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TNO report

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Correlation Factors between European and World Harmonised Test Cycles for heavy-duty engines

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Summary

A study was conducted for the determination of correlation factors between the European test cycles ESC and ETC and the World Harmonised test cycles WHSC and WHTC. The latter cycles are planned to replace the ESC and ETC in the emission legislation for on road heavy duty engines, starting with Euro VI.

With respect to the emission components, the focus was entirely on NO_x. The other components; particulates, HC and CO can be controlled relatively easily within the expected engine configurations for Euro VI.

The correlation was based on a modelling exercise in which engine measurement data was used. This made the correlation as much as possible independent of engine optimisation and the design choices of internal and external emission control devices.

Engine data such as friction losses, fuel consumption and exhaust temperature traces of five engines was available. These varied in technology from Euro III to a Euro VI demonstration engine, with aftertreatment such as DPF, SCR and DPF plus SCR.

The differences between the ETC and the WHTC are threefold:
- The average engine speed and load are different
- The WHTC includes an additional test with cold start
- The WHTC has a hot soak period before the hot test.

The activities to address the main differences between the ETC and WHTC were:
- Difference in engine cycle work (lower average speed and load for the WHTC): Evaluation of the internal friction losses of engines during the WHTC and ETC. The resulting NOx emission difference is calculated.
- The WHTC has an additional test with a cold start: Determination of the time necessary to heat-up the aftertreatment system. The additional NO_x emissions during the "cold" phase is calculated
- The WHTC has a hot soak period before the test with hot start: Determination of the possible temperature decrease of the aftertreatment system during the soak period and its possible influence on the NOx conversion of the SCR system.

For the correlation between WHSC and ESC only the first point, difference in cycle work, applies.

The analysis carried out led to the following conclusions:
- The difference in engine cycle work leads to a NO_x increase of 3% for the WHTC and 1.6% for the WHSC.
- The additional test with cold start of the WHTC leads to a NO_x increase ranging from about 8% to 13%. The 8% is projected for an optimised engine configuration and consequently used for the correlation factor.
- The hot soak does not lead to additional NO_x emissions when the soak time is 5 or 10 minutes.

The proposed correlation factors are presented in the following table. The correlation factor is defined as the factor to be applied to the ESC and ETC emission limits in order to obtain equivalent emission limits according to the WHSC and WHTC test procedures.

Proposed correlation factors between WHTC and ETC, respectively WHSC and ESC.

Emission Component	WHTC *) compared to ETC	WHSC compared to ESC
NO_x	1.10	1.00
CO	1.00	1.00
HC	1.00	1.00
Particulates	1.00	1.00

*) for composite test: WHTC with cold start, soak period and WHTC with hot start.
Weighting: 10% for cold WHTC and 90% for hot WHTC.

The table shows that all correlation factors are set to 1.00 except for a NO_x correlation factor of 1.10 for the WHTC. This is based on the NO_x increase of 11% (3% for engine cycle work and 8% for cold start).

Contents

1 Introduction

This report presents the findings and results of a study for the determination of correlation factors between the currently applicable European ESC and ETC test cycles and the World Harmonised WHSC and WHTC test cycles. The latter cycles are planned to replace the ESC and ETC in the emission legislation for on road heavy duty engines, starting with Euro VI.

The World Heavy-Duty Certification procedure has been developed within the framework of UN ECE World Forum for the Harmonisation of Vehicle Regulations (WP.29) in Geneva. The procedure includes:

- A ramped World Harmonised Steady-state Cycle (WHSC)
- A World Harmonised Transient Cycle (WHTC).

These cycles represent driving conditions in the European Union, USA and Japan.

The application of standard test cycles across the world will lead to costs reductions for engine development, since the development and production of engines and heavy-duty vehicles is a world-wide business.

The limit values proposed for the Euro VI legislation are based on the European Steady-state Cycle (ESC) and the European Transient Cycle (ETC). The emission limits in combination with the test cycles and other details of the test procedure determine the development challenge and the engine configuration. For that reason, the emission limits may need to be re-evaluated when the test procedure is changed. The correlation factors establish the relationship between the current and the world harmonized test procedure.

Under this specific service contract No 7 (S12.469892), the consortium proposed to conduct correlation measurements on three engines of various size and technology. ACEA was requested to provide these engines. During the project, it became clear that this approach was not feasible due to technical and confidentiality issues. Consequently it was agreed between the Commission services, the involved stakeholders and the contractor that the correlation would be performed based on a modelling exercise in combination with practical engine data to determine internal friction losses and heat-up time of the aftertreatment system after a cold and hot start.

This report starts with a comparison of the European and World Harmonised test cycles in chapter 2 and a theoretical assessment of the correlation methodology in chapter 3. Finally the correlation between the European and World Harmonised cycles is evaluated with experimental data of five engines in chapter 4, which leads to a proposal for correlation factors in chapter 5.

1.1 Project team and contributions

Although the project was primarily carried out by TNO personnel, much information including a large amount of second by second data was supplied by external parties. These parties, which are gratefully acknowledged, are:

– AECC: Association of Emission Control by Catalysts, Brussels
– TÜV Nord in Essen, Germany, under commission of ACEA

- DG-JRC: Joint Research Centre, Ispra, Italy
- EPA: Ann Arbor, USA.

2 European and World Harmonised test cycles

The current European emission legislation includes the following test types:
- European Steady-state Cycle (ESC)
- European Transient Cycle (ETC)
- European Load response (ELR)

The World Harmonised emission test procedure includes the following test cycles:
- WHSC: World Harmonised Steady-state Cycle
- WHTC: World Harmonised Transient Cycle

The ETC and WHTC are transient test cycles to be carried out on a transient engine dynamometer. Engine speed and load patterns are presented in respectively Figure 1and Figure 2. The ESC and WHSC are steady-state tests with 13 modes or test points within the engine map. Refer to Table 1. One key difference between the ESC and the WHSC is the fact that for the WHSC the emissions are continuously sampled. The length of each mode determines the weighing factors and the ramps between the mode points are also included.

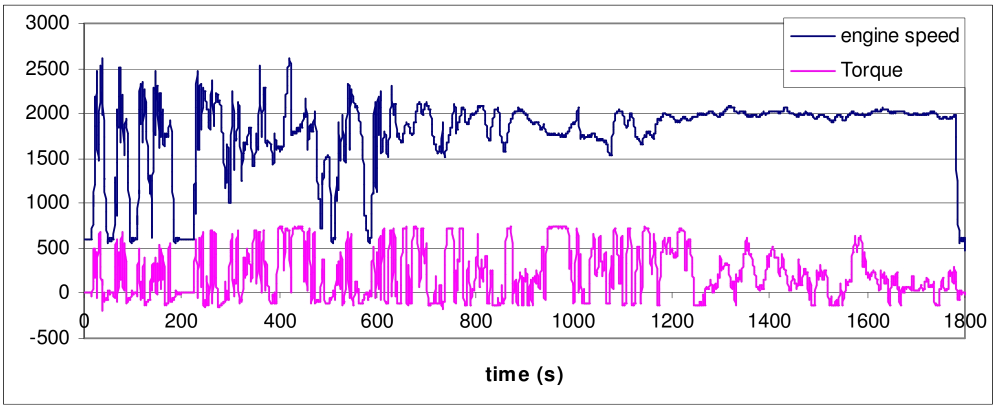

Figure 1: European transient Test cycle (ETC)

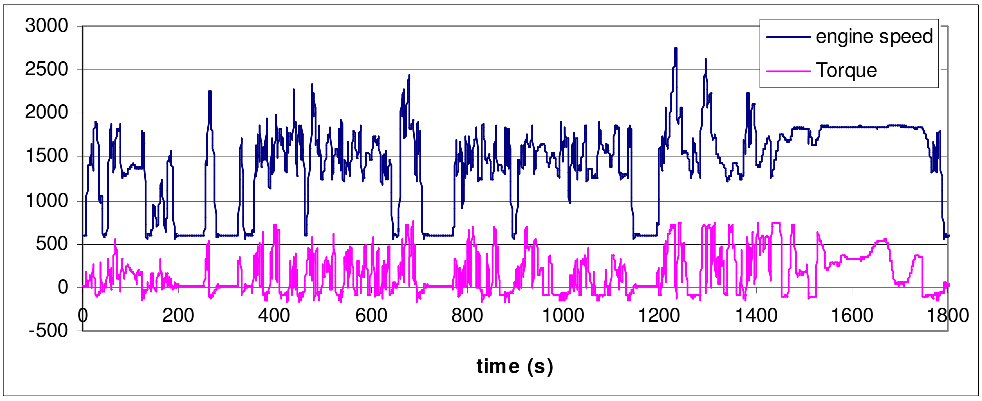

Figure 2 World Harmonised Transient Cycle (WHTC)

The differences between the ETC and WHTC are more substantial, and therefore the correlation factors evaluation is more focussed on those two cycles. The differences between the ETC and the WHTC are threefold:

- The average engine speed and load are different
- The WHTC includes an additional test with cold start
- The WHTC has a hot soak period before the hot test.

The tests with cold and hot start are weighted for respectively 10% and 90% in the end result. This is graphically presented in Figure 3. The length of the soak time is still under consideration. The most likely soak time for Europe is 5 or 10 minutes, but for the USA 20 minutes is still an option.

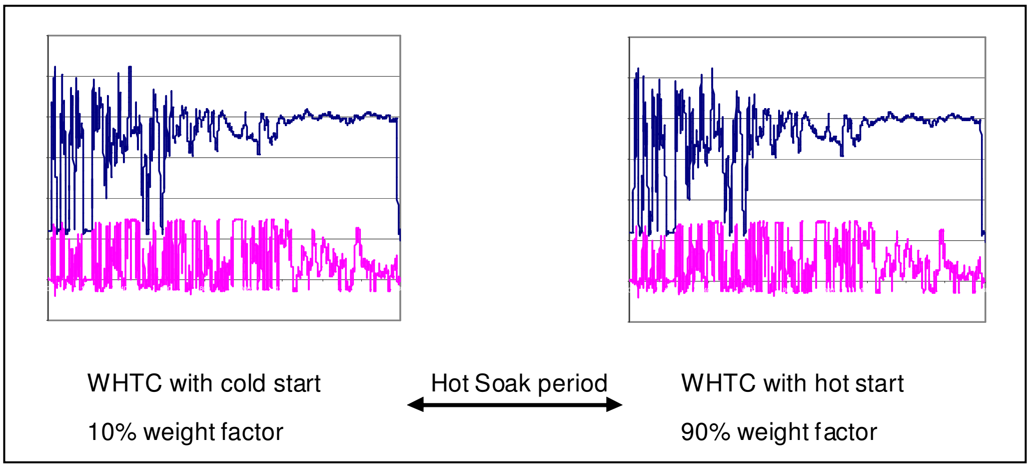

Figure 3: World Harmonised Transient Cycle (WHTC)

Table 1: a) European Steady-state Cycle b) World Harmonised Steady-state Cycle

ESC			
Mode	Normalised speed %	Normalised load %	WF %
1	idle	0	15
2	A	100	8
3	B	50	10
4	B	75	10
5	A	50	5
6	A	75	5
7	A	25	5
8	B	100	9
9	B	25	10
10	C	100	8
11	C	25	5
12	C	75	5
13	C	50	5

WHSC			
Mode	Normalised speed %	Normalised load %	Mode length incl. 20s ramp s
0	motoring	-	-
1	0	0	210
2	55	100	50
3	55	25	250
4	55	70	75
5	35	100	50
6	25	25	200
7	45	70	75
8	45	25	150
9	55	50	125
10	75	100	50
11	35	50	200
12	35	25	250
13	0	0	210

3 Correlation methodology

3.1 Emission components

The question is for which emission components correlation factors should be developed and applied. Traditionally NO_x emissions and particulates emissions are the most critical emission components. Especially NO_x is always close to the emission limit, because of its trade-off with fuel consumption and particulates emission. In other words a higher NO_x level would allow design choices leading to lower fuel consumption and engine-out particulates emission. So for NO_x correlation factors between world harmonised and European test cycles should definitely be considered. For the other emission components it depends how close they are to the limits or how difficult it would be to control them given the design solutions for Euro VI whether correlation factors are required.

For Euro VI, it is expected that a wall flow diesel particulate filter will be generally applied. That brings the particulate mass emission far below the Euro VI limit value (i.e. 20% of the limit). Also HC and CO emissions are considered not to be critical. They are often far below the limit value and also for Euro VI it is expected that their emission level can be controlled with the right catalyst formulations. Especially because oxidation catalysts are already present for the regeneration of the DPF and as NH_3 slip catalyst for SCR. So it can be concluded that the correlation factors for particulates, HC and CO can be set to 1.0. For NO_x it is necessary to determine correlation factors, both for the correlation between WHTC and ETC and WHSC and ESC. Stakeholders supported this approach.

3.2 Engine test bed versus model based approach

For the development of correlation factors it was planned, according to the project proposal, to carry out correlation tests on the engine test bed with Euro VI type of engines. The difficulty with this approach is that it is very hard to obtain engines which are equally qualified for both the European and World Harmonised test procedures. Each test procedure in combination with emission limits, and certainly when the differences are substantial, may lead to different design choices for internal and external emission control devices and control strategies. For example a different load and speed pattern may lead to different design choices for the EGR system and different catalyst formulations and dimensions, etc. This lead to the conclusion that a way had to be found to develop correlation factors as much as possible independent of emission control system optimisation choices and to focus primarily on fundamental differences between the test procedures.

The main differences between the ETC and WHTC are:

- The WHTC has a lower average cycle work than the ETC (lower average speed and load), probably leading to higher internal friction losses.
- The WHTC has an additional test with a cold start
- The WHTC has a hot soak period in between the tests with cold and hot start.

It was decided to address these differences separately and use characteristic engine data as much as possible independent from the emission control devices.

The following approach was taken:

– Engine cycle work:
  Determine the correlation based on differences in internal friction: basically allow NO_x proportional to the indicated work (output plus internal friction) or proportional to CO_2.
– Cold start:
  Determine the time necessary to heat-up the aftertreatment system. Calculate the additional NO_x emission during the "cold" period (proportional to total work or CO_2).
– Hot soak:
  Determine the possible period that the aftertreatment is below light-off temperature due to the hot soak and calculate the additional NO_x.

This approach is further described in the paragraphs below.

3.3 Differences in engine cycle work

With respect to engine cycle work the aim was to chose a technology neutral (emission control neutral) method of comparison between ETC and WHTC. This is because some NO_x control methods are easier or more efficient at low load while others are more efficient at high loads.
In an engine cycle with lower average load, relatively more energy is used to compensate for the friction losses of the engine, i.e. relatively more fuel is burned and emissions are produced. An "honest" comparison method would be to allow an equal amount of NO_x per amount of indicated work or per amount of CO_2. The results from indicated work and CO_2 are slightly different, since in the latter also the variation of indicated efficiency across the engine map plays a role in the comparison.

Three calculation methods are chosen for the comparison of ETC and WHTC:

1. Average engine efficiency
2. Mechanical efficiency with actual friction per engine and test cycle
3. Mechanical efficiency with constant average friction torque coefficient:
   $\bar{t}_{friction} = T_{average}/T_{max}$

The outcome of these calculations depends on the engine design and configuration. In particular the following engine parameters will have an influence:

- Specific power output: the higher the specific power output of the engine, the lower the relative mechanical friction and thus smaller differences can be expected between the two test cycles.
- Variation of indicated efficiency across the engine map (for method 1).

The calculations for the three methods are done for several engines.

3.3.1 Average engine efficiency

The principle idea is that a higher NO_x emission would be reasonable if the engine efficiency during the WHTC is lower than during the ETC. Engine efficiency includes mechanical and indicated efficiency. The mechanical efficiency is the ratio between effective work at the engine output shaft and the work delivered by the thermal process on the engine pistons. The indicated efficiency is the ratio between the work delivered to the engine pistons divided by the fuel energy.

The average engine efficiency can be calculated based on the end results of the ETC and WHTC (both with hot start) with the following equation:

$$\eta_{engine} = \frac{W_{out}}{W_{fuel}} = \frac{W_{out}}{m_{fuel} \cdot H_{lower}}$$

In which the work across the test cycle W_{out} and W_{fuel} is expressed in MJ or kWh. m_{fuel} is the fuel consumed during the test and H_{lower} is the lower combustion value of diesel fuel which is 42.7 MJ/kg.

Alternatively, the engine efficiency can be calculated using the average power and average fuel flow ϕ_{fuel}:

$$\eta_{engine} = \frac{\overline{P}}{\phi_{fuel} \cdot H_{lower}}$$

3.3.2 *Mechanical efficiency with actual friction*

The mechanical efficiency is defined by the following equation:

$$\eta_{mech} = \frac{W_{out}}{W_{indicated}} = \frac{W_{out}}{W_{out} + W_{friction}} = \frac{\overline{P}}{\overline{P} + \overline{P}_{friction}} = \frac{1}{1 + \overline{P}_{friction} / \overline{P}}$$

In which $\overline{P}$ and $\overline{P}_{friction}$ are respectively average power (output) and average friction power. $\overline{P}_{friction}$ can also be expressed as follows:

$$\overline{P}_{friction} = 2\pi \overline{T}_{friction} \cdot \overline{n}$$

The actual friction torque during the test cycle is described with the following function:

$$T_{friction} = a + b \cdot n$$

In which n is engine speed. The constants a and b are engine specific. These factors will be determined based on engine motoring curves or based on Willans lines.

3.3.3 *Mechanical efficiency with constant average friction torque*

The average friction torque during a test cycle is:

$$\overline{T}_{friction} = a + b \cdot \overline{n}$$

Consequently:

$$\overline{P}_{friction} = 2\pi (a \cdot \overline{n} + b \cdot \overline{(n^2)})$$

It should be noted that all friction losses during the test cycle are included, so also during engine idle and engine motoring.

In paragraph 4.2.3, the mechanical efficiencies for the different test cycles are determined in two ways:

1. Using the actual friction: calculated per engine and per test cycle. This is done for both the correlations between WHTC and ETC and WHSC and ESC.
2. Using an average friction torque coefficient $\bar{t}_{friction}$, which is the same for all engines and all test cycles. The variation across the engines evaluated will then only be dependent on the shape of the maximum torque curve. This is only done for the WHTC and ETC.

For method 2, the average engine friction is described with the following equation:

$$\bar{P}_{friction} = 2\pi \bar{T}_{friction} \cdot \bar{n} = 2\pi \cdot \bar{t}_{friction} \cdot T_{\max} \cdot \bar{n}$$

In which $\bar{t}_{friction}$ is the average friction torque coefficient.

3.4 Cold start

The introduction of a cold start and a hot soak is new within the European emissions legislation. The proposed procedure contains a test with a cold start, followed by a hot soak and a start with a warm engine. Consequently an overall result is calculated using weighting factors for the tests with cold and with hot start. For the WHTC in Europe currently weighting factors are proposed of 10% for the cold start test and 90% for the hot start test.

The majority of the Euro V engines are currently equipped with (urea) SCR deNOx exhaust aftertreatment systems. Also for Euro VI it is expected that SCR deNOx catalysts will be generally applied. This will most likely be in combination with engine measures such as EGR in order to be able to meet the NO_x emission limit. Also a wall flow diesel particulate filter will be most likely a part of the exhaust aftertreatment system in order to meet the required particulate emission level.

The NO_x emissions from such a Euro VI engine during a test with a cold start will be higher than over a test with hot start, because it takes time to heat-up the SCR catalyst. The heat-up time will be dependent on the exhaust system configuration such as the lengths and thermal inertia of the pipes and also on the position of the diesel particulate filter. The NO_x increase will be dependent on the SCR heat-up time (until light-off) and the engine-out NO_x level. The engine-out NO_x level can be seen as inversely proportional to the average SCR system NO_x conversion efficiency during a test with hot start and a fixed value for the NO_x test result.

The time to reach the light-off temperature of the SCR catalyst is dependent on many parameters:

- Configuration of the aftertreatment system: diesel particulate filter upstream or downstream of the SCR catalyst. Each catalyst or DPF represents a certain thermal inertia which causes a delay time in heating up. The delay time is proportional with the relative size and mass of the component.
- Length and diameter of the exhaust pipes upstream of the SCR catalyst and the kind of insulation.

– Possible engine control strategy to shorten heat-up period, such as fuel injection timing, EGR control strategy and turbocharger control.

Also the conditioning cycle before the cold test can have a significant influence on the emissions of a cold test. When the conditioning cycle ends with low exhaust gas temperature, a certain amount of NH_3 can be stored within the catalyst. This will likely increase the SCR efficiency during the cold start test. It is advised to precisely define the conditioning cycle, such that this effect does not lead to differences.

For the analysis of the NO_x increase due to the cold start, the exhaust gas temperature traces of the engines available for this program will be analysed and a generic SCR light-off characteristic will be determined.
The following calculations will be made:

1. Estimation of the time constant for heat-up of the exhaust aftertreatment system from the available temperature traces.
2. Calculation of the required average aftertreatment system temperature in order to meet the SCR efficiency across the WHTC.
3. Calculation of the NO_x conversion during the WHTC with cold start based on the aftertreatment system temperature.

3.5 Hot soak

A hot soak means that the engine stands still for some time before the test with hot engine is conducted. The hot soak period according to R49 is 5 minutes, although also a period of 10 minutes has been evaluated. There may be a NO_x increase in comparison with a test without soak time, because during the soak time the SCR catalyst and exhaust system components can cool down to below the light-off temperature. Whether this will happen is dependent on the exhaust system and catalyst temperature at the end of the first test (with cold start) and the cool down curves of the exhaust system components. There are also secondary effects such as:

– cooling down of the engine itself and heating up after the soak period.
– possible NH_3 storage in the SCR catalyst from the test with cold start which influences the light-off characteristics after the soak period.

These effects are considered too small or too engine specific and will not be taken into account in the analysis.

The following analysis will be done in order to analyse the influence of the hot soak period:

– the exhaust temperature during the last part of the WHTC will be collected.
– the exhaust aftertreatment system cool down during the soak will be evaluated.
– the heat-up time to reach light-off (if temperature falls below light-off temperature) will be determined
– the additional NO_x emissions will be calculated in case the temperature drops to a level where the SCR efficiency is reduced compared to the hot test.

4 Results

4.1 Engines evaluated

Data of five engines was available for this program. They were all six cylinder engines with an engine displacement ranging from 6 to 12 litres. Information about these engines is presented in Figure 4 (cylinder displacement, torque and power output) and in Table 2 (Euro class and aftertreatment configuration). Figure 5 shows sample points of various test cycles within the engine maps.

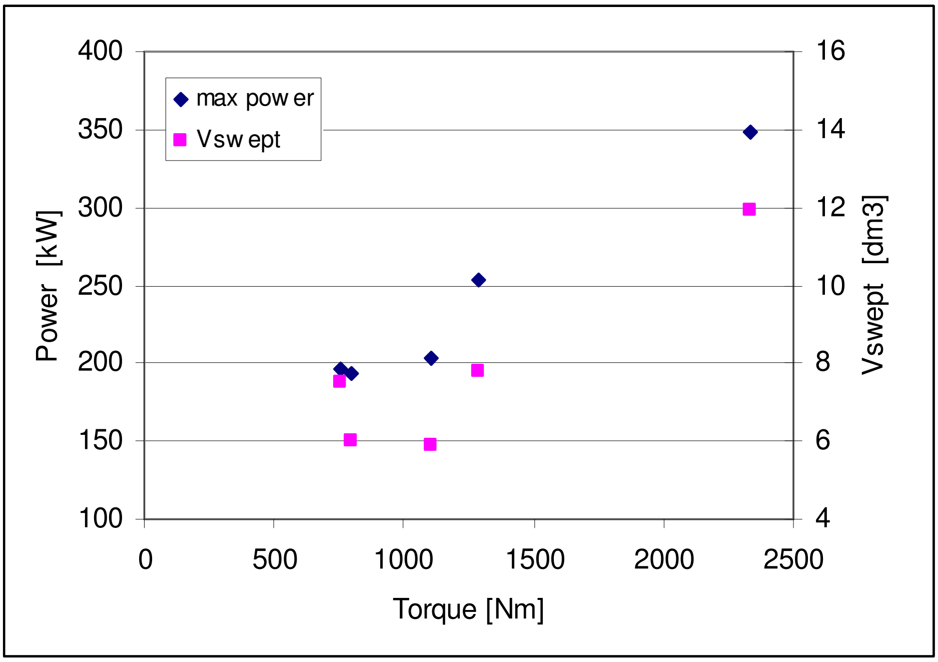

Figure 4 maximum engine power and torque of the engines used for the program

Table 2: Overview of engine configurations used for the program.

Engine	Aftertreatment	Insulation Pipes	Insulation DPF / catalysts	Total catalyst + DPF volume / Engine displacement
Euro V	SCR	No	No	3.5
Euro V / EEV	DPF + SCR	No	No	6.5
Euro VI demo	DPF + SCR	Yes	No	4.7
Euro III	CRT	Yes	No	
EPA 2004+	CDPF + SCR	No	No	4.4

The amount of data available per engine varied.
Table 3 gives an overview of the analysis that can be done with the available data.

Table 3: availability of data for the different types of analysis

Analysis type	Number of engines data available
WHTC and ETC engine efficiency	4
WHTC and ETC mechanical friction	4
WHSC and ESC engine efficiency	1
WHSC and ESC mechanical friction	3

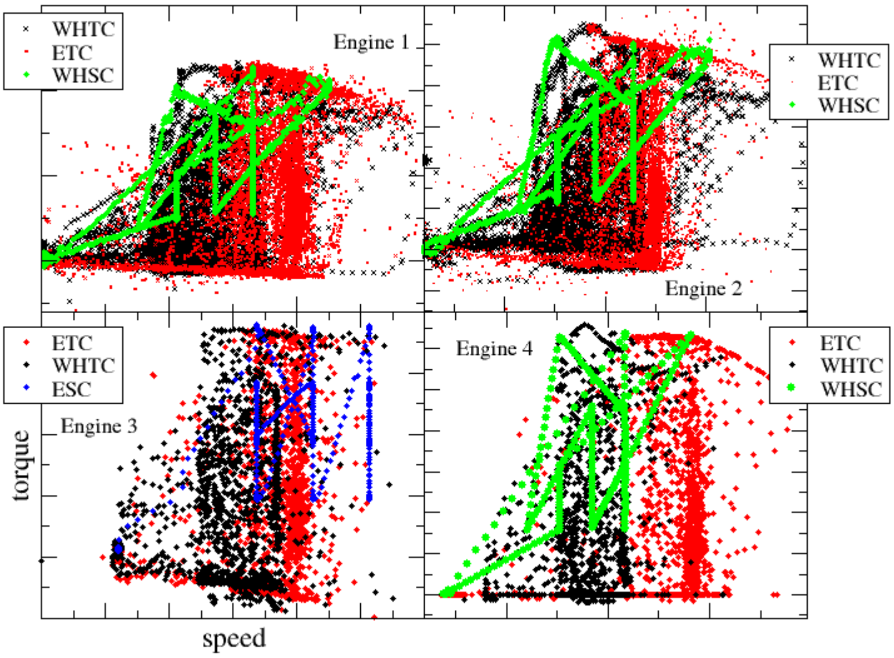

Figure 5: Engine points during different test cycles for four engines: WHTC in black, ETC in red, WHSC in green, ESC in blue

4.2 Engine cycle work

4.2.1 Average engine efficiency

Refer to paragraph 3.3.1. The average engine efficiency is calculated based on the end results of the test cycles (all with hot start) with the following equation:

$$\eta_{engine} = \frac{W_{out}}{W_{fuel}} = \frac{W_{out}}{m_{fuel} \cdot H_{lower}}$$

Or with the equation:

$$\eta_{engine} = \frac{\overline{P}}{\phi_{fuel} \cdot H_{lower}}$$

Depending on the precise information available.

It should be noted that engine efficiency is composed of two underlying efficiencies namely mechanical efficiency and indicated efficiency.

The cycle work and fuel consumption are measured during the test cycles. The cycle work is only the positive work (torque larger than 0; relevant for transient test cycles). The results are presented in Table 4 and Table 5 for respectively the WHTC - ETC and WHSC – ESC correlation.

From Table 4, it can be concluded that the variation between the engines is very large. For engines 1 and 2 the engine efficiency during the ETC is 3% to 4% higher than during the WHTC. For engines 3 and 4, it is the other way around: engine efficiency during the WHTC is 6% to 10% higher than during the ETC. It is likely that the shape of the indicated efficiency across the engine map causes this effect.
The correlation factor (last column of Table 4) is defined as a factor to be applied to the NO_x during the European test in order to obtain the equivalent value for the World Harmonised test procedure.
The average correlation factor for these engines is smaller than 1 meaning that the NO_x emissions according to this method would be slightly lower for the WHTC. For the WHSC to ESC comparison only one engine is evaluated (Table 5). This shows a slightly higher engine efficiency for the WHSC.

Table 4: Average engine efficiency for WHTC – ETC correlation

Engine	WHTC	ETC	WHTC/ETC	Correlation factor
1	0.402	0.416	0.967	1.034
2	0.373	0.389	0.960	1.042
3	0.312	0.294	1.061	0.943
4	0.410	0.373	1.102	0.908
Average	0.374	0.368	1.022	0.982

Table 5: Average engine efficiency for WHSC – ESC correlation

Engine	WHSC	ESC	WHSC/ESC	Correlation factor
1	0.4288	0.4197	1.0217	0.979

4.2.2 *Friction losses*

The friction losses are determined based on different methods depending on the type of information that was available. The following methods are used:

- based on the engine motoring curve (the curve when the engine is driven by the engine dynamometer with no fuel delivered to the engine (engines 1 and 2)
- based on the Willans lines of one of the steady-state tests. (engines 3 and 4). The Willans lines are straight lines of constant speed with fuel consumption as a function of torque. The intersection of the line with the torque axis should yield the friction torque.

The friction losses are approached as a linear function of engine speed:

$T_{friction} = a + b . n$

In which n is engine speed and a and b are constants for a particular engine.

Using the Willans lines appeared to be not straightforward, primarily because the steady-state data does not cover the full engine map. Basically with the WHSC and the ESC, low engine speeds and also low loads (0-25% load) are not or not well covered.

The fuel consumption is assumed to be a function of torque and speed, where the coefficients are fitted through a least square fit. In order to cope with the limited data sets, several equations were evaluated in order to predict fuel consumption (FC) as a function of speed (n) and torque (T). The following equations led to the best results (i.e. lowest standard deviation in combination with sensible and stable results):

$$FC/n = C . n + D . T \qquad (1)$$
$$FC/n = C . (n + 500) + D . T \qquad (2)$$

In which C and D are constants (refer to Appendix B). With these equations the standard deviation is in the range of 7% to 10% depending on the engine and equation. With the first equation, friction is fit to zero at zero speed. This led to relatively stable results ensuring proper speed dependence, but with a relatively large error at idle. This is improved with the second equation, where coefficient “a” in the $T_{friction}$ equation equals to 500 x “b”. Full details are presented in Appendix B including the results of other fit methods. For two engines motoring curves were available. For these engines the friction losses based on the Willans lines fit is compared with the friction based on the motoring curve. For one engine this led to a relatively large difference. Refer to Appendix B.

The engine friction as a function of speed of the five engines is presented in Figure 6. The lines of engines 1 and 2 are based on the motoring curve while the lines of engines 3 and 4 are based on equation 2 (Willans lines). Engine 5 is based on the data points of the ETC, which leads to a rough approximation.

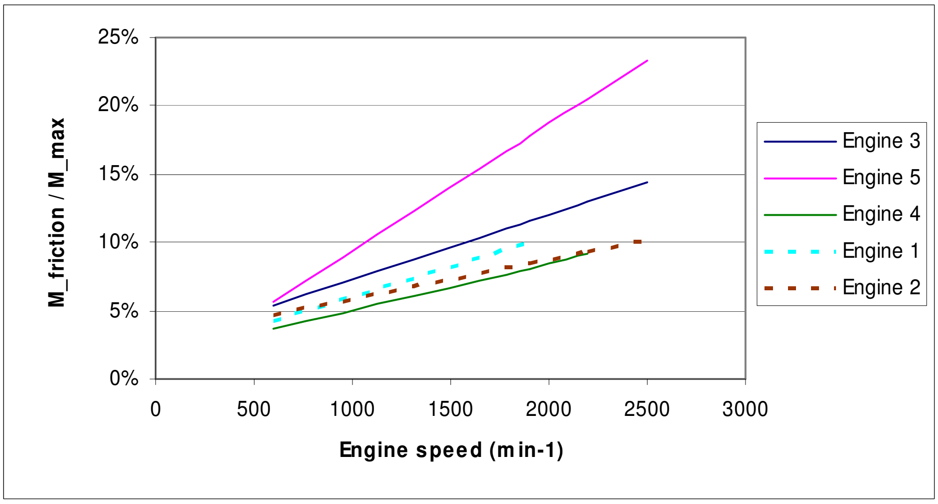

Figure 6: Engine friction of five engines used for the mechanical efficiency calculations

4.2.3 Mechanical efficiency

4.2.3.1 Actual friction per engine and test cycle

Mechanical efficiency is defined by the following equation:

$$\eta_{mech} = \frac{W_{out}}{W_{indicated}} = \frac{W_{out}}{W_{out} + W_{friction}} = \frac{\overline{P}}{\overline{P} + \overline{P}_{friction}} = \frac{1}{1 + \overline{P}_{friction} / \overline{P}}$$

The average engine friction power is calculated with:

$$\overline{P}_{friction} = 2\pi(a \cdot \overline{n} + b \cdot \overline{(n^2)})$$

Constants a en b are engine specific. Also refer to paragraph 3.3.2. $\overline{P}_{friction} / \overline{P}$ and other parameters are calculated from the s by s data available for the engines described in paragraph 4.1 From the equation it can be seen that η_{mech} is a direct function of $\overline{P}_{friction} / \overline{P}$ (or $P_{friction} / P_{average}$). This is graphically presented in Figure 7. The figure shows the relative position of the different test cycles.

For four engines both WHTC and ETC second by second data was available. The results are presented in Table 6 and Figure 8. From these, it can be concluded that the mechanical efficiency during the WHTC is in general somewhat lower than during the ETC. The difference varies between 1.5% and 4.6%. The correlation factor based on these analyses is presented in the last column of Table 6.

The results for the correlation between WHSC and ESC are presented in Table 7. From the table it can be concluded that the difference in mechanical efficiency is very small. The friction during the WHSC is 0.2% to 3.4% higher than during the ESC for the three engines evaluated.

Table 6: Mechanical efficiency of WHTC and ETC based on a s by s calculation including idle and motoring points

Engine	WHTC	ETC	WHTC / ETC	Correlation factor
1	0.796	0.822	0.968	1.033
2	0.783	0.818	0.956	1.046
3	0.736	0.747	0.986	1.015
4	0.804	0.825	0.975	1.026
Average	0.780	0.803	0.971	1.030

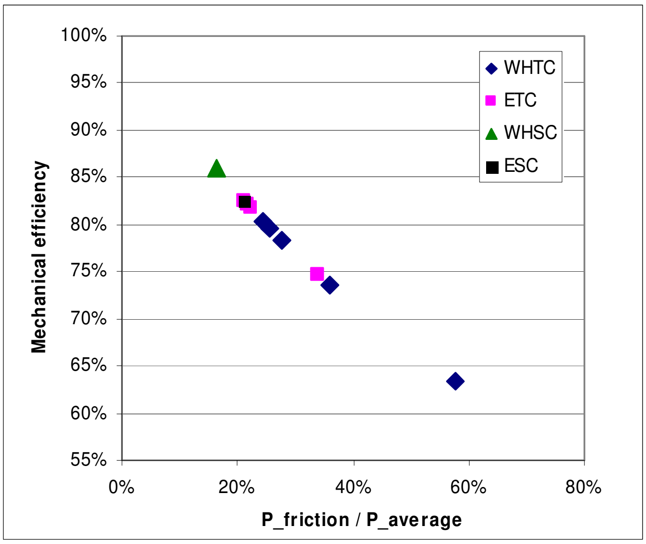

Figure 7: Relation between friction losses and mechanical efficiency for five engine types.

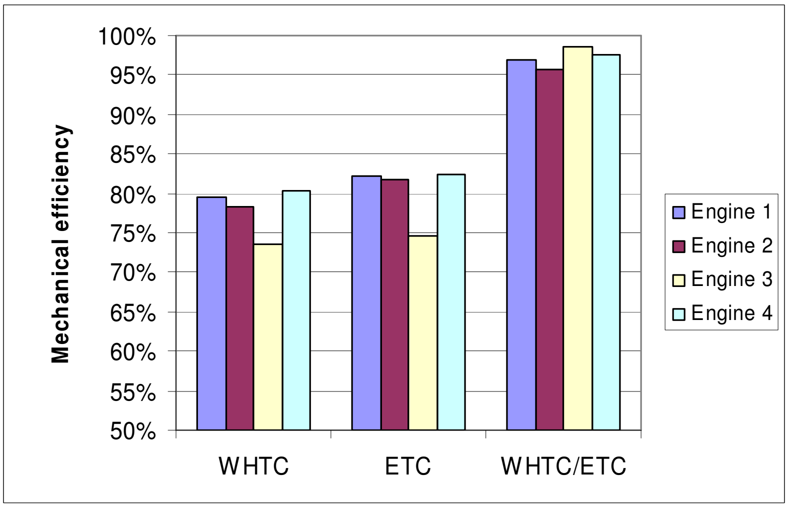

Figure 8: Mechanical efficiency WHTC and ETC including idle and motoring points

Table 7: Mechanical efficiency of WHSC and ESC based on a s by s calculation including idle

Engine	WHSC	ESC	WHSC/ESC	Correlation factor
1	0.8467	0.8577	0.9873	1.013
2	0.8369	0.8651	0.9674	1.034
4	0.9317	0.9337	0.9978	1.002
Average	0.8718	0.8855	0.9841	1.016

The average friction coefficient $\bar{t}_{friction}$ (=$T_{average}/T_{max}$) is calculated for each individual engine. In Figure 9, the relation is presented between $\bar{t}_{friction}$ and the mechanical efficiency.

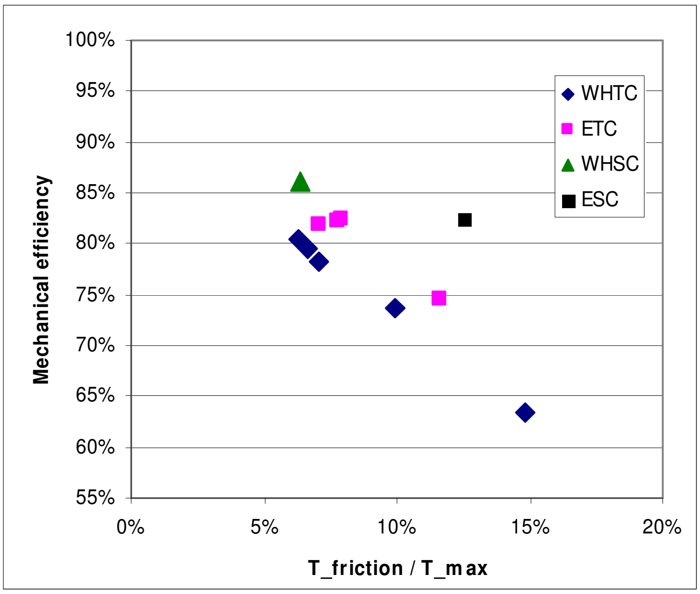

Figure 9: Relation between $\bar{t}_{friction}$ ($T_{friction}/T_{max}$) and mechanical efficiency for five engine types

4.2.3.2 *Constant average friction torque coefficient*

In this paragraph the mechanical efficiency calculations are done with a fixed friction torque coefficient, $\bar{t}_{friction}$, independent of engine and test cycle. The friction coefficient is defined as average friction torque divided by the maximum torque.

The averages for $\bar{t}_{friction}$ of four engines for WHTC and ETC are presented in Table 8 (individual engines in Figure 9). The table shows that the average friction coefficient for the WHTC and ETC are respectively 0.075 and 0.085. The overall average is 0.080. This number is substituted in the equation for average friction power:

$$\bar{P}_{friction} = 2\pi \bar{T}_{friction} \cdot \bar{n} = 2\pi \cdot \bar{t}_{friction} \cdot T_{max} \cdot \bar{n}$$

Table 8: Average friction coefficient $\bar{t}_{friction}$ for different test cycles

	WHTC	ETC	Average WHTC/ETC
average $\bar{t}_{friction}$	0.075	0.085	0.080
based on # of engines	4	4	

Consequently the mechanical efficiency is calculated with:

$$\eta_{mech} = \frac{W_{out}}{W_{indicated}} = \frac{\overline{P}}{\overline{P} + \overline{P}_{friction}} = \frac{1}{1 + \overline{P}_{friction} / \overline{P}}$$

The result of this calculation for the four engines is presented in Table 9. It can be concluded that the average ratio between the mechanical efficiencies of the WHTC and the ETC is 0.943. This corresponds to a correlation factor of 1.061.

Table 9: Mechanical efficiency ratio between WHTC and ETC based on method with constant friction torque coefficient $\bar{t}_{friction} = 0.080$.

Engine	WHTC	ETC	WHTC/ETC	Correlation factor
1	0.764	0.817	0.935	1.070
2	0.761	0.800	0.952	1.050
3	0.776	0.811	0.957	1.045
4	0.763	0.823	0.927	1.078
average	0.766	0.813	0.943	1.061

4.2.4 *Comparison different methods for engine cycle work*

The results of the previous paragraphs are summarised in the tables below. Table 10 shows the comparison between the WHTC and ETC, while Table 11 shows the comparison between WHSC and ESC. Table 10 shows that the correlation factors vary, depending on the comparison method, from 0.982 to 1.061. A similar result is seen in Table 11 for the correlation between WHSC and ESC.

Mechanical efficiency with actual friction is seen as the best method of the three methods evaluated. With the first method, engine efficiency, indicated efficiency variations across the engine map result in a large variation of the correlation factor between 0.91 and 1.04 (Table 4). This influence is engine (calibration) specific and consequently undesirable for this comparison. With method 3, mechanical efficiency with constant $\bar{t}_{friction}$, the influence of the speed dependency of the friction losses can not be taken into account. This is also less desirable.
Consequently the second method is the preferred method for the determination of the difference in engine cycle work (differences in relative internal friction). This leads to the following correlation factors for difference in engine cycle work:

- WHTC / ETC: 1.030
- WHSC / ESC: 1.016

Table 10: comparison engine cycle work WHTC and ETC

Comparison method	WHTC	ETC	WHTC / ETC	Correlation factor
Engine efficiency	0.374	0.368	1.022	0.982
Mechanical efficiency with actual friction	**0.780**	**0.803**	**0.971**	**1.030**
Mechanical efficiency with constant $t_{friction}$	0.766	0.813	0.943	1.061

Table 11: Comparison engine cycle work WHSC and ESC

	WHSC	ESC	WHSC/ESC	Correlation factor
Engine efficiency	0.4288	0.4197	1.0217	0.978
Mechanical efficiency with actual friction	**0.8718**	**0.8855**	**0.9841**	**1.0163**

4.3 Cold start

During the WHTC with cold start a certain time period is needed to heat-up the aftertreatment system including the SCR catalyst. This can be seen as a delay time before the SCR system has its normal efficiency. Of course the NO_x emission control will probably also include an EGR system and depending on the engine operating conditions, the NO_x control can be based more on EGR than on SCR or visa versa. On top of that it can be decided to apply a special cold start strategy, in order to lower engine-out NO_x for the duration of the heat-up period. This can include for example:

- Injection timing retard
- EGR cooler and/or aftercooler bypass
- Increased EGR level

For the analysis within this paragraph, the additional NOx due to the cold start is calculated based in a variety of boundary conditions such as aftertreatment heat-up time, engine-out NO_x level and average SCR efficiency.

4.3.1 Simple approximation of the additional NO_x during the WHTC with cold start

Figure 10 shows a simple approximation of the additional NO_x emission during the WHTC with cold start as a function of the delay time, the time to heat-up the aftertreatment system. This is done for several SCR NO_x conversion values ranging from 80% to 95%. The engine-out NO_x is assumed to be linearly proportional to the fuel consumption (Figure 11) and the SCR efficiency is assumed to be zero before, and constant after the delay time.
The graph shows that the SCR conversion efficiency has a large impact on the additional NO_x emission. The higher the SCR efficiency, the higher the additional NO_x value. This is because with a fixed value for the hot NO_x test result, the engine-out NO_x is proportional with the SCR efficiency and therefore, as long as the SCR system is not operational, more NO_x passes the SCR catalyst.

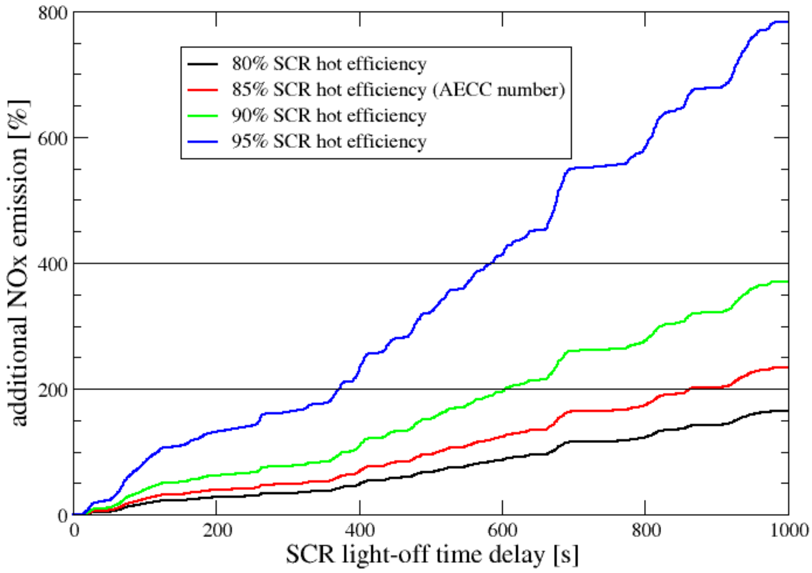

Figure 10: NO_x increase (in %) for WHTC with cold start as a function of SCR light-off time delay and SCR NO_x conversion

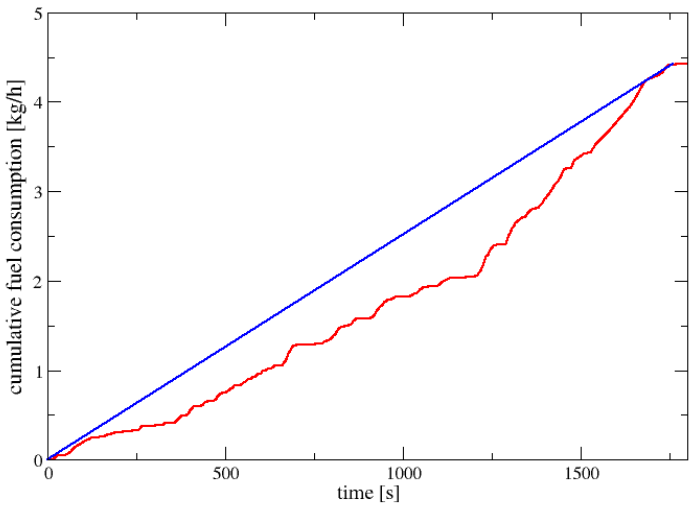

Figure 11: Cumulative fuel consumption during the WHTC test.

4.3.2 *SCR system NO_x conversion*

Some literature references about SCR conversion efficiencies for Euro VI are presented in Table 12. This shows NO_x conversion efficiencies in the range of 80% to 90%, with an average of 85%. Taking into account the capabilities of NO_x reduction with EGR and future technical capabilities of SCR, a wider range may be possible. Reference [AECC, 2007] shows an engine-out NO_x level of about 1.2 g/kWh. This would mean that an SCR conversion efficiency of about 70% would be sufficient to reach 0.4 g/kWh NO_x during the hot WHTC cycle. On the other hand, future developments of SCR catalysts, possibly in combination with engine thermal management, may make an SCR conversion efficiency of 95% possible.

Table 12: References about SCR NO_x conversion efficiency

Reference	Reference to engine technology	SCR NO_x conversion
[Ellensohn, 2007]	EGR + DPF + SCR	80%
[Schlick, 2007]	Internal engine + SCR	90%
[Rickert, 2007]	EGR + DPF + SCR	85%
[AECC, 2007]	EGR + DPF + SCR	85%

4.3.3 SCR system light-off

A typical SCR system light-off temperature curve is presented in Figure 12. This is characteristic for an SCR catalyst with upstream oxidation catalyst to increase the NO_2/NO ratio. The curve is based on various publications: [van Helden, 2004], [Winkler, 2003], [Walker, 2003], [Spurk, 2002] and [Chandler, 2000]. 200 °C is generally seen as the temperature where SCR conversion can start. This is more determined by the urea injection and hydrolysis (without risks of deposits formation) than by the SCR conversion itself. Under conditions that NH_3 can be buffered within the catalyst, NO_x conversion can start at slightly lower temperatures. In this characteristic curve, full NO_x conversion efficiency is reached at 250°C.

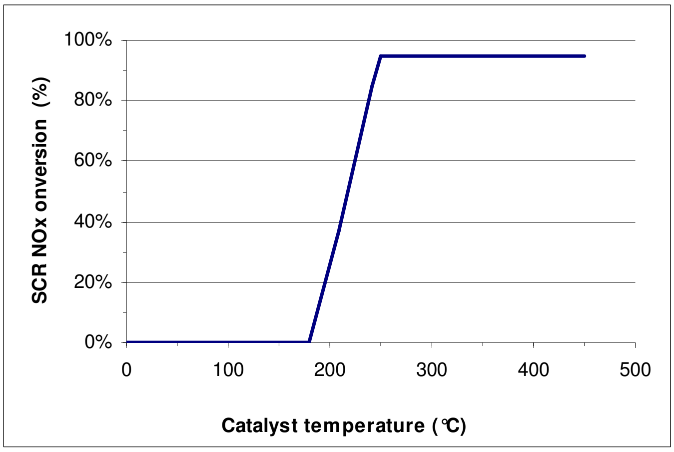

Figure 12: Typical SCR system light-off curve for SCR catalyst plus upstream NO-NO_2 catalyst.

4.3.4 Aftertreatment system heat-up time

Exhaust gas temperature traces of the engines used for this program are presented in Appendix A (Figure 18 and Figure 19) and in Figure 13 below. The figures in Appendix A show the temperatures upstream and downstream of a catalyst and/or DPF. Figure 13 shows only post catalyst temperatures.
The following notes should be made:

– The exhaust system configurations vary. Refer to Table 13. For other specifications about the engine technology refer to paragraph 4.1.

- The measurements were done at several laboratories within Europe and USA
- Measurements were made with simple temperature sensors which do not accurately measure the actual gas temperature. Temperature can lag behind because of thermal inertia, heat conduction and radiation to the pipe wall.
- None of the engines, of which data was available for this analysis, was optimized for cold start or had any thermal management components or strategies. For a diesel passenger car, time to reach light-off temperature is in the range of 200 – 250 s (SCR catalyst upstream of DPF) [Tennison, 2004] .

Looking at Figure 13, the following can be concluded:
- The post catalyst temperature of engine number 4 remains also in the hot test during the same large part of the cycle as the cold test (1319 seconds), below the light-off temperature. This would not be acceptable for an SCR engine. It does not correspond well with the pre-catalyst temperature which passes 200°C after 430 s. Consequently the post catalyst temperature of this engine for the determination of light-off is not very relevant.
- For three out of the five engines, it can be seen that after 500 to 600 seconds the post catalyst temperatures of the hot and cold test are very close to each other (max 30°C difference) and move parallel with time. The point where they start moving parallel indicates the end of the cold start phase.

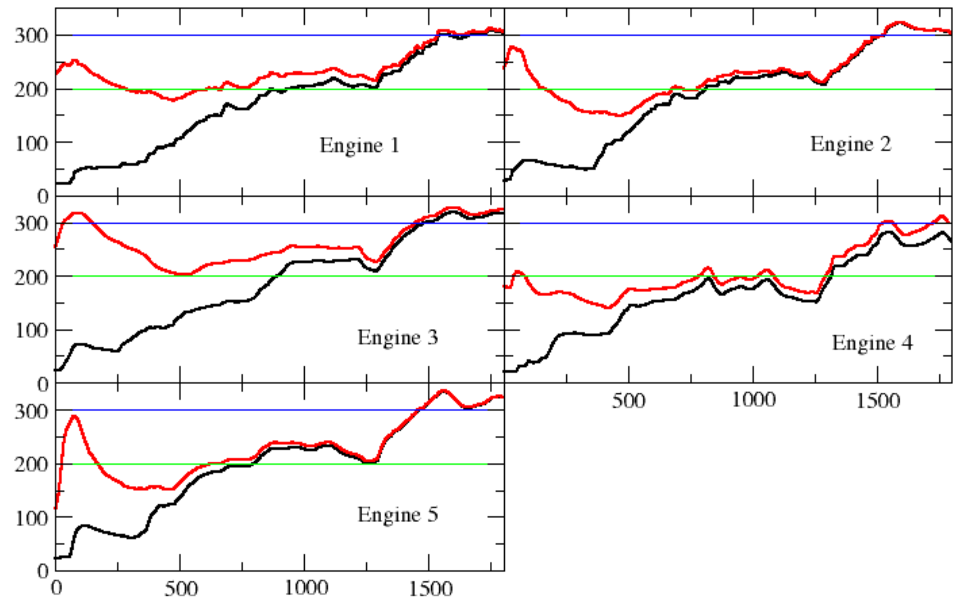

Figure 13: Post catalyst exhaust gas temperaturestraces for five different engines. Red = temperature of hot WHTC after soak period. Black is cold WHTC. Temperature in °C, time in s (horizontally).

4.3.5 *Calculation additional NO_x emission with cold start*

The additional NO_x emission during a test with cold start occurs during the period that the catalyst temperature is lower than during the test with hot start. The precise temperature level of the engines presented in Figure 13 is not very relevant, because this is likely to change for several reasons when going to Euro VI. For example EGR (cooled or non-cooled) might change the temperature level. Also the desired SCR conversion efficiency may change the exhaust temperature level (i.e. through air-system settings and/or EGR). For this reason the length of the period in which the exhaust temperature is lower with the cold start is important to obtain from the engine data and not the temperature level itself.

The following steps are taken to calculate the additional NO_x due to the cold start taking the above into account:

1. Estimate the time constant for heat-up of the exhaust aftertreatment system from the available temperature traces.
2. Calculate the required average aftertreatment system temperature in order to meet several SCR efficiencies across the WHTC. A generic SCR conversion characteristic is used (Figure 12).
3. Calculate the NO_x conversion during the WHTC with cold start based on aftertreatment system temperature according to an exponential temperature increase. The engine-out NO_x during the WHTC is taken linearly proportional with fuel consumption (Figure 11).

Ad 1. Time constant for heating up of aftertreatment system

In order to assess the heating up time, the differences in exhaust gas temperature between cold start and hot start are plotted in Figure 14. An exponential decrease curve is fitted through the post catalyst temperatures differences. These figures are also presented in Appendix A, including the pre catalyst and/or post turbine temperature differences (Figure 20 to Figure 24 in Appendix A). The time constants of the exponential curves for the five engines are presented in Table 13, page 30. The time constant, t, of the exponential curve corresponds to the time at which the temperature difference is reduced by 63%. With a time corresponding to 2t, the difference is reduced by 87% (consequently 13% remains).

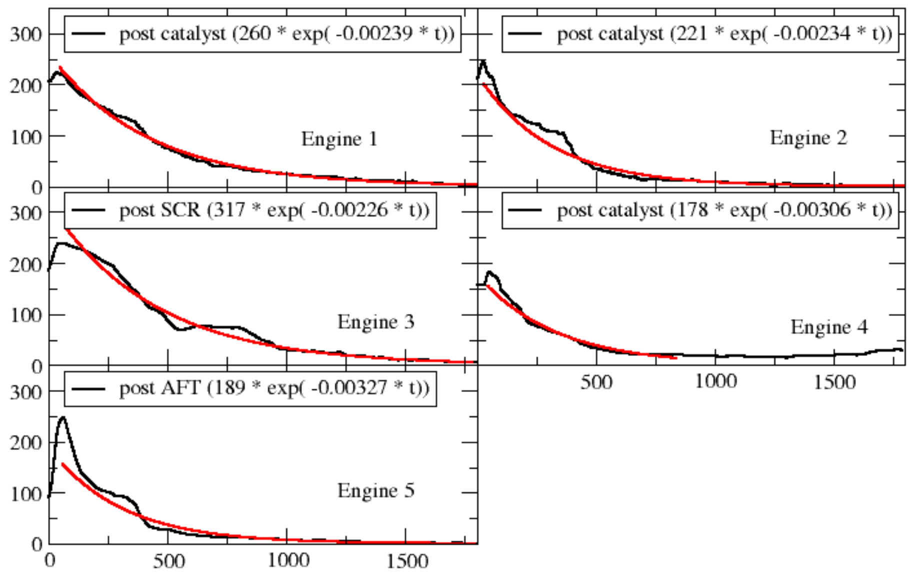

Figure 14: Post catalyst temperature difference between hot WHTC and cold WHTC for five different engines (black lines). Red lines: exponential decrease curve. Temperature in °C (vertically), time in s (horizontally).

Table 13: Time constant of the decrease of the temperature difference between hot start and cold start. Temperature downstream of aftertreatment system.

Engine	Configuration	Time constant of exponential decay (s)	
		t 63%	2 t 87%
1	SCR	420	840
2	DPF + SCR	308	616
3	DPF + SCR	442	884
4	DPF	326	652
5	DPF + SCR	306	612

Table 13 shows that the time constant for the exponential decrease of the temperature difference between the test with hot and cold start ranges between 306 s and 442 s. Three engines are somewhat above 300 s while two engines are in the 400 to 450 s time range. A relationship between the number or size of the aftertreatment system components and this time constant could not be established. The time constant for heating up of the aftertreatment system after a cold start is equal to the time constant of the decrease of the difference in temperature between the hot and cold start.

Ad 2: Aftertreatment system temperature dependency on SCR efficiency.

For Euro VI probably a combination of EGR and SCR will be used. EGR is an engine measure which reduces the engine-out NO_x, while SCR is an aftertreatment measure which reduces the tailpipe NO_x. The higher the SCR efficiency, the higher the engine-out NO_x emission level that can be allowed with the same (Euro VI) NO_x emission level result from the cycle. In Table 14, this relation is presented. The SCR efficiency is varied from 70% to 95%. As a consequence the allowed engine-out NO_x ranges from 1.33 g/kWh to 8 g/kWh. The resulting tailpipe NO_x emission level is kept at 0.4 g/kWh. Table 14 also shows the average catalyst temperature which is required to meet the SCR efficiency according to the characteristic curve presented in Figure 12.

Table 14: Engine-out NO_x and required average catalyst temperature in order to meet SCR efficiency and Euro VI emission level.

SCR efficiency	Engine-out NO_x (g/kWh)	Average catalyst temperature (°C)
70%	1.33	231
75%	1.6	235
80%	2	239
85%	2.66	243
90%	4	246
95%	8	250

Ad 3. Calculation of NO_x emission during cold WHTC

This calculation is based on:

- The engine-out NO_x (Table 14). The distribution across the WHTC is proportional to the fuel consumption. Refer to Figure 11.

- The time constant of heating up of the aftertreatment system: several time constants will be evaluated.
- The SCR characteristic of Figure 12: NO_x conversion as a function of temperature.

The additional NO_x is calculated on a second by second basis. This starts with no SCR NO_x conversion as long as the catalyst temperature is below 180°C. The NO_x conversion increases with temperature linearly between 180°C and the temperature listed in Table 14. This is also the maximum temperature used in this simulation. The calculation is schematically presented in Figure 15, for two time constants and also for two average SCR conversion levels. The additional NO_x emissions are calculated for the whole period where there is a temperature difference between the cold and the hot cycle. E.g. for a time constant of 200 s this will even be up to 1500 s.

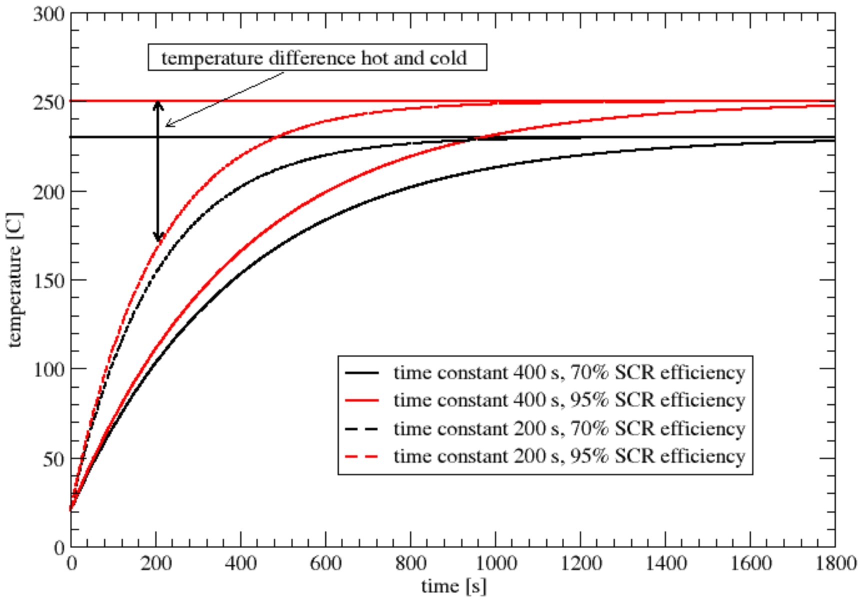

Figure 15: Schematic representation of temperature differences between WHTC with cold start and hot start with several time constants and several SCR conversion efficiencies.

The results of the calculations are presented in Table 15. The additional NO_x is calculated for the test with cold start and also the weighted result. For the calculations the SCR efficiency is varied from 70% to 95% in steps of 5%. Three time constants for the heating up of the aftertreatment systems are used:

– 400 s: this is a somewhat above the average value of the five engines evaluated. Refer to Table 13 (average is 360 s)
– 300 s: this corresponds to the minimum of the five engines evaluated
– 200 s: this is a value which possibly can be achieved after optimisation. There is no direct evidence that this is possible (for the entire aftertreatment system). On the other hand it could represent a partial heating up of the SCR catalyst (first zone) system with reasonable NO_x conversion at less than optimal temperature levels.

Table 15: Additional NO_x due to cold start WHTC as a function of SCR NOx conversion and time constant of heating up of the aftertreatment system.

	Additional NO_x due to cold start					
time constant	200 s	300 s	400 s	200 s	300 s	400 s
SCR conversion	cold test			weighted: 10% cold + 90% hot		
70%	33%	**60%**	89%	3%	**6%**	9%
75%	42%	**76%**	111%	4%	**8%**	11%
80%	54%	98%	145%	5%	10%	14%
85%	**75%**	**135%**	200%	**8%**	**13%**	20%
90%	116%	209%	310%	12%	21%	31%
95%	241%	430%	641%	24%	43%	64%

Table 15 shows a wide variation in NO_x increase due to the cold start, depending on average SCR efficiency and heat-up time constant. The additional NO_x ranges from 33% to 641% for the WHTC with cold start. This corresponds to a range of 3% to 64% for the weighted result. These results are graphically presented in Figure 16.

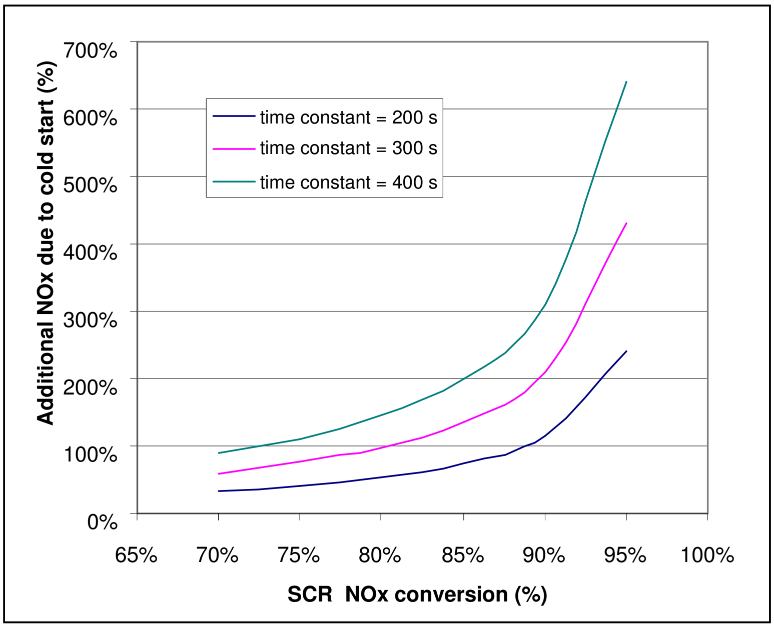

Figure 16: Additional NO_x emissions due to the cold start with WHTC

4.3.6 *Conclusion additional NO_x due to cold start*

The question is which time constants and SCR NO_x conversion values are optimal and reasonable. A time constant of 300 s may already present a development challenge, because for the not so good engines it means a reduction of about 30% in heat-up time. Also economical aspects can be taken into account such as one size catalyst which is fitted to engines with different power ratings. This can lead to a relatively large aftertreatment system with a corresponding longer heat-up period. For optimal fuel economy an SCR efficiency of more than about 85% is attractive. 85% corresponds with an average engine-out NO_x emission of 2.66 g/kWh. But of course during the

engine heat-up phase there is the option to increase the EGR level in order to maintain low NO_x emissions with reduced SCR efficiency. With a NO_x level of 1.33 to 2 g/kWh an average SCR efficiency of 70% to 80% would be sufficient. Taking into account the fairly low average power and load level during the heat-up phase, this may be possible with a not too complex EGR system.
Thermal management is another possibility. The energy flow into the exhaust system is increased during the cold start phase in order to reduce the heat-up time of the aftertreatment system. Additional energy can for example be realised by injection timing retard. This increases the exhaust gas temperature at the cost of reduced engine efficiency. Another possibility is injection of fuel into the exhaust system or post injection. This is possible, once a minimum temperature level is reached in order to secure the light-off (oxidation) of hydrocarbons within the oxidation catalyst. Post injection is not attractive for heavy-duty engines because of its negative influence on engine durability (such as lubricant dilution). Post injection and fuel injection into the exhaust system do of course lead to an increase in fuel consumption.

Taking all this into account, the following three options were defined and evaluated:

Option 1: “Best conventional”
This combines the best time constant of the engines evaluated in combination with an average SCR efficiency (85%).

Option 2: “Extra EGR”
The best time constant of the engines evaluated is combined with a low SCR efficiency and an engine-out NO_x emission of 1.33 to 2 g/kWh . This can be achieved with a high EGR rate, possibly only applied during the cold start phase when SCR is not effective.

Option 3: “Thermal management”
This combines a shorter time constant of 200 s with an average SCR efficiency of 85%. The shorter time constant would need to be achieved with thermal management.

The additional NO_x emission due to the cold start for these three options is summarised in Table 16.

Table 16: Engine configuration options and corresponding effect on additional NO_x emissions during the WHTC with cold start

Option	SCR conversion	time constant	Additional NO_x due to cold start	
	%	s	cold test	weighted: 10% cold + 90% hot
1	85%	300	135%	13%
2	70% - 75%	300	60% - 76%	6% - 8%
3	85%	200	75%	8%

Table 16 shows that for these three options, the cold start results in an additional NOx emission in the range of 60% to 135%. Corresponding numbers for the weighted test result (10% cold, 90% hot) are 6% to 13%.

4.4 Hot soak

The hot soak is also described in paragraph 3.5. The influence of the hot soak on NO_x emissions is primarily dependent on:

- The temperature of the catalysts and DPF (if upstream of SCR) at the end of the test with cold start. This is dependent on the engine exhaust temperature, exhaust pipe surface area and insulation.
- The cool-down curve of the SCR catalyst and DPF (if upstream of SCR). This is dependent on insulation of these components.
- The Light-off characteristic of the SCR catalyst.

SCR catalyst cool down:
Figure 13 shows that the post catalyst temperatures during the WHTC with both cold start and hot start (varying soak times). The figure shows that the post catalyst temperatures at the end of the WHTC with cold start are all fairly close to 300°C (also refer to Figure 18 and Figure 19 in Appendix A). Table 17 gives more precise data of temperature changes for various engines and soak times. Post catalyst temperature data is used to represent the catalyst temperature. The average catalyst temperature at the end of the WHTC with cold start for the five engines evaluated is 303°C. The temperature change as a function of soak time (from Table 17) is also presented in Figure 17. It shows that the variations between the engines are large. Of course these heat losses during stand still are strongly dependent on test set up conditions including ventilation in the test cell, insulation and relative catalyst size.

Table 17: Catalyst temperature change during the soak period for various soak times.

Engine	Soak time min	Catalyst temperature change during soak (°C)	Temperature change °C
1	10	From 310 to 250	-60
2	10	From 306 to 276	-30
3	5	From 305 to 255	-50
3	10	From 305 to 220	-85
3	20	From 305 to 185	-120
4	5	From 267 to 206	-61
5	20	From 325 to 289	-36

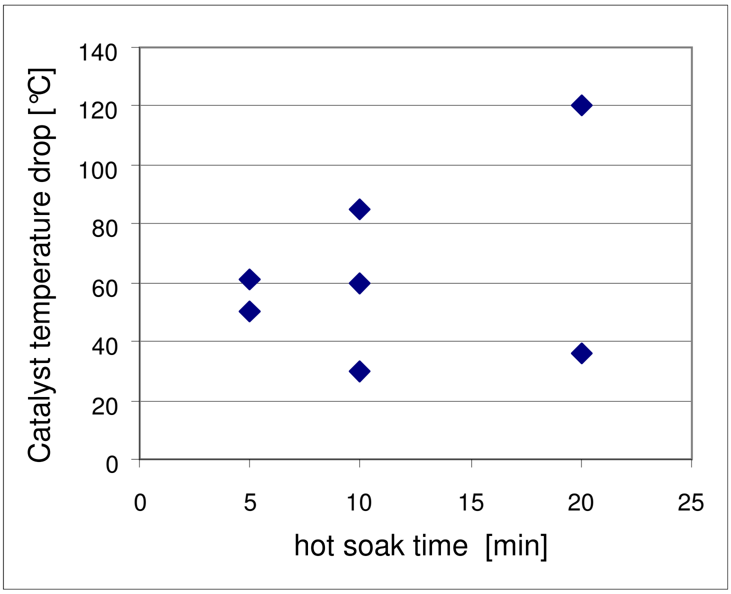

Figure 17: Catalyst temperature change during the soak period

The SCR catalyst reaches its maximum conversion efficiency at 250°C (Figure 12). This means that there is an average margin of 53°C for cooling down before SCR conversion is compromised. The average of the 3 engines with 10 minutes hot soak is 58°C, which is very close to the value of 53°C. From this is can be concluded that in general SCR conversion will not suffer from a hot soak period of 5 or 10 minutes. For some engines some optimisation might be necessary to accomplish this.

5 Proposal for correlation factors

5.1 Summary of the differences between the European and World Harmonised cycles

The transition from European to World Harmonised test cycles includes the replacement of the ETC by the WHTC and the replacement of the ESC by the WHSC.

The differences between the ETC and WHTC are threefold:
- The WHTC has a lower average cycle work than the ETC (lower average speed and load).
- The WHTC includes an additional test with a cold start while the ETC does not.
- The WHTC has a hot soak period in between the tests with cold and hot start, while the ETC starts immediately after pre-conditioning.

The ESC and WHSC are both steady-state test cycles and the differences are limited to the points within the engine map and for the WHSC the transitions (slow transient) between the engine points are included in the test result. The reason for a difference in emissions is the difference in engine cycle work, or more specifically the difference in internal friction in relation to the effective work delivered to the output shaft.

With respect to the emission components, the focus is entirely on NO_x. The other components; particulates, HC and CO can relatively simply be controlled within the expected engine configurations for Euro VI. Refer to paragraph 3.1.

The differences between the World Harmonised and the European test cycles were analysed in Chapter 4. The results for the WHTC in comparison to the ETC are presented in Table 18. The total NO_x increase of the WHTC in comparison to the ETC is estimated to be 16% in a conservative approach and 11% with a rather optimal engine configuration.

Table 18: NO_x increase in the WHTC compared to the ETC

NO_x in WHTC compared to ETC *)	Conservative	Optimised
Difference in engine cycle work	3 %	3 %
Cold start	13%	8%
Hot soak (5-10 minutes)	0 %	0 %
Total	16%	11 %

*) for composite test: WHTC with cold start, soak period and WHTC with hot start.
Weighting: 10% for cold WHTC and 90% for hot WHTC

For the correlation between the WHSC and the ESC only the difference in engine cycle work is relevant. The NO_x increase for the WHSC in comparison to the ESC is limited to 1.6%.

5.2 Proposal for correlation factors

Based on the analysis done in chapter 4, which is summarised in paragraph 5.1, the proposed correlation factors are presented in Table 19. The correlation factor is defined as a factor to be applied to the ESC and ETC emission limits in order to obtain equivalent emission limits according to the WHSC and WHTC test procedures.

Table 19 shows that all correlation factors are 1.00 except for a NO_x correlation factor of 1.10 for the WHTC. This is based on the NO_x increase of 11% under optimised condition from Table 18. For the WHSC the NO_x increase is only 1.6%, which is very close to 1.00.

Table 19: Proposed correlation factors between WHTC and ETC, respectively WHSC and ESC

Emission Component	WHTC compared to ETC *)	WHSC compared to ESC
NO_x	1.10	1.00
CO	1.00	1.00
HC	1.00	1.00
Particulates	1.00	1.00

*) for composite test: WHTC with cold start, soak period and WHTC with hot start.
Weighting: 10% for cold WHTC and 90% for hot WHTC

6 References

[Ellensohn, 2007] Ellensohn, Rudolf: OEM Euro VI Strategy and Developments in Technology to meet the Emission Standards, Diesel Emissions Conference 07 - Integer Research Frankfurt - 26/27 June 2007

[Schlick, 2007] Schlick, Thomas: Global Harmonization of Vehicle Emission Standards, Diesel Emissions Conference 07 - Integer Research Frankfurt - 26/27 June 2007

[Rickert, 2007] Rickert G.: Overview of Emissions on World-Harmonised and Non-road Cycles from the AECC Euro VI Programme, AECC Technical Seminar on Heavy-duty Engine Emissions, Brussels, 25 October 2007

[Signer, 2006] Signer, Meinrad: Future Emission Regulations for trucks: Global harmonisation and technology development, Euro V and VI Conference Amsterdam 28-29 June 2006

[Tennison, 2004] Tennison, Paul, Christine Lambert and Michael Levin: NO_x Control Development with Urea SCR on a Diesel Passenger Car, SAE paper 2004-01-1291

[Winkler, 2003] Winkler, C., Flörchinger, P., Patil, P., Gieshof, J., Spurk, P., Pfeifer, M.: Modeling of SCR DeNOx Catalyst – Looking at the Impact of Substrate Attributes, SAE 2003-01-0845

[Chandler, 2000] Chandler, G.R.: An integrated SCR and Continuously Regenerating Trap System to Meet Future NO_x and Pm Legislation, SAE 2000-01-0188

[Walker, 2003] Walker, A.: The Development and On-Road Demonstration of Four-Way Emission Control Systems for HDD Applications, Heavy Duty Diesel Emissions Control TopTec Symposium, Sweden 2003

[Spurk, 2002] Spurk, P., Pfeifer, M., Gieshoff, J., Lox, E.: A Breakthrough in SCR Technology, Auto Technology 3/2002

[van Helden, 2004] van Helden, Rinie, Ruud Verbeek, Frank Willems: Optimization of Urea SCR deNOx Systems for HD Diesel Engines, SAE 2004-01-0154

[ACEA, 2007] ACEA: Correlation ETC – WHTC (updated with additional information. May 31st 2007.

[AECC, 2007] Rickert, G.: Overview of emissions on World-Harmonised and non-road cycles from AECC Euro VI programme. AECC technical seminar on heavy-duty engine emissions, Brussels, 25 October 2007

7 Signature

Delft, 1 December 2008

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A Exhaust gas temperatures

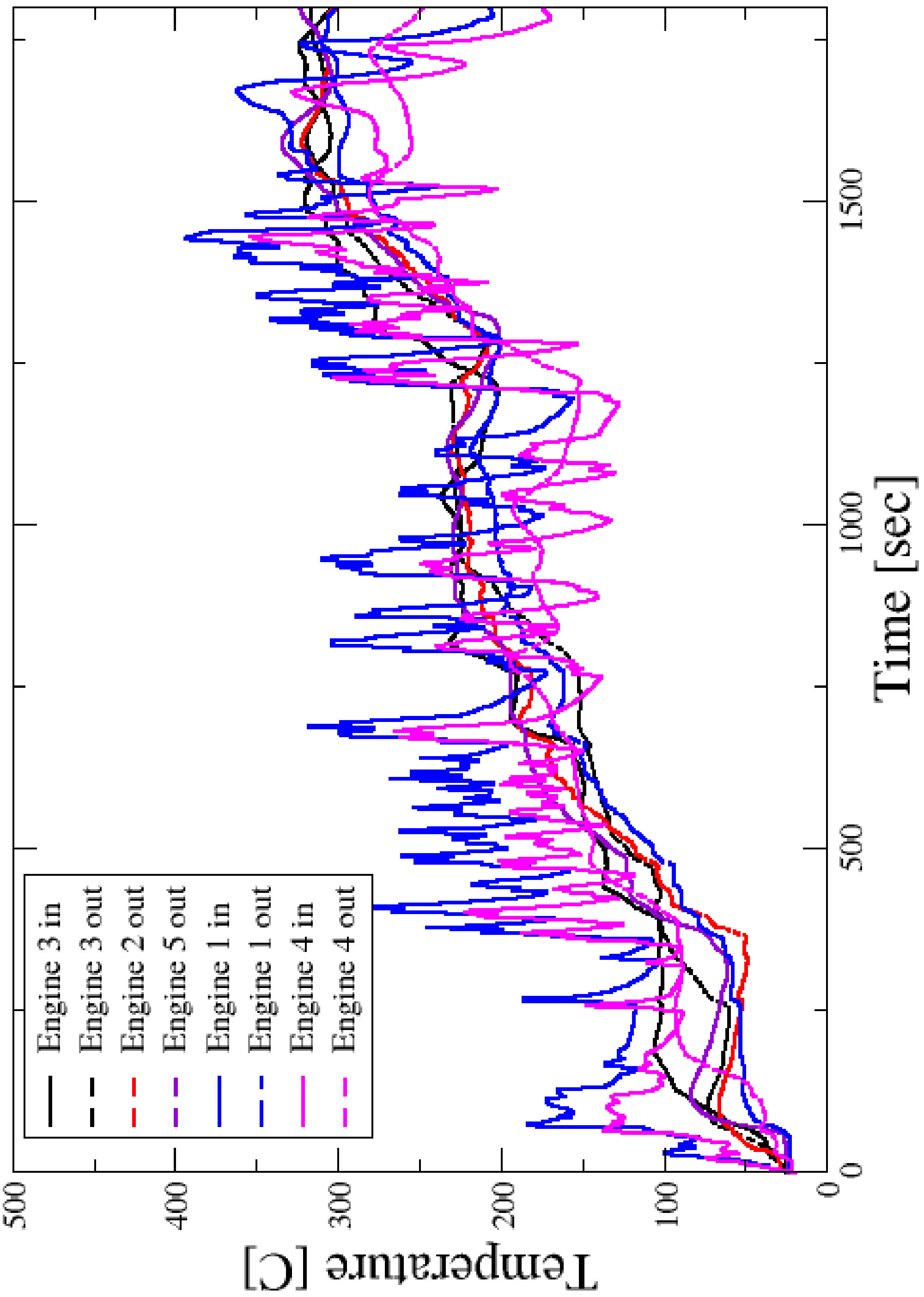

Figure 18: Pre and post catalyst exhaust gas temperatures of 5 engines

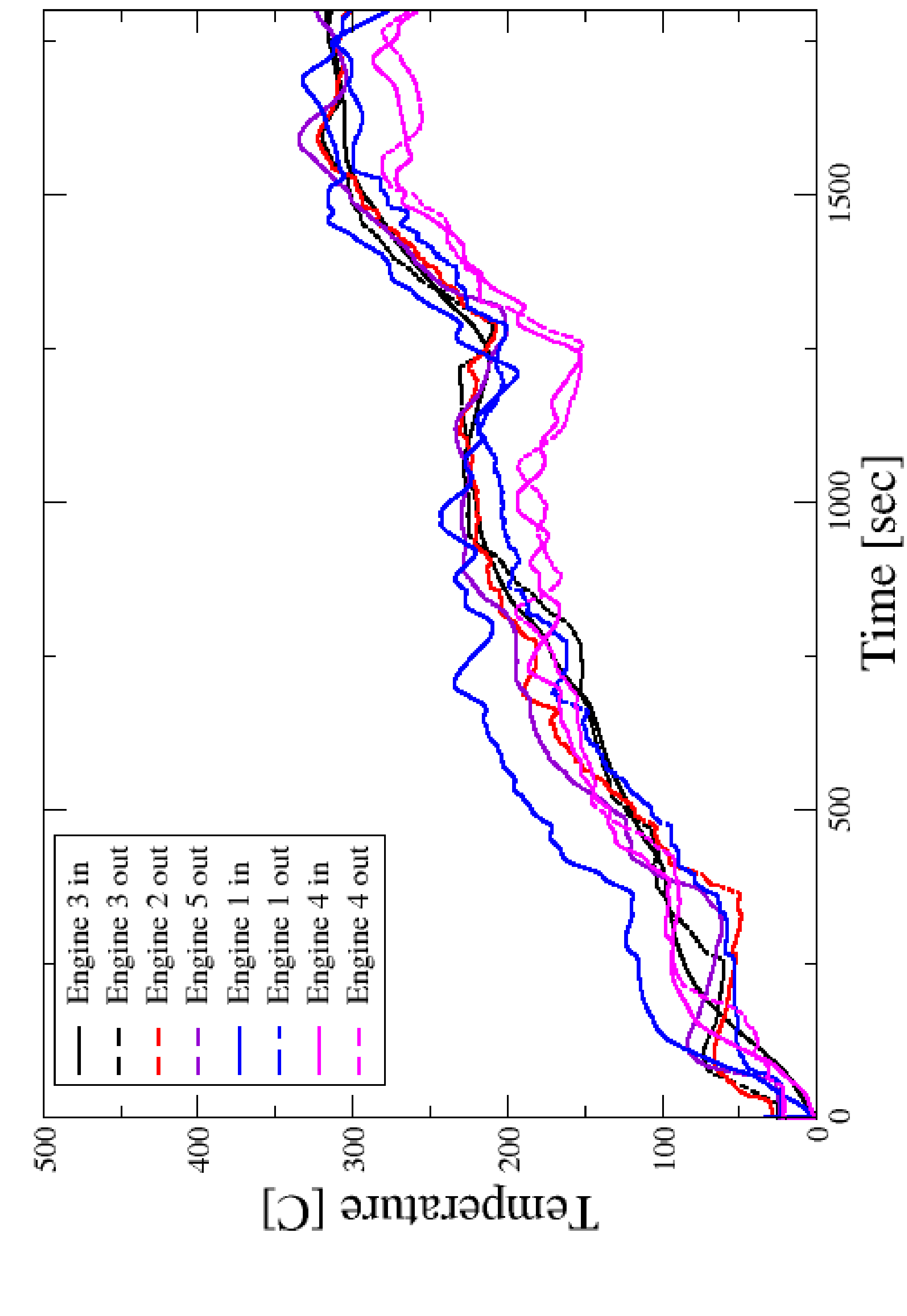

Figure 19: Pre and post catalyst exhaust gas temperatures of 5 engines. Pre catalyst with smoothing via relaxation method with decay time of 60 s.

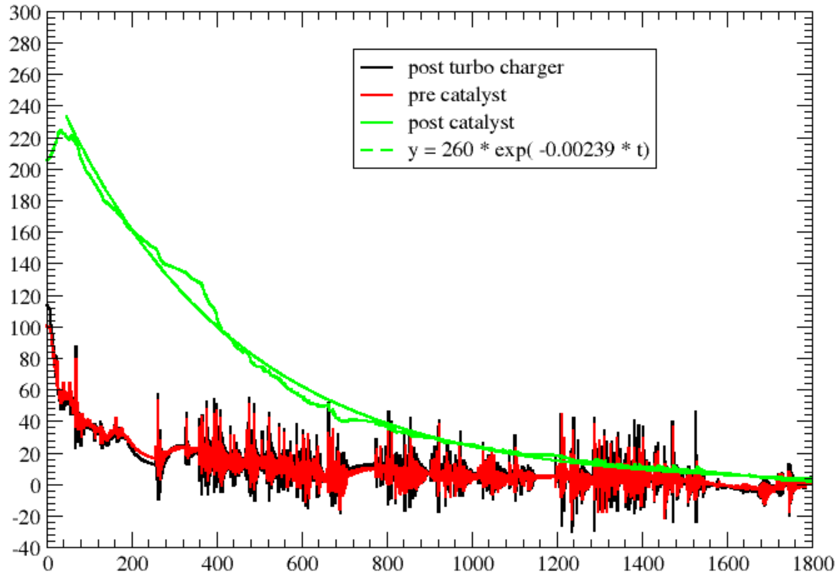

Figure 20 : Engine 1, pre and post catalyst temperature difference between hot WHTC and cold WHTC. Temperature in °C (vertically), time in s (horizontally).

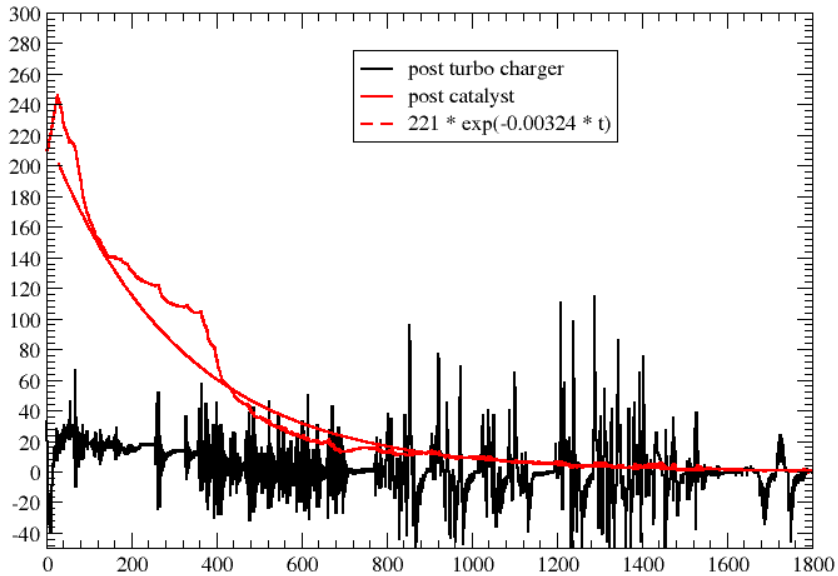

Figure 21: Engine 2, pre and post catalyst temperature difference between hot WHTC and cold WHTC. Temperature in °C (vertically), time in s (horizontally).

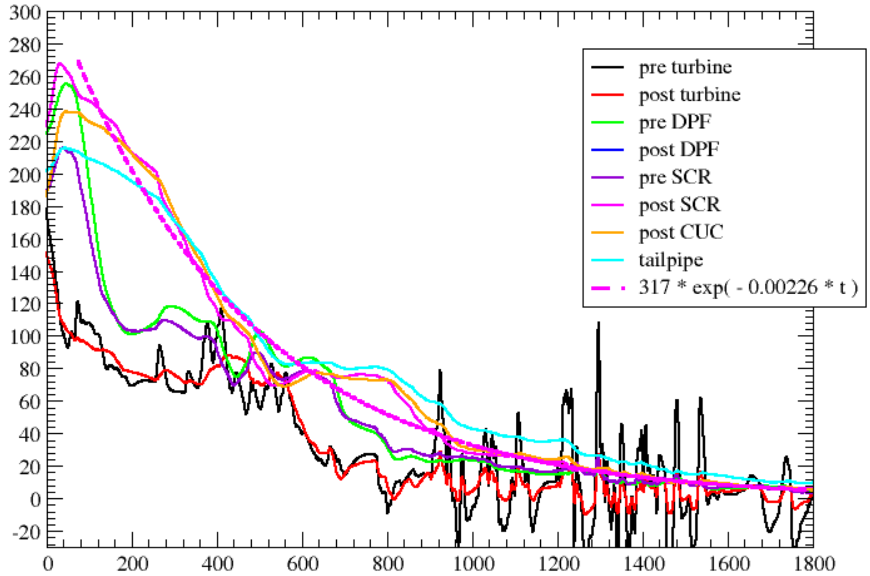

Figure 22: Engine 3, pre and post catalyst temperature difference between hot WHTC and cold WHTC. Temperature in °C (vertically), time in s (horizontally).

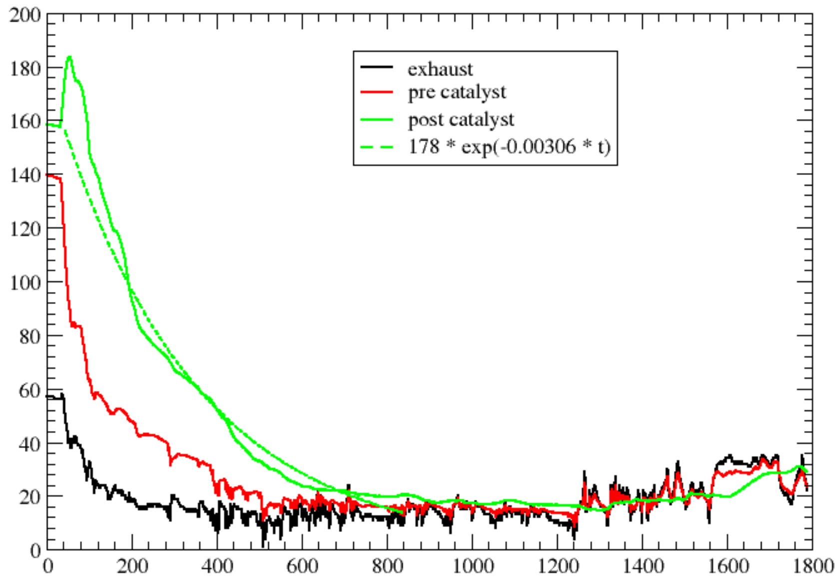

Figure 23: Engine 4, pre and post catalyst temperature difference between hot WHTC and cold WHTC. Temperature in °C (vertically), time in s (horizontally).

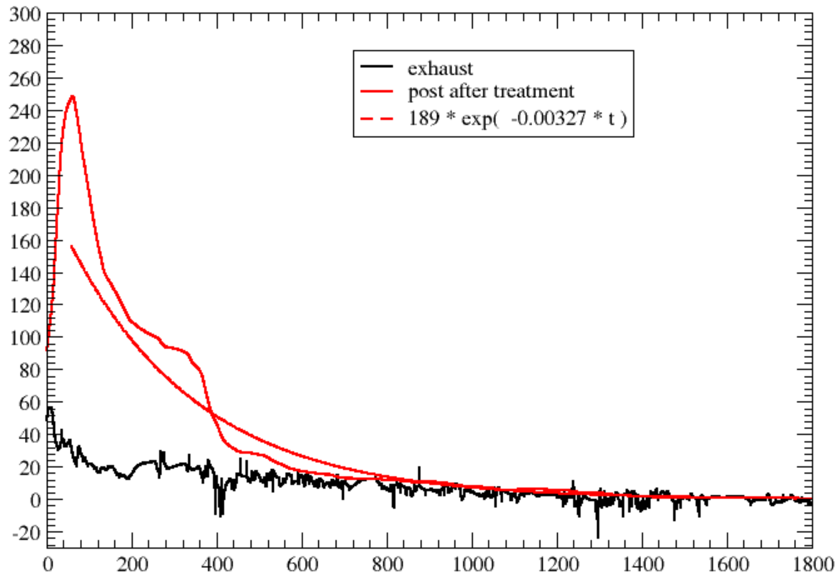

Figure 24: Engine 5, pre and post catalyst temperature difference between hot WHTC and cold WHTC. Temperature in °C (vertically), time in s (horizontally).

B Analysis friction losses

The friction losses are approached as a linear function of engine speed:

$T_{friction} = a + b \cdot n$

In which n is engine speed and a and b are constants for a particular engine. For this exercise, the friction losses are assumed to be independent of torque. The friction losses can be determined via several methods. The methods used here are via motoring curves (torque) and via Willans lines.

Motoring curves
For the engines 1 and 2 motoring curves were available. These are curves of negative torque where the dynamometer is driving the engine (zero fuel flow). The results are presented Table 20. The sample points and fit are also graphically presented in Figure 25.

Table 20: Coefficients a and b

	a	b
engine 1	36.2	0.1041
engine 2	33.0	0.0318

Willans lines
The Willans lines are straight lines of constant speed with fuel consumption as a function of torque. The intersection of the line with the torque axis gives the friction torque. The data points at the steady-state points of the WHSC and ESC are used for this assessment. This appeared to be not straightforward, primarily because the steady-state data does not cover the full engine map. The WHSC and the ESC do not cover very well low engine speeds and also low loads (0-25% load).

The fuel consumption is assumed to be a function of torque and speed, where the coefficients C and D are fitted through least square fit. In order to cope with the limited data sets, five equations were evaluated in order to predict fuel consumption (FC) as a function of speed (n) and torque (T):

$FC/n = C \cdot n + D \cdot T$ (1)
$FC/n = C \cdot (n + 500) + D \cdot T$ (2)
$FC = C + D \cdot T/n$ (3)
$FC/n = C + D \cdot T/n$ (4)
$FC = C \cdot n + D \cdot T$ (5)

From the intersection of the fuel consumption FC=0 follows the relation between the friction torque and the speed. The results are presented in Table 21. All equations except for number 2 fit constant “a” in the $T_{friction}$ equation to zero and b=C/D. This leads to stable results and looks reasonable taking into account the shape of the motoring curves. With equation 2 an offset is manually implemented. In this case the coefficient “a” equals 500 x “b” worked well. The idle fuel consumption estimation is improved at a slightly worse fit at higher speeds and loads. The overall standard deviation is marginally better than with equation 1. Refer to Table 22.

Table 21: Coefficients a and b for friction torque based on different equations

Equation ->	FC/n = n + T		FC/n = (n+500) + T		FC = 1 + T/n		FC/n = 1 + T/n		FC = n + T		based on
Coefficient ->	a	b	a	b	a	b	a	b	a	b	cycle
engine 1	0	0.074	25.9	0.052	0	0.006	0	0.044	0	0.007	WHSC
engine 2	0	0.041	15.5	0.031	0	0.013	0	0.032	0	0.020	WHSC
engine 3	0	0.045	18.2	0.036	0	0.068	0	0.050	0	0.091	ESC
engine 4	0	0.061	21.8	0.044	0	0.000	0	0.028	0	0.025	WHSC

Table 22: Standard deviation for the different fit methods

	FC/n = n + T	FC/n = (n+500) + T	FC = 1 + T/n	FC/n = 1 + T/n	FC = n + T
engine 1	8%	7%	45%	26%	24%
engine 2	10%	8%	48%	27%	25%
engine 3	10%	10%	69%	43%	33%
engine 4	8%	7%	47%	27%	24%

In Figure 25 the friction torque curve based on the motoring curves and based on the Willans lines are compared for engines 1 and 2 (the only engines for which both were available). The graph shows that for engine 2 there is a reasonable agreement between the two friction torque curves. For engine 1 this is however not the case. The reason for this difference is not very well understood.
9

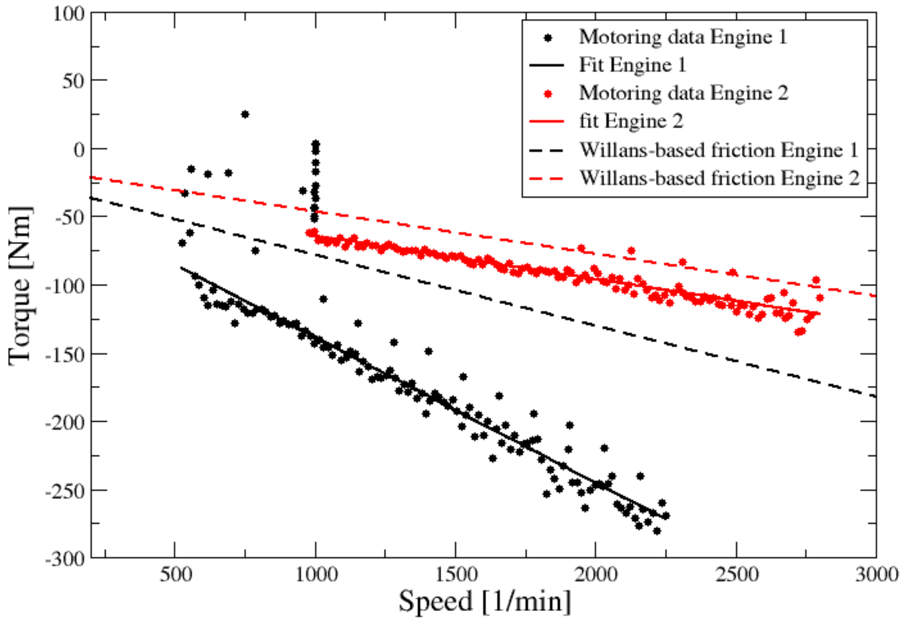

Figure 25: Friction torque curves based on motoring curves and based on Willans lines.

C Evaluation different cold start weighting factors

A comparison is made between the 10% + 90% and 14% + 86% weighting factors for respectively the cold and hot WHTC. The updated tables are given below.

Table 23: Influence of weighting factors on WHTC NO_x emission level

	Additional NOx due to cold start								
time constant	200 s	300 s	400 s	200 s	300 s	400 s	200 s	300 s	400 s
SCR conversion	cold test			weighted: 10% cold + 90% hot			weighted: 14% cold + 86% hot		
70%	33%	**60%**	89%	3%	**6%**	9%	5%	**8%**	12%
75%	42%	**76%**	111%	4%	**8%**	11%	6%	**11%**	16%
80%	54%	98%	145%	5%	10%	14%	8%	14%	20%
85%	**75%**	**135%**	200%	**8%**	**13%**	20%	**11%**	**19%**	28%
90%	116%	209%	310%	12%	21%	31%	16%	29%	43%
95%	241%	430%	641%	24%	43%	64%	34%	60%	90%

Table 24: NO_x increase in the WHTC compared to the ETC

NO_x in WHTC compared to ETC *)	Conservative	Optimised
Difference in engine cycle work	3 %	3 %
Cold start	19 %	11 %
Hot soak (5-10 minutes)	0 %	0 %
Total	22 %	14 %

*) for composite test: WHTC with cold start, soak period and WHTC with hot start.
Weighting: 14% for cold WHTC and 86% for hot WHTC

Table 25: Proposed correlation factors between WHTC and ETC, respectively WHSC and ESC

Emission Component	WHTC compared to ETC *)	WHSC compared to ESC
NO_x	1.15	1.00
CO	1.00	1.00
HC	1.00	1.00
Particulates	1.00	1.00

*) for composite test: WHTC with cold start, soak period and WHTC with hot start.
Weighting: 14% for cold WHTC and 86% for hot WHTC