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Comparative study of turbocharged diesel engine emissions during three different transient cycles

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SUMMARY

Starting from the 1980s (diesel-engined) vehicles have been tested for exhaust emissions, prior to type approval, using sophisticated standardized transient tests (Transient Cycles). These are usually characterized by long duration consisting of both speed and load changes under varying operating schedules. In the present work, a fast and, relatively, easy to apply approach was developed in order to be able to make a first approximation of the engine performance and emissions during a speed/torque vs time Transient Cycle. The procedure is based on a previous steady-state experimental investigation of the engine for the formulation of polynomial expressions of all interesting engine properties with respect to engine speed and torque. Correction coefficients are then applied, based on experiments conducted on the engine under study, to account for transient discrepancies. Using the developed algorithm, a comparative study was conducted for the European, American and the Worldwide heavy-duty Transient Cycles. It was revealed for the current engine that the European ETC, being the most aggressive and having the shortest idling period, is also the most demanding in terms of absolute emissions (g), particularly soot. At the same time, the importance of abrupt transients (primarily experienced during urban driving) on engine emissions was highlighted. A comparative analysis was also performed that detailed the individual technical and transient characteristics of each cycle. Copyright © 2009 John Wiley & Sons, Ltd.

KEY WORDS: heavy-duty diesel engine; transient operation; nitric oxide; soot; fuel consumption; Transient Cycle

1. INTRODUCTION

The turbocharged diesel engine is nowadays the most preferred prime mover in medium and medium–large units' applications (truck driving, land traction, ship propulsion, electrical generation). Moreover, it continuously increases its share in the highly competitive automotive market owing to its reliability that is combined with excellent fuel efficiency. Particularly its transient operation is of great importance in the everyday operating conditions of engines, being often linked with off-design

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(turbocharger lag) and consequently non-optimum performance; the latter is realized as slow response as well as overshoot in particulate, gaseous and noise emissions. Acknowledging these facts, various legislative Directives in the European Union, Japan and the U.S., have drawn the attention of automotive manufacturers and researchers all over the world to the transient operation of (diesel) engines, in the form of Transient Cycles Certification for new vehicles [1,2]. A Transient Cycle is usually characterized by long duration (up to 30 min) consisting of both speed and load changes under varying operating schedules simulating real driving conditions.

During the last decades, the modeling and the experimental investigation of the thermodynamic and gas dynamic processes in diesel engines have intensively supported the study of engine transient operation [3–8]. However, the vital issue of exhaust emissions as well as the overall engine and vehicle performance during a Transient Cycle—although of primary concern to engine manufacturers—has not been investigated adequately but rather segmentally owing to the following two major issues:

(a) As regards the modeling approach, incorporation of exhaust emission predictions into transient simulation codes, via two- or better still multizone models, would lead to high or even prohibitive computational times for the thousands of engine cycles needed to be simulated during a Transient Cycle.
(b) As regards the experimental approach, Transient Cycles require highly complicated, sophisticated and costly experimental facilities to be accurately reproduced such as the Constant Volume Sampling system; this fact has limited the availability of transient experimental investigations and results primarily to individual load acceptances or speed accelerations of the order of a few seconds rather than investigation of the engine performance during the whole Transient Cycle [8].

The aim of this paper is to fill an apparent gap in the literature by applying a fast and, relatively easy in terms of cost and laboratory reproduction, evaluation of the performance and emissions (for the present study these include nitrogen monoxide NO and soot) of a heavy-duty diesel engine during a Transient Cycle. The approach, which lies between simulation and experiment, is based on a quasi-steady experimental mapping of the engine in hand, applying suitable correction coefficients to account for transient discrepancies. By so doing, the above-mentioned obstacles for the reproduction of a Transient Cycle, that is, extremely high computational time or costly and sophisticated experimental facilities can be overcome. Similar engine mapping-based approaches have been developed in the past following in general the context of quasi-linear modeling [9]. Nonetheless, as far as the authors are concerned, none of these earlier approaches has ever been employed for the analysis of engine performance during a Transient Cycle.

The developed algorithm is applied to a diesel engine running on the European (ETC), American (Federal Test Procedure, FTP) and Worldwide (WHTC) Transient Cycles used for the certification of heavy-duty diesel engines but can be easily employed to any other torque/speed vs time transient schedule (e.g. nonroad mobile engines). Unlike past research, emphasis in this work is given on the most influential, in terms of emissions, transient schedules (load acceptances and accelerations) of each Cycle, which are studied and detailed.

2. BACKGROUND ON TRANSIENT CYCLES

A Transient Test Cycle is a sequence of test points each with a defined vehicle speed to be followed by the vehicle under study or with a defined rotational speed/torque to be followed by the engine under transient conditions. These test points are divided in time steps, mostly seconds, during which acceleration is assumed constant. Such standardization is necessary as it makes it possible to compare different vehicles/engines that fulfill the same operation. In order for the exhaust emission measurements to be representative of real engine operation, Transient Test Cycles incorporate some or all of the following driving conditions:

- cold and hot starting,
- frequent accelerations and decelerations,

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- changes of load,
- idling conditions typical of urban driving,
- sub-urban or rural driving schedule, and
- motorway driving.

By applying a Transient Cycle for the testing of new vehicles, the complete engine operating range is tested and not just the maximum power or torque operating points. Moreover, the serious discrepancies that are experienced during abrupt transients are taken into account. It should, however, be pointed out that the primary objective of a Transient Cycle procedure is to establish the *total* amount of exhaust emissions rather than indicate the specific parts or conditions under which these emissions are produced [1,2,8].

2.1. European Transient Cycle

The ETC has been introduced for emission certification of heavy-duty diesel engines in Europe starting in the year 2000 (EC Directive 1999/96/EC) [10]. Different driving conditions are represented by three parts (each of 600 s duration) of the Cycle, incorporating urban, rural and motorway driving as well as motoring sections.

- Part one represents city driving corresponding to a maximum speed of $50\,\mathrm{km\,h^{-1}}$, frequent starts, stops, and idling.
- Part two is rural driving starting with a steep acceleration segment; the corresponding average speed of the vehicle is about $72\,\mathrm{km\,h^{-1}}$.
- Part three is motorway driving with corresponding average vehicle speed of about $88\,\mathrm{km\,h^{-1}}$.

The engine under study needs first to be mapped for determining the speed vs torque curve. Figure 1 (lower two sub-diagrams) illustrates normalized engine speed and normalized engine torque vs time for the ETC Cycle; for the specific engine under test, speed is de-normalized using the following equation [10]:

$$\text{Actual speed} = \frac{\%\ \text{speed (reference speed} - \text{idle speed)}}{100} + \text{idle speed} \quad (1)$$

with the reference speed N_{ref} corresponding to the 100% speed values specified in the engine dynamometer schedule, defined as follows:

$$N_{\mathrm{ref}} = N_{\mathrm{lo}} + 95\%\ (N_{\mathrm{hi}} - N_{\mathrm{lo}}) \quad (2)$$

with N_{hi} the highest engine speed, where 70% of the declared (by the manufacturer) maximum power occurs and N_{lo} the lowest engine speed, where 50% of the declared maximum power occurs.

Similarly, engine torque is de-normalized to the maximum torque at the respective rotational speed using the following equation:

$$\text{Actual torque} = \frac{\%\ \text{torque} \times (\text{max. torque})}{100} \quad (3)$$

with the maximum torque value found from the respective engine mapping curve.

2.2. American heavy-duty Transient Cycle (FTP)

The FTP heavy-duty Transient Cycle is currently used for emission testing of heavy-duty on-road engines in the U.S. The equivalent average speed is about $30\,\mathrm{km\,h^{-1}}$ and the equivalent distance travelled is 10.3 km for a running period of 20 min.

This Transient Cycle consists of four phases (Figure 1—middle two sub-diagrams):

1. NYNF (New York Non-Freeway) phase typical of light urban traffic with frequent starts and stops.
2. LANF (Los Angeles Non-Freeway) phase typical of crowded urban traffic with few stops.
3. LAFY (Los Angeles Freeway) phase simulating crowded expressway traffic in Los Angeles.
4. NYNF phase.

A de-normalization procedure similar to that described by Equations (1)–(3) applies here too [2].

2.3. Worldwide harmonized Transient Cycle (WHTC)

At its 34th session in June 1997, The UNECE Group of Experts on Pollution and Energy (GRPE) mandated the *ad hoc* group WHDC with the development of a 'Worldwide harmonized Heavy-duty Certification' procedure [11]. The objective of the research program was to develop a worldwide harmonized engine Test Cycle for the

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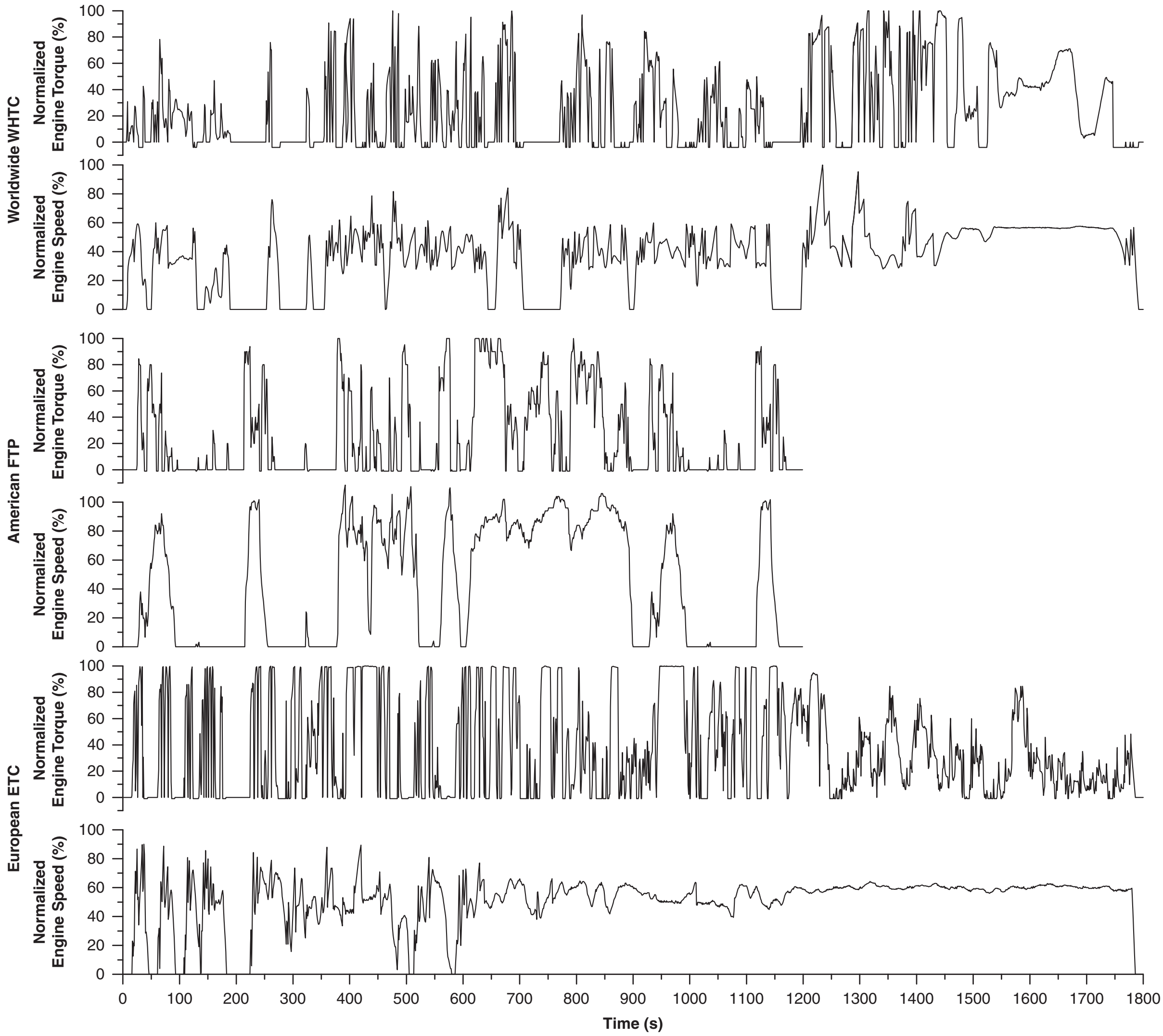

Figure 1. Normalized speed and torque vs time for the European ETC, American FTP and Worldwide WHTC Transient Cycles.

emissions certification procedure of heavy-duty vehicles/engines that would

- become a uniform global basis for engine certification regarding exhaust emissions;
- be representative of worldwide real-life heavy-duty engine operation;
- give the highest potential for the control of real-life emissions;
- be applicable in the future to state-of-the-art technology; and
- match emissions in relative terms for accurate ranking of different engines/technologies.

In order to proceed to the development of the worldwide harmonized engine Test Cycle, the United Nations research group conducted a collection and analysis of driving behavior data as well as a statistical investigation of heavy-duty vehicle usage in different regions of the world. From this database, a representative Worldwide Transient Vehicle Cycle (WTVC) of 1800 s

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duration and 40 km h^{-1} average speed has been derived, as well as a normalized engine speed and torque vs time Transient Cycle for heavy-duty diesel engines illustrated in the upper two sub-diagrams of Figure 1.

3. SIMULATION

3.1. General procedure

For the estimation of the performance and emissions during the Transient Cycle, the procedure illustrated schematically in Figure 2 is adopted. A detailed experimental investigation of the engine in hand is initially undertaken for the (steady–steady) mapping of the engine operation; the more the discrete engine speed points taken into account, the better it is for the accomplishment of the engine mapping. It is also important that measurements at very low engine loads are available for a more accurate formulation of engine mapping, as as will be discussed later, various load changes in the Cycle commence from a very low or even zero loading. For each engine rotational speed, a third or fourth order polynomial is formulated for every interesting property with respect to engine torque (recall that a heavy-duty engine Transient Cycle is expressed in terms of engine speed and torque vs time). For the current investigation, the engine properties under consideration are nitric oxide (NO), soot, fueling (hence carbon dioxide emissions) and power, although any other property can be easily formulated with the above-mentioned procedure. NO and soot emissions were chosen for the present analysis owing to the fact that a detailed set of experimental measurements for these two pollutants was available for the engine under study under both steady-state and transient conditions. In the analysis that follows, it is not, by any

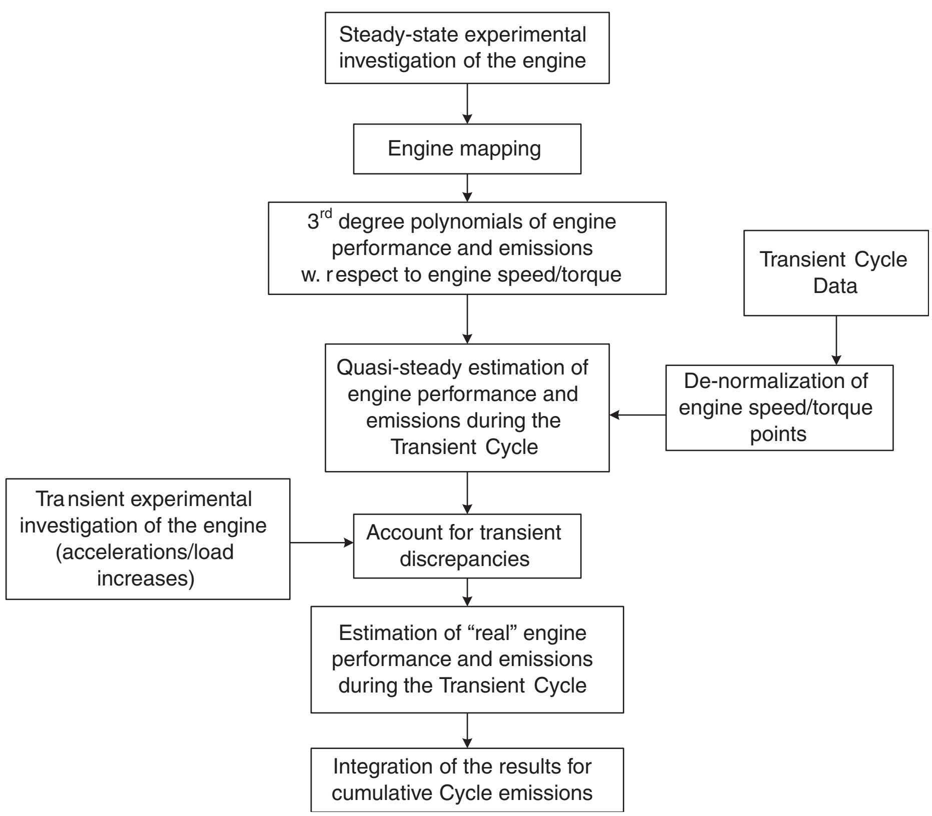

Figure 2. Block diagram of applied procedure.

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Table I. Data of the engine used for the analysis.

Engine type	In-line, six-cylinder, turbocharged, heavy-duty, diesel engine
Speed range	800–1500 rpm
Bore/Stroke	140 mm/180 mm
Compression ratio	17.7:1
Injection system	Mechanical, 6-cylinder fuel injection pump
Maximum power	320 HP (236 kW) @ 1500 rpm
Maximum torque	1520 Nm @ 1250 rpm

means, implied that the NO or soot values computed should be used as surrogates for the legislated NO_x and particulate matter.

The main technical characteristics of the engine under study, which has no after-treatment control, are given in Table I. For each Transient Cycle operating point, that is, at each 'second' of the cycle, a linear interpolation is performed in order to calculate the actual emissions for the particular (de-normalized) engine rotational speed and torque; artificial neural network [12] could be used as an alternative technique.

3.2. Transient discrepancies

As it has long been established, turbocharger lag is the most notable off-design feature of diesel engine transient operation that significantly differentiates the torque pattern from the respective steady–steady conditions. It is caused because although the fuel pump responds rapidly to the increased fueling demand after a load or speed increase, the turbocharger compressor air-supply cannot match this higher fuel-flow instantly, but only after a number of engine cycles owing to the inertia of the whole system as there is no mechanical connection between engine crankshaft and turbocharger. The above phenomenon is enhanced by the unfavorable turbocharger compressor characteristics at low loads and speeds. As a result of this slow reaction, the relative air–fuel ratio during the early cycles of a transient event assumes very low values (even lower than stoichiometric), thus deteriorating combustion and leading to slow engine (torque and speed) response, long recovery period and overshoot in particulate, gaseous and noise emissions. On the other hand, the high fuel–air equivalence ratios experienced after a speed or load increase transient event produce high combustion temperatures, which favor NO and soot formation, with the latter being identified as black smoke coming out of the exhaust pipe [6,8].

In order to account for the above serious transient discrepancies but at the same time not to deviate from the initial goal of a simple and fast algorithm, correction coefficients were applied to the steady–steady emissions based on experience gathered from an extended set of individual transient tests (various load acceptances and accelerations) conducted on the specific engine at the authors' laboratory. These correction coefficients (different values for soot and NO, with the soot constant found to have a value one order of magni tude higher than its NO counterpart) increase the instantaneous emissions at each operating point (second) in the cycle based on the specific load and/ or speed increase encountered from the previous to the current operating point (second), that is,

$$\begin{aligned}\text{Transient emissions} &\\ &= \text{Steady-steady emissions}\\ &\quad\times\{1+[c_{\text{load}}(\text{load-increase})]\\ &\quad\times[c_{\text{speed}}(\text{speed-increase})]\}\end{aligned} \tag{4a}$$

Initially, different correction coefficients were assumed for load and speed increases as described by Equation (4a). However, owing to the narrow speed range of the engine in hand, the actual engine speed deviation during all Transient Cycles was very short (cf. Figures 3–5 later in the text), the absolute engine speeds all very low, and the respective accelerations almost insignificant (cf. third and fourth column of Table III later in the text). Hence, for the current engine, it is *only* the load changes that actually contribute to the transient emission increase. Moreover, as all the load steps during the Cycles examined commence from low initial engine speeds, the well-known fact that load changes commencing from high speeds are less severe than the ones commencing from lower engine speeds (owing to higher initial turbocharger boost pressure) was not taken into account for the formulation of Equation (4a). From the elaboration of the experimental transient emissions for various load and speed increases, it was

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found that for the current engine configuration the transient emissions increase could be very effectively accounted for by applying the following simpler equation

$$\text{Transient emissions} = \text{steady-steady emissions} \times \{1 + c_{\text{load}}(\text{load-increase})(\text{speed-increase})\} \quad (4b)$$

A slightly different approach was followed by Ericson *et al*. [13], who applied correction coefficients based on air–fuel ratio variations during transients.

The following two assumptions are also valid for the analysis: (a) during the motoring segments of the cycle, fueling as well as NO and soot emissions are zero and (b) cold start emissions are ignored—instead the engine is assumed to be in fully warmed-up conditions from the beginning of the cycle. Ignoring the cold start emissions may, in general, influence instantaneous and total emission results but this holds particularly true for the CO and HC emissions that are not considered in this

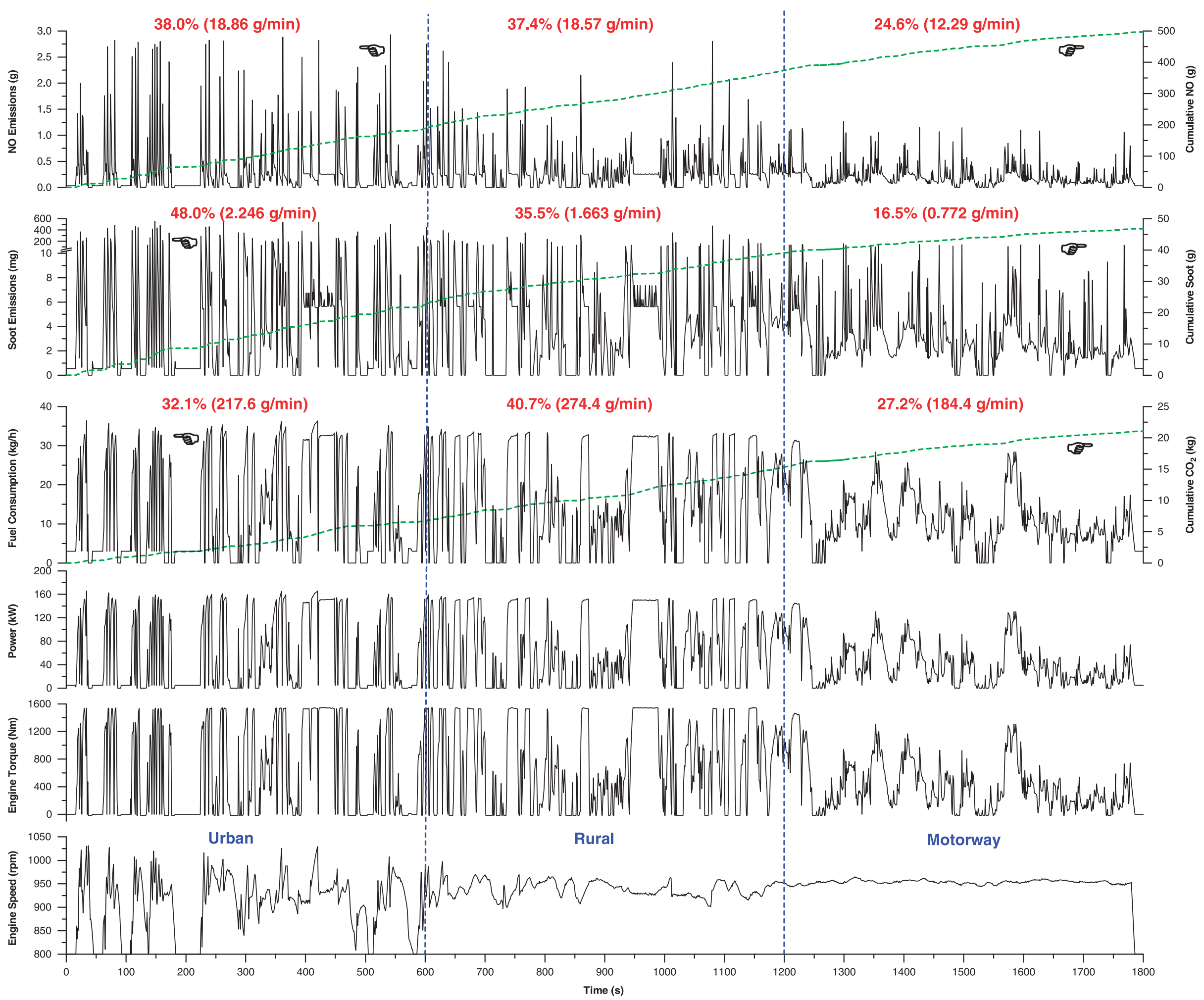

Figure 3. Calculated performance and emission results during the European ETC Transient Cycle for the current engine (percentage values inside the graph indicate the individual segment contribution to the total emissions or fuel consumption).

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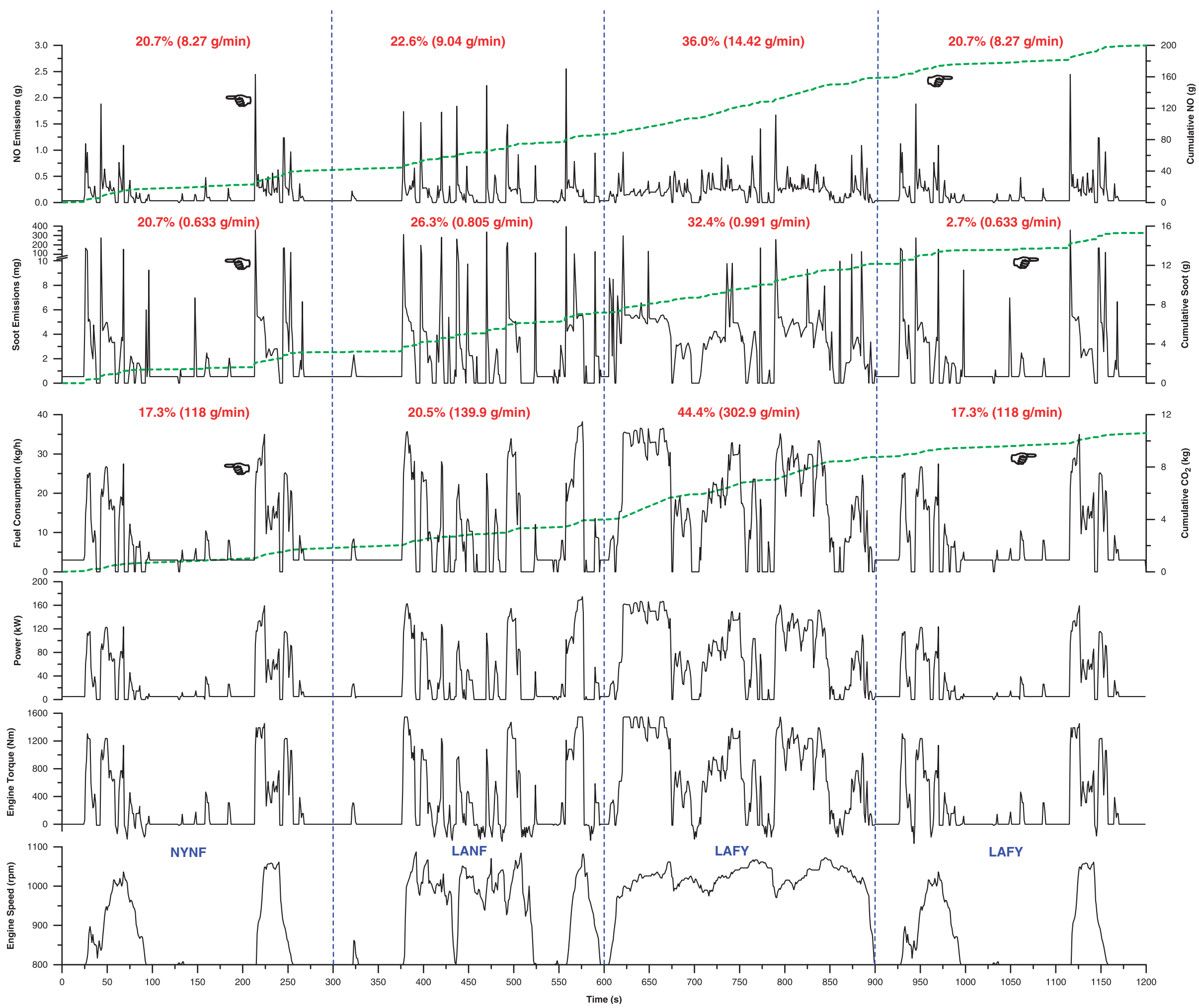

Figure 4. Calculated performance and emission results during the American FTP Transient Cycle for the current engine (percentage values inside the graph indicate the individual segment contribution to the total emissions or fuel consumption).

study. On the other hand, NO (and NO_x) emissions during cold starting are very low owing to the low gas temperatures involved.

4. RESULTS AND DISCUSSION

Application of the procedure described in the previous section will be demonstrated, as an indicative example, in Figures 3–5 for a turbocharged and after-cooled diesel engine without any after-treatment control. Figure 3 illustrates the engine performance during the ETC Transient Cycle; Figure 4 focuses on the American FTP and Figure 5 on the Worldwide WHTC. The following interesting results can be reached from the above figures:

(a) There is a strong influence of instantaneous torque on all performance results (power, fueling, CO_2 emissions).
(b) During the last (motorway) part of the ETC and the WHTC Cycle, the engine speed is practically constant, typical during this kind of driving, partially also enhanced by the rather narrow speed range of the engine in hand.

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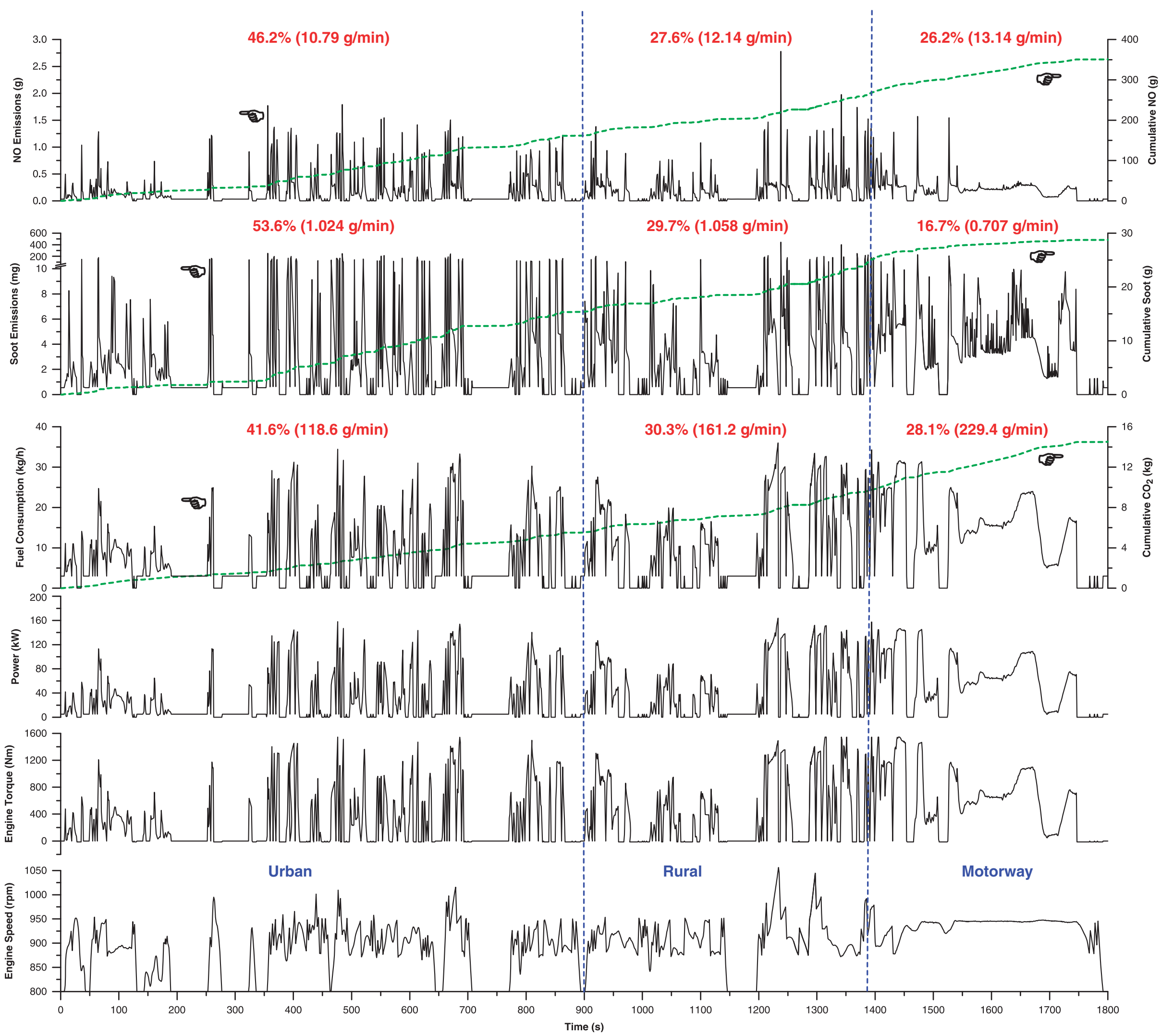

Figure 5. Calculated performance and emission results during the Worldwide WHTC Transient Cycle for the current engine (percentage values inside the graph indicate the individual segment contribution to the total emissions or fuel consumption).

(c) The American FTP Cycle is characterized by very long idling or motoring segments (particularly the NYNF first and last part), which, as will be discussed later, influences considerably the amount of total emissions.

(d) The ETC and the WHTC have much more frequent load and speed changes compared with the FTP (hence the much more 'dense' speed and torque diagrams—see also Table III later in the text), primarily during urban driving (during rural driving too), which results in a significant contribution of these segments to the total amount of emitted pollutants (particularly soot), as is quantified inside Figures 3 and 5.

(e) Following Equations (4), the points where the most abrupt load increases are experienced lead also to a considerable overshoot in emissions, that is why a 'broken' *y*-axis was chosen for the depiction of soot. Net soot production is mainly dependent on engine load. During a transient load or speed

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increase, locally very high values of fuel–air ratios are experienced owing to turbocharger lag. The instantaneous lack of air due to turbocharger lag, aided by the initial sharp increase in ignition delay during the early transient cycles after each individual load acceptance or acceleration, is responsible for the observed overshoot in particulate matter emissions [8]. Rapid increases in fuel injection pressure upon the onset of each instantaneous load step increase cause the penetration of the liquid fuel jet within the combustion chamber to increase. As the initial higher pressure fuel jets are injected into an air environment that is practically unchanged from the previous (steady-state) conditions, the higher momentum fuel jet is not accompanied by equally enhanced gas motion. Liquid fuel impingement on the cool combustion walls increases, lowering the rate of mixture preparation and enhancing the heterogeneity of the mixture. Moreover, the subsequent harder combustion course prolongs combustion and reduces the available time for soot oxidation [6,8,14].

(f) For NO emissions, it is, again, the lag between increased fueling and the response of the air-charging system that is responsible for the overshoot in emissions. As the main parameter affecting NO formation is the burned gas temperature, local high temperatures due to close to stoichiometric air–fuel mixtures increase NO emissions. Another primary mechanism that affects NO formation is the in-cylinder oxygen availability. Following the rapid decrease in the air–fuel ratio, oxygen concentration is rather limited during the turbocharger lag cycles after each transient event in the cycle, a fact that is responsible for a reduced production of NO, that is why the correction coefficient for NO in Equations (4) has been found one order of magnitude lower than its soot counterpart.

(g) The LAFY proves to be the most polluting part of the American FTP Cycle; this is attributed to the higher average speed, more frequent load increases and less idling period compared with the other parts.

(h) Owing to the long duration of urban driving for the WHTC (50% of the total), the corresponding contribution to the total emissions is higher than for the other Cycles. For example, urban driving contributes almost 54% of the total soot emissions during the WHTC, whereas for the ETC it is 48% and for the American FTP 41.4% (for both NYNF segments—these comprise 50% of the total cycle duration also but with much longer idling period than the WHTC). Similar results hold for the NO emissions and fuel consumption. However, when the emissions are expressed in $\mathrm{g\,min^{-1}}$ for each segment (which is a more representative value) things may differentiate; for example, the rural segment of the WHTC proves for the current engine to be at least as polluting as the urban driving, and the motorway part more fuel consuming owing to the higher average engine speed and less idling period than the other two segments. The most important finding is that the differentiation between each segment's (or each cycle's) emissions is largely dependent on the number of individual transients experienced by the engine, as will be discussed later in the text (comments regarding Figures 6 and 7).

(i) The fueling curve in Figures 3–5 corresponds roughly to carbon dioxide (CO_2) emissions also. As a first approximation following Equation (5), assuming complete combustion with a fuel–air equivalence ratio $\Phi<1$ and neglecting dissociation of the combustion products, $1\,\mathrm{kg\,h^{-1}}$ of diesel fuel ($\sim C_{12}H_{26}$ having a molecular weight of 170) corresponds to (1/170) $\mathrm{kmol\,h^{-1}}$ of fuel $\rightarrow 12\times(1/170)$ $\mathrm{kmol\,h^{-1}}$ $CO_2 \rightarrow 12\times(1/170)\times 44\,\mathrm{kg\,h^{-1}}$ $CO_2 = 3.106\,\mathrm{kg\,h^{-1}}$ CO_2.

$$\begin{aligned} C_{12}H_{26} + \overbrace{\frac{1}{\Phi}18.5\cdot O_2 + 3.76\frac{1}{\Phi}18.5\cdot N_2}^{\text{Air:}79\%N_2/21\%O_2\ v/v} \\ \rightarrow 12CO_2 + 13H_2O + \left(\frac{1}{\Phi}-1\right)18.5\cdot O_2 \\ + 3.76\frac{1}{\Phi}18.5\cdot N_2 \end{aligned} \tag{5}$$

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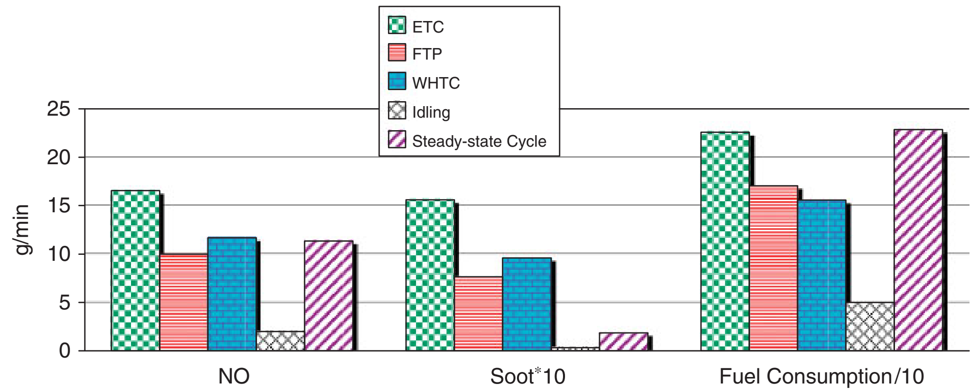

Figure 6. Calculated comparative emissions and fueling between the examined Cycles for the current engine (g min^{-1}).

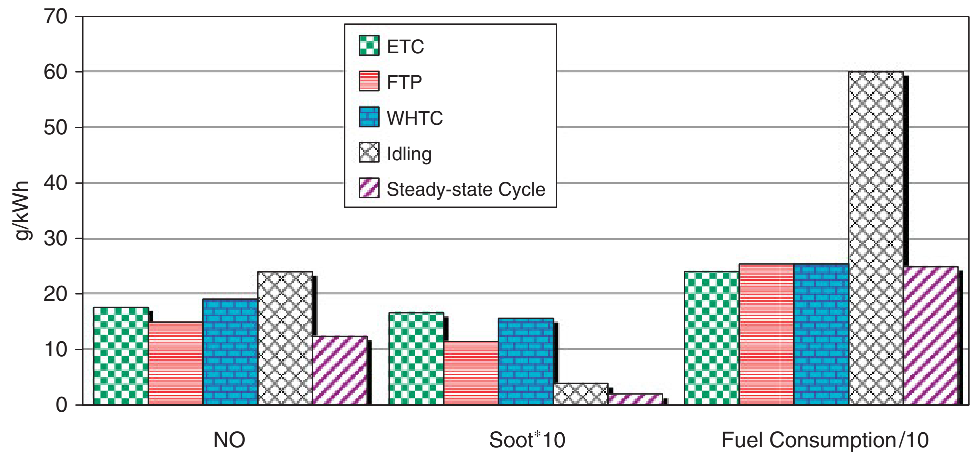

Figure 7. Calculated comparative emissions and fueling between the examined Cycles for the current engine (g kWh^{-1}).

Of course, owing to the lack of air during the early cycles after each acceleration in the cycle, the above relation between CO_2 and fueling is not always valid (CO is produced at the expense of CO_2) eventually leading to overestimated carbon dioxide emissions from Equation (5) (that is why these emissions are not analyzed in detail in the text but are only illustrated in Figures 3–5 for comparative purposes).

Figure 6 expands on the results of Figures 3–5 by detailing the cumulative emissions (g min^{-1}) in the three Cycles. In order to interpret these results, Tables II and III are provided, which summarize the main technical characteristics of the three examined heavy-duty Cycles, primarily as regards the most influential in terms of exhaust emissions transient schedules (load acceptances and accelerations) of each Cycle.

From Tables I and III, a first important remark, which constitutes a rather unique feature of the engine under study, is its narrow rotational speed range. As a result of this, only modest speed changes are experienced throughout each driving schedule, that is, 11.58% maximum acceleration for the FTP Cycle with 'only' 4.62% mean acceleration throughout the cycle (the values for the ETC and the WHTC are even lower).

It follows then, that the load changes, which are notoriously higher as documented in Table III, have the maximum effect on the emitted pollutants

Table II. Comparison between the individual steady-state characteristics of the three transient cycles.

	Duration (s)	Average normalized speed (%)	Maximum normalized speed (%)	Average torque (%)	Maximum torque (%)	Idling period (%)	Motoring period (%)
European ETC	1800	50.9	90.1	36.5	100	9.33	18.00
American FTP	1200	41.5	100	23.5	100	37.67	14.58
Worldwide WHTC	1800	37.1	100	23.7	100	21.72	22.22

Table III. Comparison between the individual transient characteristics of the three transient cycles.

	Number of torque reversals (min^{-1})*	Maximum acceleration for current engine (%)	Mean acceleration for current engine (%)	Maximum load increase	Mean load increase (%)	Number of greater than 300%/500%/1000% load increases (min^{-1})
European ETC	1.53	10.52 @ $t = 21$ s	2.81 (26.3 min^{-1})†	0–98% @ $t = 148$ s	15.51 (21.57 min^{-1})‡	4.53/3.80/3.30
American FTP	1.25	11.58 @ $t = 380$ s	4.62 (15.8 min^{-1})†	0–78.53% @ $t = 558$ s	13.80 (12.2 min^{-1})‡	3.00/2.90/2.50
Worldwide WHTC	0.83	4.58 @ $t = 774$ s	3.34 (25.43 min^{-1})†	'm'—84.7% @ $t = 1238$ s	12.28 (18.30 min^{-1})‡	3.50/3.27/3.00

*From motoring to at least 5% positive torque within one second.
†Number of accelerations per minute.
‡Number of load increases per minute.
'm' denotes motoring point.

(cf. remarks concerning Equations (4)). It is noted from Table III that the ETC Cycle (compared with the other two Cycles) is characterized by

- highest average torque (36.5%),
- shortest idling period (9.33%),
- highest frequency of torque reversals (1.53 per minute),
  - highest frequency of (rather soft in any case) accelerations, of the order of 26.3 per minute,
  - highest maximum load increase (0–98% at $t = 148$ s),
  - highest mean load increase (15.51%),
  - highest frequency of load increases per minute (this is the most important feature responsible for high emissions following Equations (4)—notice, for example, the small contribution of the ETC motorway driving (which has fewer transients than the other two segments) to the total soot emissions in Figure 3).

All the above-mentioned features are well known to contribute substantially to the exhausted pollutants. Unsurprisingly, the ETC turns out to be the most demanding of the three Cycles in terms of absolute engine emissions (in g), particularly soot and particularly its first segment, that is, urban driving, as has already been quantified in Figure 3. On the other hand, the American FTP Cycle, although having the steepest acceleration (at $t = 380$ s) is characterized by the longest (by far) idling period, least average torque (this is applicable for WHTC also) and fewest (12.2) load increases per minute. The WHTC falls somewhere between the other two in terms of transient events (as regards both magnitude and frequency), a fact that is responsible for its emissions being, roughly, in the middle of the European and the American Cycle.

All in all, the American FTP leads, for the current engine, to 40% less NO, 50% less soot and 24% less fuel consumption compared with the ETC, whereas for the WHTC the reduction in NO emissions is 29%, in soot 38.5% and in total fuel

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consumption 31% compared with the ETC. The fact that the average engine speed during the WHTC is lower than during the FTP is responsible for its lower fuel consumption compared with its American counterpart.

Similar comments to the ones mentioned above regarding Figure 6 apply if we want to compare and analyze the emissions of each cycle's segment (urban, rural, motorway) based on the values presented in Figures 3–5.

Things differentiate—sometimes considerably—when the *normalized emissions* are evaluated, that is, when the absolute emission (or fueling) values are reduced to the total work produced by the engine during the Cycle, and the emissions and fueling are calculated in $\mathrm{g\,kWh^{-1}}$—Figure 7. According to the current legislation, the following formula should be applied in order to calculate the normalized emissions from their instantaneous counterparts [10]:

$$\text{Normalized emissions} = \frac{\sum \text{Emissions}\,(t)}{\sum \text{Work}\,(t)} \quad (6)$$

There are three further characteristics of a Transient Cycle that influence strongly the normalized emissions (apart from the actual rate of emission production during each engine cycle or 'second'), namely the duration of the idling and motoring segments as well as the instantaneous power produced by the engine. The FTP and the WHTC have produced lower emissions (in $\mathrm{g\,min^{-1}}$) than the ETC but are also characterized by longer idling and motoring periods and smaller mean engine loading. Eventually, the total work produced by the engine is much smaller now compared with the ETC. As a result, the cumulative (from Equation (6)) normalized emissions of the FTP and WHTC are much more cohesive compared with the ETC Cycle.

For better understanding of the effect of emissions normalization through the total work produced by the engine, two fictitious Cycles have been considered: the first comprises continuous idling operation and the second is a constant steady-state Cycle at the average engine speed and load of the ETC (hence no load increases involved in either of these Cycles).

As was expected, during the continuous idling operation, very low emissions are produced (particularly soot, which is mainly dependent on engine load) and the fuel consumption is low (Figure 6). However, when these values are reduced to the negligible power produced by the engine, the normalized emissions increase (sometimes dramatically as is the case with the fuel consumption in Figure 7) being, thus, quite interesting and revealing of the real engine behavior.

For the 'average' engine Cycle, the absence of load increases has resulted in very low soot emissions (Figure 6); however, the lack of idling or motoring segments has produced normalized NO emissions and fueling of rather comparable magnitude compared with the three legislated Cycles.

As a final comment, it is remarkable that during the ETC, FTP and WHTC (but also during the fictitious 'average' Cycle), mean fueling in $\mathrm{g\,kWh^{-1}}$, that is, engine efficiency during the Test, are within 2–3% deviation.

5. SUMMARY AND CONCLUSIONS

A fast and, relatively, easy to apply approach was developed in order to be able to make a *first approximation* of engine performance and emissions during a Transient Cycle. The procedure is based on a previous steady-state experimental investigation of the engine for the formulation of polynomial expressions of all interesting engine properties with respect to rotational speed and torque (for the current investigation these properties were limited to power, fueling, NO and soot emissions). Correction coefficients are then applied to cater for transient emissions based on extended load and speed increase experimental investigation; owing to the narrow engine speed range a simple bi-linear correction function was found satisfactory. The simulation was applied for the case of a heavy-duty diesel engine running on three Transient Cycles, but can be easily employed to any other torque/speed vs time transient schedule and for any other emission property. It was shown that the most abrupt load increases (mainly experienced during urban driving) contribute mostly to the total emissions (primarily soot) during each Cycle. On the other hand, the motorway segments are characterized by higher fuel consumption (owing to the higher engine speeds)

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but contribute less to the total emissions owing to the more 'steady-state'-like driving profile.

A comparative study was performed for the European, American and the Worldwide Transient Cycles, American and the WHTC, which revealed in the most explicit way the importance of frequent abrupt transients on total emissions; the European ETC, being the most aggressive and having the shortest idling period, was found *for the current engine* to be the most demanding in terms of total produced emissions (in $\mathrm{g\,min^{-1}}$ not $\mathrm{g\,kWh^{-1}}$). This conclusion does not, by any means, suggest that the American Emission Standards are also leaner that the European or vice versa (see, e.g. [8] for a detailed tabulation of European, American and Japanese Emission Standards for diesel-engined vehicles). It should be pointed out that the engine used for the analysis has no EGR or after-treatment control, hence the rather high amount of total emissions. Its narrow speed range has also limited the effect of accelerations on the emitted pollutants. Engine efficiency during all three Cycles showed remarkably similar behavior.

A comparative analysis was also performed that detailed the individual steady-state and, mainly, transient characteristics of each segment of the Cycle.

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