Le modèle est construit afin de prendre en compte les phénomènes thermiques ayant lieu au sein de cet élément. Un modèle thermodynamique est couplé à un modèle de transferts thermiques représentant les différentes parties métalliques ainsi que les différents fluides (eau de refroidissement et huile de lubrification). La dépendance de leur température moyenne en fonction de différents paramètres comme la vitesse, ou bien la charge est étudiée dans le but de réduire la consommation de carburant. Le modèle thermique du moteur à combustion interne est finalement intégré dans une simulation d'un véhicule complet. Le comportement thermique d'un véhicule full-hybride électrique parallèle utilisant un moteur à allumage commandé est alors présenté utilisant cet outil de simulation. Les résultats de simulation montrent un impact certain de l'utilisation particulière du MCI dans ce type de véhicule sur le comportement thermique global de cet élément. En particulier, il semble que le rendement du moteur thermique est moins pénalisé que ce qui était attendu lors du fonctionnement à faible température. Finalement, une étude paramétrique du moteur modélisé ainsi qu'une recherche sur les possibilités d'optimisation du fonctionnement du moteur lors de la phase de montée en température sont réalisées.

Abstract — A Phenomenological Heat Transfer Model of SI Engines – Application to the Simulation of a Full-Hybrid Vehicle — *A hybrid thermal-electric vehicle allows some significant fuel economy due to its peculiar use of the Internal Combustion Engine (ICE) that runs with better efficiency. However, this propulsion system impacts its thermal behaviour, especially during its warm-up after a cold start. The ICE can indeed be shut down when the vehicle is stopped (Stop&Start system) and during full-electric propulsion mode (allowed at light speed and load if the battery state of charge is high enough) resulting in a lack of heat source and a slow down of the warm-up. Moreover, the use of the ICE at higher loads while charging the batteries provides an increase of the heating power generated by the combustion. Control strategies in a hybrid vehicle (energy repartition between the two propulsions: thermal and electric) have a significant effect on its final consumption. Therefore, the simulation of hybrid vehicles is then useful to evaluate the efficiency of these strategies. However, the consideration of the warm-up of the ICE in such a propulsion system was done in only few published studies. A simulation tool using the Amesim software has been developed in order to simulate the warm-up of an ICE used in a hybrid parallel propulsion system. The corresponding model is developed in order to take into account the thermal phenomena occurring between the different ICE components. Thus, a thermodynamic model is coupled with a thermal model of the metallic parts and the different fluid loops (water and oil). Their mean temperature dependence with different parameters like speed, the load, the cylinder geometry and the spark advance, is studied with the aim at reducing fuel consumption. The thermal model of the engine is finally integrated in a simulation of the whole vehicle. The thermal behaviour of a parallel electric full-hybrid vehicle using a spark ignition engine is then presented using this simulation tool. The simulation results show the impact of the peculiar use of the ICE on its thermal behaviour. Especially, it appears that the efficiency of the engine is less penalized than expected by the cold state of the engine. Finally, a parametric study of the modeled engine and a research of a possible optimization of the engine efficiency and the warm-up period are done.*

NOMENCLATURE

A_i	Thermal exchange area of part i (m^2)
BMEP	Break Mean Effective Pressure (bar)
c_{Dim}	Caracteristic dimension of flow (m)
C_i	Thermal capacity of i ($J.kg^{-1}.K^{-1}$)
Δt_{Charge}	Battery charging delay (s)
$FMEP_i$	Friction Mean Effective Pressure on i (bar)
λ_i	Thermal conductivity of i ($W.m^{-1}.K^{-1}$)
Gr	Grashof number
h_i	Convection coefficient of fluid i ($W.m^{-2}.K^{-1}$)
m_i	Mass of part i (kg)
μ_i	Dynamic viscosity of fluid i ($kg.m^{-1}.s^{-1}$)
Nu	Nusselt number
η_{Meca}	Engine mechanical efficiency
P_{Pump}	Pump power (W)
Pr	Prandtl number
Φ_{Losses}	Combustion chamber thermal losses (W)
$\Phi_{X \to Y}$	Thermal losses between X and Y (W)
Q_m	Mass flow rate ($kg.s^{-1}$)
Re	Reynolds number
ρ_i	Density of i ($kg.m^{-3}$)
T_i	Temperature of i (K)
τ	Torque (Nm)
v	Fluid velocity ($m.s^{-1}$)
V_{ICE}	Engine displacement (cm^3)
ω_{ICE}	Engine speed (rpm)

Abbreviations

ICE	Internal Combustion Engine
NEDC	New European Driving Cycle
SI	Spark Ignition

Subscripts

i	Node i
w	Water

INTRODUCTION

For several years, some ecologic and economic considerations led to the development of innovative propulsion concepts as hybrid propulsion systems. The difficulty of that type of propulsion conception is not only the design of its different parts (ICE, electric motor, transmissions systems, etc.) but the development of control strategies allowing an optimization of each element operation too. These electrical-thermal complex architectures allow a significant reduction of the vehicle consumption thanks to a better use of the Internal Combustion Engine (ICE) which is operated at particular conditions of speed and torque involving best efficiencies.

Temperatures have significant effects on the ICE consumption. Especially, different energy losses like thermal transfers or friction losses decrease when the engine is warming-up. Thus, these losses are particularly high at ambient temperature during the first minutes after the start of the ICE (cold start) until it reaches its optimal thermal state (coolant and lubricating oil temperatures at about 90°C). Therefore, the ICE warm-up is a period of significant over-consumption [1].

Numerical simulations of hybrid vehicles are used as a help for such type of vehicle conception. However, the numerical model of the global vehicle becomes complex. In order to allow fast calculation, some physical phenomena are often neglected. Especially, only few studies were published about the simulation of the warm-up of the ICE used in a hybrid propulsion system [2, 3].

1 HYBRID VEHICLE

1.1 Objective

The hybrid systems objective is a global reduction of the vehicle consumption. The fuel economy is allowed because of a better use of the ICE, especially because of more flexibility on the engine operating point (torque and speed). For conventional vehicles, because of the direct link between the ICE and the wheels (through the gear box and transmission), the engine work is directly dependent on the vehicle driving conditions. Hybrid vehicles are equipped with a second propulsion system, generally using electric energy, that can be used both as a motor and a generator. This system, connected to the engine by various methods (serie-hybrid, parallel-hybrid), allow more flexibility on the ICE working conditions. The lighter systems (μ-hybrid) only use the electric motor to restart quickly the ICE allowing to stop it instead of using at idle speed. Heavier systems (mild-hybrid, full-hybrid) allow electric propulsion (help of the ICE, full electric operation) and an energy recovery of the ICE mechanical power (use of the electric motor as a generator to charge the battery).

1.2 Hybrid Control Strategy

The hybrid vehicle modeled here is a parallel full-hybrid one working with a ruled based control strategy. The main objective of this system is to use the ICE near or at its best efficiency. In particular, the electric motor is linked to enable more flexibility on its working conditions (speed and torque) and the best of them could therefore be chosen to reduce consumption. The work with high load and an middle speed is interesting because it provides the best mechanical efficiency with average friction torque. For a given speed, the best engine efficiency is indeed reached at high load. An optimal efficiency curve dependent on the engine speed is defined (see *Fig. 1*).

But, the difficulty is to keep this operating point whatever the driving conditions by two complementary behavior of the hybrid electric propulsion [4]:

- the work of the ICE at higher load while using the exceeding energy to charge the battery;

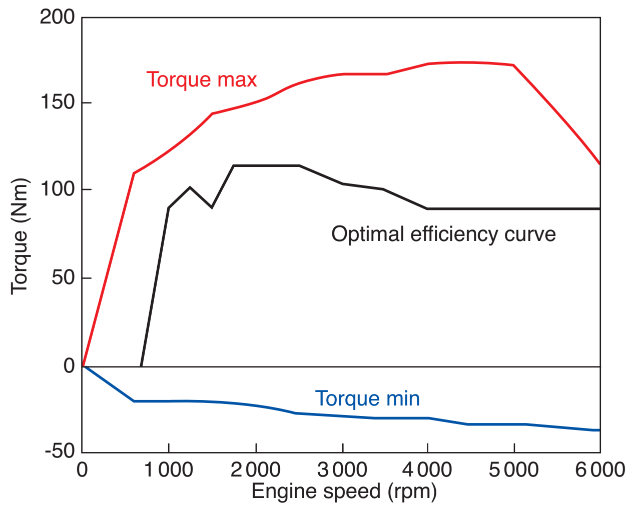

Figure 1

Example of the torque map of a 2L spark ignition engine.

- the work in full-electric mode when the electric motor performances are sufficient (generally for a speed under 50 km/h).

These two behaviors are used consecutively in order to keep the State of Charge (SOC) between two extreme values.

If the vehicle speed is too high (over 60 km/h), the batteries could still be charged but the ICE alone is assuring the vehicle propulsion at an operating point given by the vehicle movement.

2 DESCRIPTION OF THE MODEL

A nodal model [4-6] of the ICE has been developed. It has been presented in details in a previous publication [4]. The different parts of the engine are modeled by isothermal volumes connected with thermal resistances representing the thermal transfers between them.

The ICE metallic parts are divided in 5 elements *(Fig. 2)*:

- *the combustion chamber walls*: this part representing the cylinder liners is in direct contact with the combustion gas and thus receives a large amount of thermal energy during engine operation. It is cooled by the coolant by convection;
- *the upper and the lower engine blocks*: the engine block in this model represents most of the solid parts of the real ICE. It is divided in 2 elements in order to model the temperature gradient between the hottest parts that are close to the combustion chamber (engine head, parts of the engine block exchanging with the coolant) and the coolest parts (for example the oil tank). Because they are in reality a unique element, the lower and upper engine blocks exchange thermal energy by conduction;

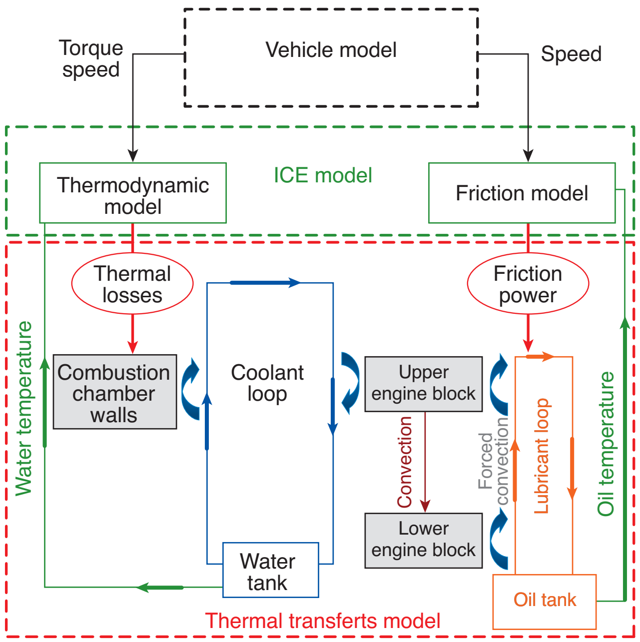

Figure 2

Internal combustion engine nodal modeling.

– *the water coolant loop*: the coolant allows forced convective thermal transfers with the hottest elements. It receives thermal energy from the hotter combustion chamber walls and transfers it to the upper engine block. In order to maintain the water in a liquid state, the engine radiator is used when its temperature exceeds 90°C. This element is not used during the warm-up period. When the engine is stopped, the forced convection becomes a natural convection between the metallic parts and the coolant;
– *the lubricant loop*: because of the oil flow trough all the solid elements to lubricate the moving parts (piston, crankshaft and camshaft), some forced convection between it and the engine blocks are included in the model. However, a large part of the received thermal energy comes from the power generated by the friction. Finally, the warmed oil returns to the oil tank where a natural convection with the lower engine block occurs.

For more details about the modeling of the thermal transfers (conduction, convection) occurring between the different elements, see reference [4].

3 SIMULATION RESULTS

The thermal transfer model is used for the simulation of the warm-up of a 2L spark ignition engine. The next step in this study is to validate the thermal transfer model before using it for the simulation of a hybrid vehicle. Because of the lack of publication about the warm-up of the ICE in the particular case of a hybrid vehicle, this validation was made by comparison with experimental results [1, 5].

3.1 Simulation of the ICE Warm-up on the NEDC Cycle

To evaluate the thermal transfer model performance and then to allow a comparison between the conventional and hybrid vehicle ICE warm-up, a simulation of the run of a 2L spark ignition engine was made on the NEDC cycle (New European Driving Cycle) *(Fig. 3)*.

Jarrier *et al.* [5] made a similar study with a real 1.9 L Diesel engine with a NEDC cycle (experiments and simulation). Because of the similarities of the cooling systems, lubricant loop and materials of both type of ICE (SI and Diesel), their global thermal behavior should be rather equivalent. So, a comparison between their results and the thermal transfer model results should be possible. It turns out that the simulation results are really similar to the experimental measurements done by Jarrier *et al.* [5]. In particular the magnitude of the different temperatures and the time needed to warm-up the ICE are really close. Thus, Jarrier *et al.* [5] measures validate the thermal model.

The warm-up of the different elements of the ICE model is shown in Figure 4. These results indicate the global thermal behavior of an ICE for a conventional use during an urban driving type cycle (speed never exceeding 50 km/h, frequent stops) represented by the 13 first minutes of the NEDC. In these conditions, the two engine fluids (water coolant and lubricating oil) have a similar temperature increasing rate (slightly faster for the coolant) and their optimal temperature is obtained after about 15 minutes. However, some studies show that the mean length of car driving is rather similar. Thus, it appears that cars are mostly

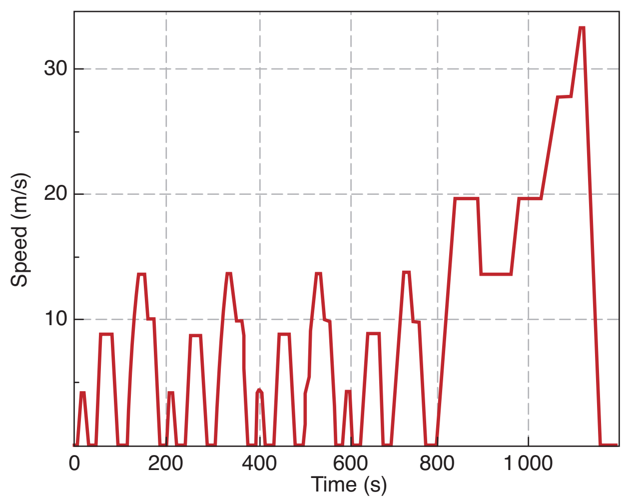

Figure 3

New European Driving Cycle.

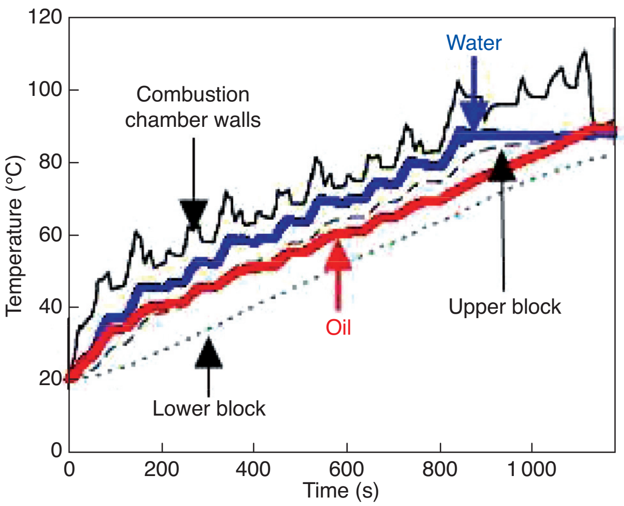

Figure 4

Simulation of the warm-up of an ICE from a conventional vehicle on the NEDC cycle.

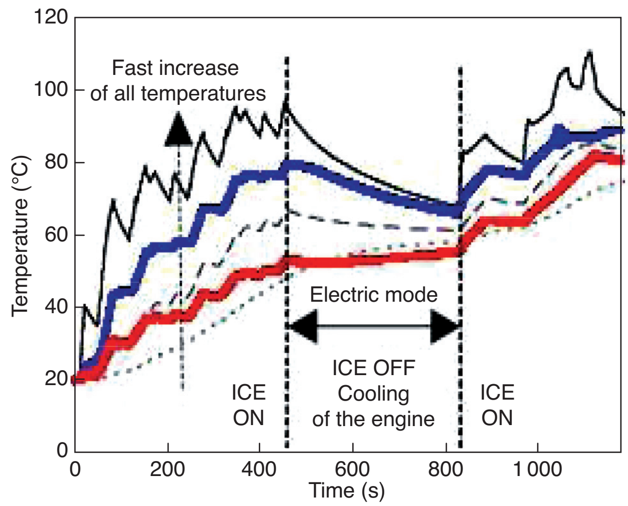

Figure 5

Simulation of the warm-up of an ICE from a parallel full-hybrid vehicle on the NEDC cycle.

used with an engine that is not at optimal temperature and the study of its warm-up becomes even more important.

One objective of this study is to investigate the impact of the hybrid association on this global thermal behavior.

3.2 Simulation Results for the Hybrid System

The new engine model is now used in an hybrid architecture. Because of the particularity of the ICE operation in such a vehicle, another thermal behavior is expected. Especially, the work with high load should impact strongly the coolant and oil warm-up. For the thermal behavior comparison between the conventional and hybrid propulsion systems, the initial charge of the battery is chosen at its minimal value in order to force the use of the Hybrid mode of the vehicle at the start of the NEDC cycle and obtain the warm-up of the vehicle during all the cycle.

The simulation results are presented in Figure 5. The first part of the NEDC cycle (450 first seconds) is done in hybrid mode while charging the battery by the engine. The two fluids reach their final temperature prior to the end of the NEDC cycle. The high load work increases the water warm-up speed. The coolant reaches almost its optimal temperature at the end of this period, it is faster than the conventional vehicle case. This result is logically explained by the direct contact of the coolant with the combustion chamber walls. The lubricant oil temperature, on the contrary, is less influenced. It is even lightly slower than for the conventional vehicle. The fact that the engine is regularly stopped, and thus there is not any friction energy produced during several seconds could be the cause of this phenomenon.

However, this study has been done with a particular driving cycle. There is a doubt about the impact of the cycle specificity (different speeds, transitional phases, accelerations, etc.) on the observed behavior.

So, the previous study was repeated for the different classic driving cycle *(cf. Appendix 1)* and in the case of different constant driving speed. The necessary time for the water to reach its optimal temperature (90°C) and for the lubricant to reach a sufficient temperature to have a significant reduction of the friction power (40°C) was evaluate for all driving conditions. The results are presented in Tables 1 and 2. The simulation for all driving cycles tends to confirm the fact that the hybrid use of the ICE allows a fast increase of the water temperature with almost a 50% reduction of the time needed to reach its maximum temperature in all the cases. Concerning the lubricant, the results seem to differ depending on the type of driving cycle. In most cases, the hybrid propulsion system tends to slow down the lubricant warm-up. The regular stops of the ICE is likely the cause of this phenomenon, the more impacted cycles being those that present the more vehicle stops during the first minutes (1015, ECE-EUDC, NEDC). This observation is confirmed by the driving at constant speed where the vehicle never stops. A speed up of the oil temperature rate can even be observed.

These results are consistent with the experimental results obtained by Trapy and Damiral [1]. Their study about the evolution of the engine oil temperature rate and its dependence on torque and speed show indeed that the oil warm-up is strongly dependent on the ICE speed but is almost not influenced by the engine torque. A simulation of the oil warm-up using the thermal transfer model on the same conditions of Trapy and Damiral [1] experiments are presented

TABLE 1

Water and oil temperature increasing rate for different driving cycles (*X* = temperature not reached, conventional vehicle: blue, hybrid vehicle: red)

	1 015	ECE-EUDC	FTP	HWFET	NEDC	SC03	UDDS
t (s) for	X	698.3	458.2	311.5	662.2	X	458.4
T_{Water} = 90°C	X	387.7	258.9	176.2	347.7	333.2	258.8
t (s) for	276,6	190.6	210.5	122.9	159.2	218.2	207.7
T_{Oil} = 40°C	465,1	296.1	223.4	116.6	256.1	292.2	225.1

TABLE 2

Water and oil temperature increasing rate for different driving speeds

	15 km/h	32 km/h	50 km/h	70 km/h	100 km/h
t (s) for T_{Water} = 90°C	428.6	255.9	262.6	223.9	145.9
	174.4	130.5	139.6	132.6	108.2
t (s) for T_{Oil} = 40°C	127.6	95.2	93	83.5	67.3
	108.2	86.1	82.7	76.3	61.3

in Figures 6 and 7. This is explained by the fact that the lubricant oil draws most of its energy from the viscous friction occurring in the ICE that is strongly dependent on its rotating speed.

However, this speed is unchanged for the control strategy *(cf. Sect. 1.2)* used in the hybrid vehicle model.

Moreover, Figure 5 shows that the engine is cooled down during the full-electric mode occurring after the hybrid mode (between 450 s and 850 s). It appears that the duration of this period is insufficient to obtain an important decrease of the ICE global temperature. Thus the next engine start is not a new cold start. Another observation is the homogenization of the temperatures of the engine parts. The coolest elements like the lower engine block and the oil receives some energy from the hottest like the combustion chamber walls and the coolant. Finally, it seems that the global warm-up time of the ICE is not significantly increased because of the particular use of the engine in this hybrid configuration.

This paper does not present the effect of the hybrid propulsion on the pollutant emissions and their conversion in the catalyst. This part should not be neglected and will probably be included in further studies. The work with high load of the engine should indeed have a significant effect on the thermal energy released in the exhaust gas that are used to warm-up the catalyst. Lescot *et al*. [2] studied the effect of an optimization-based strategy (applied to a parallel hybrid vehicle) on the vehicle consumption and on the time that should be needed to warm-up the catalyst. Their results shows that a better use of the ICE during the hybrid vehicle operation reduces the catalytic converter warm-up time and thus should reduce pollutant emissions. The comparison of the energy released with time in the exhaust gas is done between the hybrid and the conventional vehicle. The results presented in Figure 8 shows that the hybrid mode generates more exhaust thermal energy during the first minutes of the NEDC cycle.

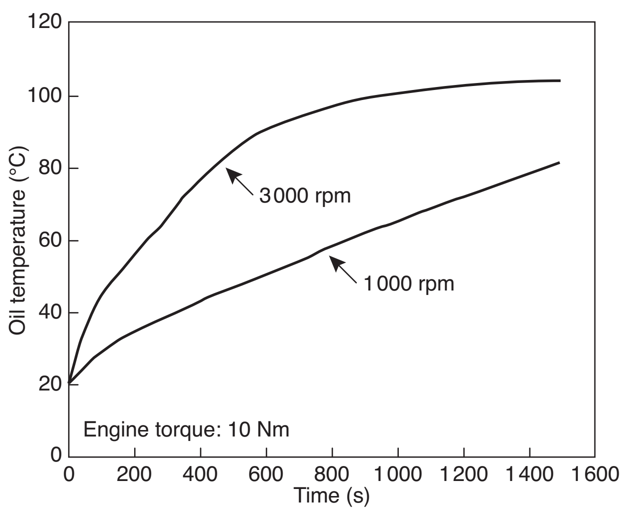

Figure 6

Influence of the engine speed on the oil warm-up (constant speed and torque).

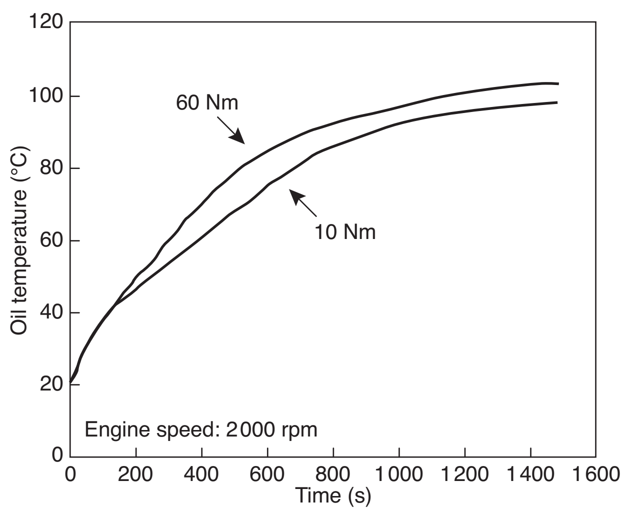

Figure 7

Influence of the engine load on the oil warm-up (constant speed and torque).

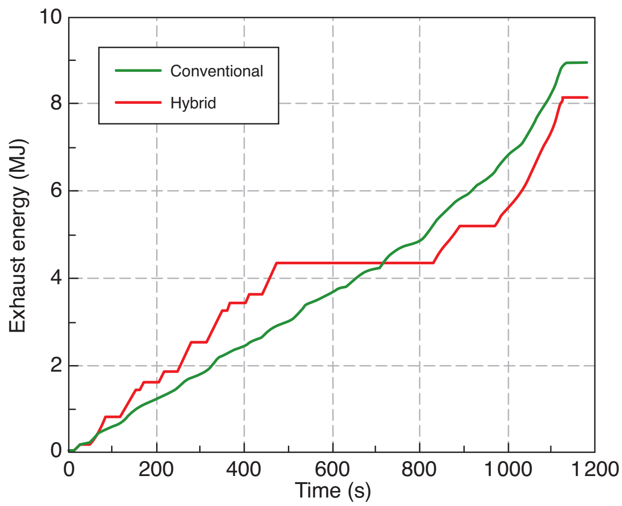

Figure 8

Energy released in the exhaust gas during the NEDC cycle.

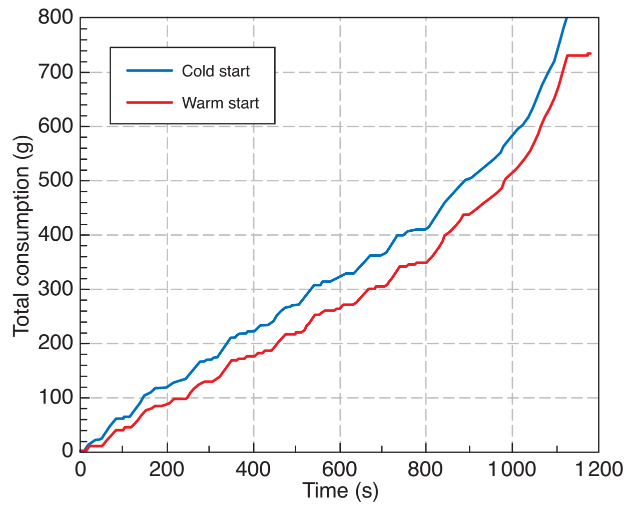

Figure 9

Effect of the cold start on the engine total consumption (conventional vehicle).

3.3 Effect on the Fuel Consumption

The aim of this section is to study the behavior of the ICE consumption during its warm-up in a parallel full-hybrid vehicle. Before, it is interesting to analyze the engine warm-up for a conventional vehicle. A simulation of the warm-up of the ICE on a NEDC cycle was done. The results are presented in the following figures. The total consumption of a vehicle is represented in the 2 following cases:

– an engine starting at ambient temperature (20°C);

– a start with a fully warmed engine.

The simulation predicts (for a NEDC cycle) an overconsumption of about 10-12%. These results are consistent with those obtained by Kunze *et al*. [7] for the same driving cycle. Thus, this result tends to validate the thermal transfers model. It also indicates that the effect of the losses increase for a cold thermal state is not negligible. The results obtained on the NEDC cycle are presented in Figures 9 and 10.

It is also interesting to study the evolution of the engine overconsumption through time. The instantaneous overconsumption is calculated by:

$$InstOvercons = \frac{InstCons_{cold}(t) - InstCons_{hot}(t)}{InstCons_{hot}(t)} \tag{1}$$

The analysis of the instantaneous overconsumption of the conventional vehicle on a NEDC cycle shows that most of the overconsumption is obtained during the first minutes of the cycle. It is explained by the fact that the engine is at its coldest thermal state. It appears that the mean overconsumption of a conventional vehicle is higher when the vehicle is exclusively used during short trips (what is mostly the case in urban conditions).

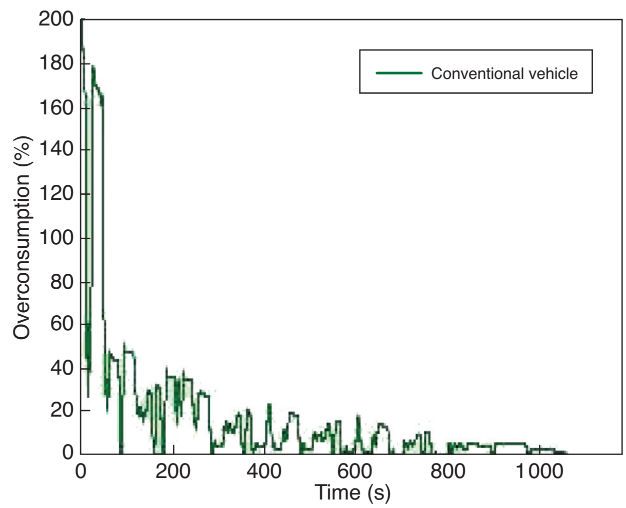

Figure 10

ICE instantaneous overconsumption for a conventional vehicle.

3.4 Hybrid Vehicle Fuel Overconsumption

The second part of the study is dedicated to the particular case of the hybrid vehicle. Like the previous study, 2 simulations (cold-start/warm-start) are made on the NEDC cycle. The results are presented in Figure 11. For this study, the initial battery charge is once more at its minimal value.

Firstly, the total overconsumption of this type of vehicle is also about 10% at the end of the NEDC cycle, which is equivalent of the conventional vehicle overconsumption (due to the thermal phenomena). However, the overconsumption behavior seems to differ between the two types of propulsion.

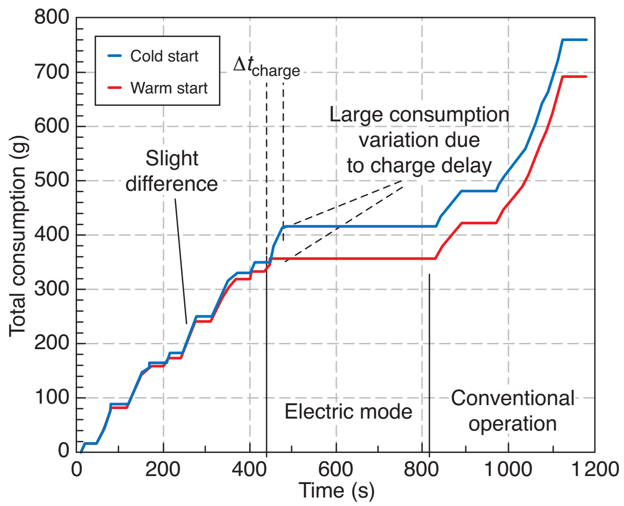

Figure 11

Effect of the cold start on the engine total consumption (hybrid vehicle).

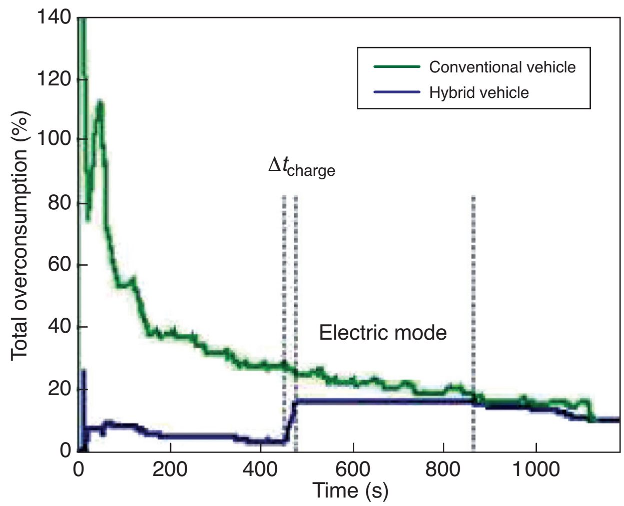

Figure 12

Total overconsumption over time on a NEDC cycle after a cold start (conventional and hybrid vehicle).

It appears especially that the overconsumption of the vehicle is really light at the beginning of the cycle when the ICE is used for both the vehicle propulsion and the battery charge. The main overconsumption due to the thermal effects appears because of a charging delay of the battery $\Delta t_{\mathrm{Charge}}$.

This last result is more visible with the draw of the engine global overconsumption. The total overconsumption is defined as follows:

$$TotCons\ (t) = \int_0^t InstCons(t)dt \qquad (2)$$

$$TotOvercons = \frac{TotCons_{cold}(t) - TotCons_{hot}(t)}{TotCons_{hot}(t)} \qquad (3)$$

The total overconsumption, in contrary to the instantaneous consumption, does not vanish when the ICE reaches its optimal temperature, the gap between the two consumption (after a cold and a warm start) becoming constant. However, the total consumption still increases and thus the total overconsumption decreases through time. The study of the total overconsumption gives therefore some information in order to evaluate the impact of the duration of the vehicle trip.

The total overconsumption of both types of vehicle (conventional and hybrid) is presented in Figure 12.

The plot of the total overconsumption of both vehicles confirms the effect of the hybrid mode on the ICE overconsumption due to its cold state during the first minutes of the NEDC cycle. It shows indeed that the total overconsumption of a conventional vehicle can be very high (over 50% during the first 100 seconds of the cycle).

The plot of the total overconsumption of both vehicles confirms the effect of the hybrid mode on the ICE overconsumption due to its cold state during the first minutes of the NEDC cycle. It shows indeed that the total overconsumption of a conventional vehicle can be very high (over 50% during the first 100 seconds of the cycle). For the hybrid vehicle, the total overconsumption never exceeds 20%. During the first minutes, it is lower than 10%. Because of the delay of the battery charge period, a rapid increase of the total overconsumption is observed for a few seconds (also observed in *Fig. 11*). Because of the duration of the NEDC cycle, the ICE reaches its optimal thermal state for both cases and the total overconsumption decreases during the last minutes. However, during the electric mode the ICE overconsumption of the hybrid vehicle stays constant. This explains why both total overconsumptions are equivalent at the end of the cycle.

So, it appears that the hybrid work has a beneficial effect during short trips by reducing the overconsumption caused by the cold start during the first minutes after the vehicle start. For longer use of the vehicle, the effect of a cold thermal state of the ICE on the consumption seems to be the same for both vehicles (about 10%).

This result could be explained by the peculiar use of the ICE during the hybrid mode. The engine is indeed used at high load (on the optimal efficiency curve) with recuperation of a part of the exceeding mechanical energy to charge the battery. It seems that this ICE operation not only increases its efficiency but also has a positive effect during the warm-up period. So, some simulations of the use of the ICE at different loads and speeds are done with an analysis of the total overconsumption resulting from the cold start. The results (presented in *Fig. 13*) confirm the benefic impact of the high torque on the ICE consumption after a cold start. These results are comparable to these presented in Figure 12.

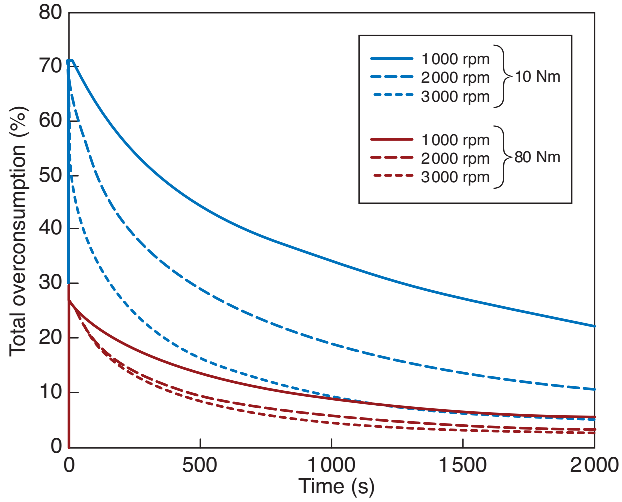

Figure 13

Total overconsumption of a 2L spark ignition engine depending on load and speed.

These results are logically explained by a simple mathematic example. An engine is considered during its work at a fixed speed. Thus, the friction torque for this speed is constant (for more simplification, only the friction losses are taken into consideration here). We assume (for this example) that for this fixed speed, the friction losses are two times higher (value chosen in according to friction torque simulation with dependence on its temperature) when the engine start at ambient temperature. The engine work with 2 different final torques (measured on the crankshaft), one equivalent and one largely superior to the friction torque. For the calculation the following values are chosen:

– $\tau_{Friction}$ = 10 Nm (value quantitatively coherent for optimal oil temperature and usual engine speed);
– $\tau_{Friction\_ambiant} = 2\tau_{Friction}$ = 20 Nm;
– $\tau_{Crankshaft\_1} = \tau_{Friction}$ = 10 Nm;
– $\tau_{Crankshaft\_2} = 8\tau_{Friction}$ = 80 Nm (corresponding to the mean engine torque for hybrid mode operation in the hybrid vehicle model).

The ICE mechanical efficiency is calculated as follows:

$$\eta_{Meca} = \frac{\tau_{Crankshaft}}{\tau_{Crankshaft} + \tau_{Friction}} \tag{4}$$

The calculation gives the variation of mechanical efficiency for both final torques due to the friction increase at ambient temperature. The results are presented in Table 3. This simple example gives an illustration to the fact that the different supplementary losses generated by the cold thermal states of the engine could be easily compensated by using the ICE with high load. This phenomenon is occurring for the modeled control strategy used in the simulation.

TABLE 3

Mechanical efficiency reduction due to ambient oil temperature for two different loads

	Optimal temperature (90°C)	Ambiant temperature (20°C)	η_{Meca} reduction
τ_1 = 10 Nm	0.50	0.33	– 33.33
τ_2 = 80 Nm	0.89	0.80	– 10.00

3.5 Cooling of the ICE During Electric Mode

The modeled hybrid vehicle is a full-hybrid one. So the electric system is powerful enough to assure alone the vehicle propulsion independently for relatively low speed and torque (generally for urban conditions). Thus, the ICE is regularly stopped during several minutes and therefore, its global temperature decrease during these periods (see *Fig. 5)*. It is interesting to evaluate the supplementary losses generated by the need for the engine to warm-up again to reach its optimal temperature after every new start. Two simulations of the run of the modeled hybrid vehicle on several successive NEDC cycle with a start at optimal temperature are done. In the first case the engine is maintained at its maximal mean temperature (90°C). The second simulation uses the thermal transfer model and the engine is therefore cooled when stopped.

The comparison of both simulations results (presented in *Fig. 14*) shows that the overconsumption resulting from the regular engine stops is negligible (less than 1% after two NEDC). The analysis of the water and oil temperature variations through time shows that the duration of the full-electric mode allowed by the battery is too low to obtain a significant temperature decrease.

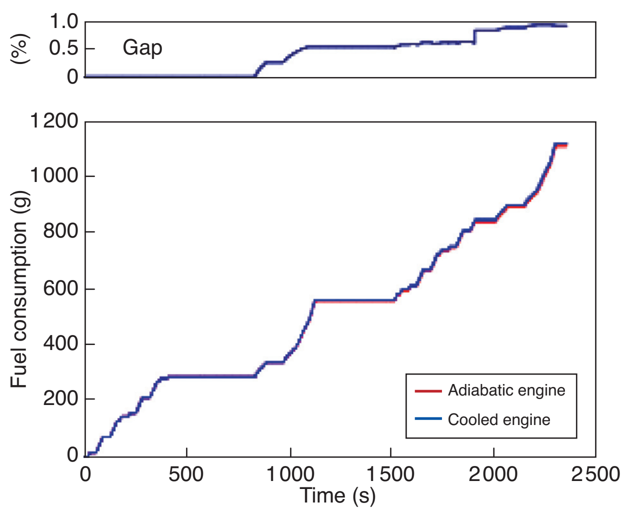

Figure 14

Overconsumption induced by the cooling of the engine during stop.

Because of the lack of information about the thermal transfers to the ambient air occurring under the vehicle hood (responsible of the engine cooling), this calculation has been done for several transfer coefficient value. The results presented previously are still observable even for really high thermal transfers to the ambient.

4 EXAMPLE OF USING: RESEARCH ON THE WARM-UP OPTIMISATION

4.1 Water and Oil Temperature Variation

The thermal behavior of water and oil dependent on the engine load and speed were determined *(cf. Sect. 3.2)*. However, this previous study is insufficient in order to find a better control strategy to optimize engine warm-up.

A parametric study has been done to evaluate quantitatively the effect of engine speed and torque on oil and water temperature increasing rate. After a few simulations of the ICE warm-up at different speeds and torques, the times needed to obtain characteristic temperatures values have been recorded:

- T_{Water} = 90°C: maximum water temperature;
- T_{Oil} = 80°C: oil sufficiently warmed to be considered at optimal temperature;
- T_{Oil} = 40°C: temperature allowing non negligible friction reduction after the cold start.

The results are presented in Table 4.

The results show that the water temperature is dependent on both the torque and the speed. The thermal energy received by the water comes indeed directly from the amount of fuel burned in the engine that changes with the operating point. It can be observed in Table 4 that the time to warm-up the water can significantly be reduced for both chosen torques if the engine is used with relatively high speed (2 000-2 500 rpm).

These results confirm the thermal behavior of oil with a high dependence on the engine speed. It is indeed possible to reduce the oil warm-up by a half if the engine is used with a speed of about 2 000 rpm instead of 1 000 rpm. The oil warm-up time can even become really short (75-80 seconds to reach 40°C at 2 500 rpm). An increase of the engine load has only a small effect on the oil temperature increase.

So, in order to reduce the engine warm-up duration it seems to be interesting to increase the engine speed (for the chosen control strategy, the engine already work with high load). However, speed also means friction and increasing it should results in supplementary losses. To evaluate them, the engine mean efficiencies (calculate from the engine start to the time needed to obtain the chosen characteristic fluid temperature) were computed for all the cases presented in Table 4. These efficiencies are given in Table 5.

TABLE 4

Oil and water temperature increasing rates for different speeds (rpm)

10 Nm	1 000	1 500	2 000	2 500
t_1 (s) for T_{Water} = 90°C	2 064	805	391	233
t_2 (s) for T_{Oil} = 40°C	314	169	110	80
t_3 (s) for T_{Oil} = 80°C	1 883	892	581	429

80 Nm	1 000	1 500	2 000	2 500
t_1 (s) for T_{Water} = 90°C	462	269	181	117
t_2 (s) for T_{Oil} = 40°C	248	150	102	75
t_3 (s) for T_{Oil} = 80°C	933	619	458	342

TABLE 5

Effect on the efficiency for different speeds (rpm)

10 Nm	1 000	1 500	2 000	2 500
t_1 (s) for T_{Water} = 90°C	7.60	7.37	6.97	6.58
t_2 (s) for T_{Oil} = 40°C	6.11	6.28	6.20	5.98
t_3 (s) for T_{Oil} = 80°C	7.53	7.47	7.29	7.01

80 Nm	1 000	1 500	2 000	2 500
t_1 (s) for T_{Water} = 90°C	23.80	24.40	24.20	23.00
t_2 (s) for T_{Oil} = 40°C	23.20	23.70	23.50	22.40
t_3 (s) for T_{Oil} = 80°C	24.60	25.50	25.50	24.60

The computed mean efficiencies show that the use with higher speed is not interesting at relatively low load because of a relatively important reduction of the engine efficiency with speed:

$$0.0737_{(1\,500\text{ rpm})} \rightarrow 0.0687_{(2\,000\text{ rpm})} = -5.7\%$$

On the contrary, the efficiency degradation through speed is lower at high load:

$$0.244_{(1\,500\text{ rpm})} \rightarrow 0.242_{(2\,000\text{ rpm})} = -0.8\%$$

Thus, one way to improve the hybrid vehicle consumption during the engine warm-up could be to increase the ICE speed during the high load operations of the hybrid mode. The warm-up duration could therefore be reduced with only a little decrease of the engine efficiency. Through time, the little overconsumption generated by the speed increase (and the induced supplementary friction losses) should be inferior to these occurring for a longer warm-up period. Of course, this study must be developed to verify those expectations given by the current thermal transfers model.

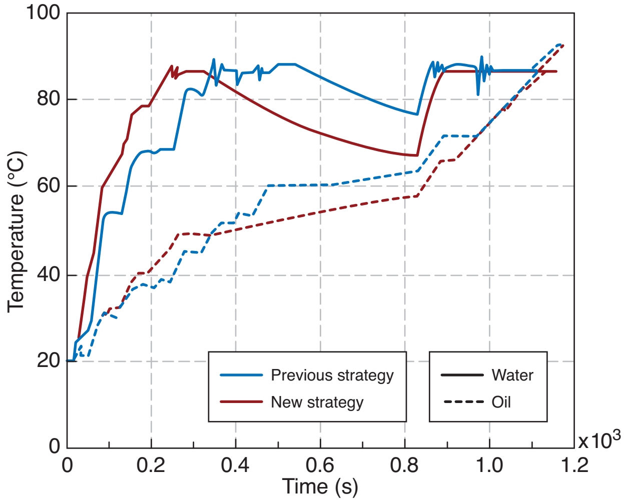

Figure 15

Evolution of water and oil temperature if the engine stops are prevented during the first charge.

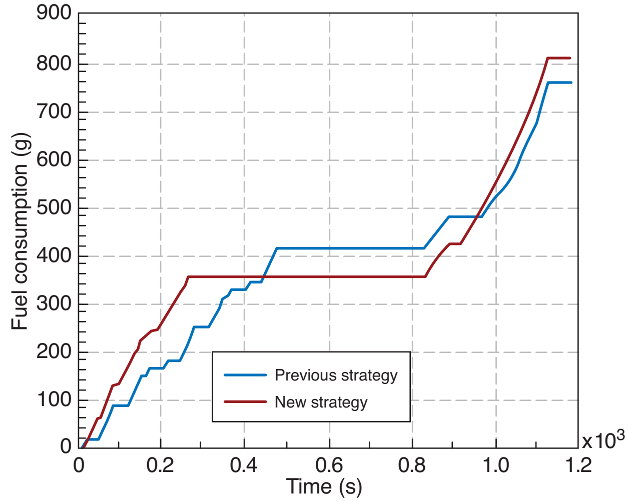

Figure 16

Evolution of fuel consumption if the engine stops are prevented during the first charge.

4.2 Effect of the Regular Engine Stops

The previous Section 3.2 allows the observation of the effect of the several engine stops during a classic driving cycle (NEDC) on the oil temperature increasing rate. Even if the operation at high load allows a little increase of the oil temperature increasing rate, the regular engine stop when the vehicle do not run leads to a global reduction of the oil warm-up rate *(cf. Tab. 1, 2)*. Thus, the thermal behavior of the hybrid vehicle could probably be optimized by reducing the number of engine stops during the first minutes of the vehicle use. In particular, the impact of the prevention of engine stops during the first battery charge in hybrid mode of a hybrid vehicle run was evaluated by the simulation on the NEDC cycle. The control strategy was modified to force the ICE to run at constant speed and torque (2 000 rpm, 80 Nm where the engine should be at a maximum efficiency) during the first battery charge if a stop of the vehicle was detected by the ECU (Engine Control Unit). The generated energy is not used to assure the vehicle propulsion but is redirected to the electric motor/generator to charge the battery. The oil and water temperature increasing rates, engine fuel consumption and battery SOC evolutions are presented in Figures 15, 16 and 17.

So, it appears that the prevention of engine stops does not improve significantly the water and oil warm-up duration. This control strategy modification allows at best the reduction of a few seconds to obtain the characteristic fluid temperatures during the first minutes of the cycle. However, the battery charge is significantly faster and thus the electric mode begins rapidly. The observation of the fuel consumption

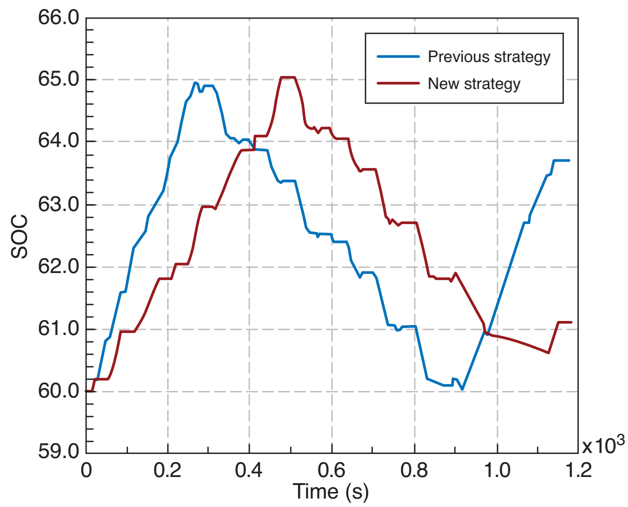

Figure 17

Evolution of battery SOC if the engine stops are prevented during the first charge.

shows that the total fuel consumed to simultaneously charge the battery and assure the propulsion is lower for the new strategy. However, the repercussion of this fast charge increasing rate is the earlier need to start the second battery charge in hybrid mode because of the constant length of the full-electric mode. At the end of the NEDC, it is hard to determine which solution is the best for fuel consumption reduction, the first one inducing more burned fuel, the second one inducing a lower battery charge at the end of the cycle.

Thus, it is not evident if the new strategy has really a benefic or negative effect on the engine warm-up for a parallel full-hybrid vehicle. The simulation of the vehicle run for different conditions (cycles, initial SOC) should give more information on this control strategy efficiency.

However, these results are already interesting in the case of a plug-in vehicle. In this case, the first part of the NEDC cycle is the more interesting because it corresponds to most of the drivers day to day displacements (short trips in the city). The first battery charge generates indeed less fuel consumption for the new strategy. The plug-in hybrid vehicle battery is recharged every time with the possibility to connect it to the power sector. So, for short trips in a city, because of the battery autonomy of several minutes (about 10 minutes for urban conditions), in most of the cases only one charge of the battery (hybrid mode) will be needed, and fuel can therefore be saved.

Finally the information given by the developed thermal transfer model could be used in order to test new strategies in order to improve the warm-up behavior of its engine. There are many applications for this model in order to find a way to reduce vehicle consumption.

CONCLUSION

A simulation of heat transfers within an ICE using the lumped capacitance method has been developed. The use of this model allows an observation of the different fluids and solid parts warm-up behaviors for all engine operating conditions.

The model is only partially validated yet. However, the comparison between the simulation results and experimental measurements made in other studies [1, 5] already shows good agreement between the model and reality. Moreover, the fact that all observed behaviors could be logically explained is also encouraging for the following studies. Of course, it is necessary not to ignore the precisions default induced by such a simulation and the model can still be developed and improved. Indeed, only the engine warm-up was studied yet. However, it is not the only part of the vehicle that is the place of high energy transfers. In particular, the thermal behavior of the electric motor and the batteries are an important concern too. Finally, the impact of an hybrid architecture on pollutant emissions, especially the warm-up of the different aftertreatment devices like the catalyst, could be studied.

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APPENDIX 1

A1 DRIVING CYCLES

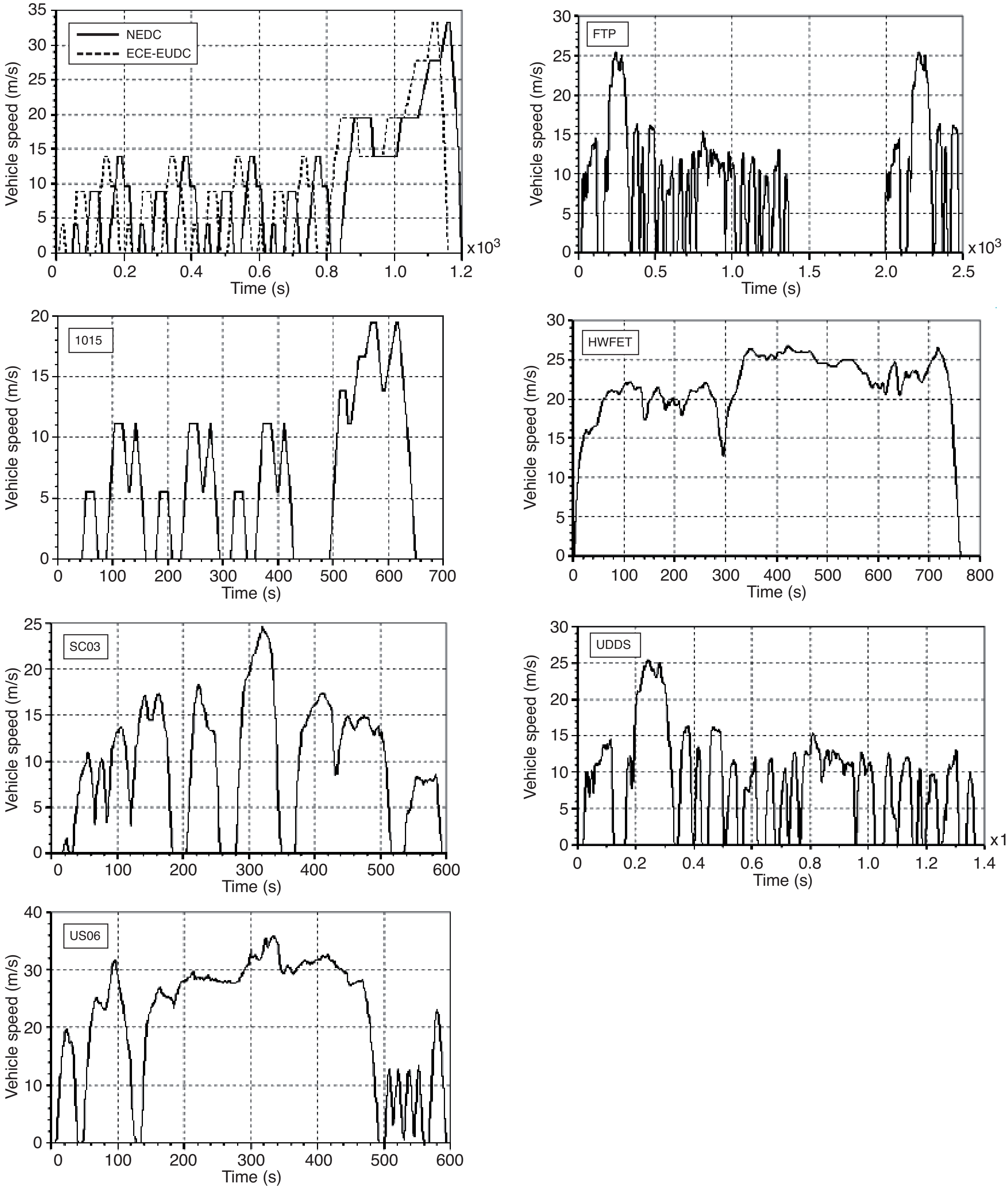

Figure A1

Common driving cycles.