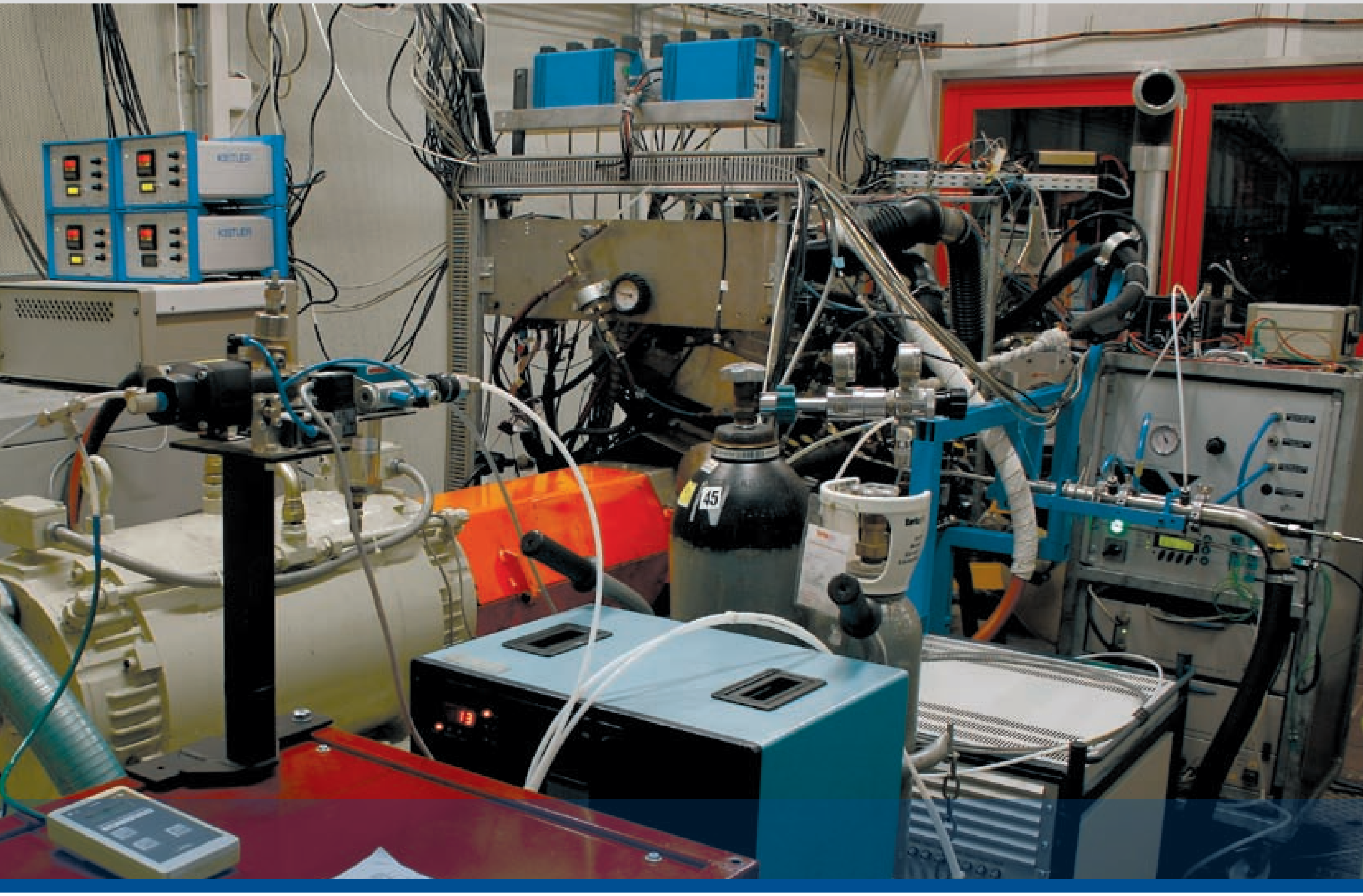

A Phenomenological Mean Value Soot Model for Transient Engine Operation

A means of characterizing the raw soot emissions from modern common-rail diesel engines is required so that their control and the regeneration of an associated particulate filter can be optimized, though no suitable sensor currently exists. Thus, under the framework of a FVV research project, a fast soot model to calculate the raw soot emissions from a common-rail diesel engine operating under both steady state and transient operating conditions was developed and validated at the Aerothermochemistry and Combustion Systems Laboratory (LAV) at the ETH Zurich.

1 Introduction and Problem Definition

Diesel soot emissions are known to have negative health and environmental impacts, requiring that their emission be reduced. Exhaust stream particulate filters have proven to be an effective means of reducing the tailpipe soot emissions, though their regeneration is coupled with an increase in fuel consumption. To reduce the fuel consumption penalty, it is necessary to optimize the filter regeneration strategy, which requires accurate knowledge of the instantaneous and integral raw soot emissions.

As physical soot sensors are not yet feasible for production engine applications, a fast – and ideally – realtime soot model was developed in this FVV research project (No. 855) to determine the raw soot emissions from a common-rail diesel engine operating under steady state and transient conditions. This article describes the development of a mean value soot model and the associated steady state and transient soot measurements used for its validation.

2 Testbench

For the parameterization and validation of the mean value soot model, the raw soot emissions were measured from two different engines operating with different fuels. The details of the two engines can be found in **Table 1**. Engine one was used for steady state and transient measurements, while engine two was used only for steady state measurements. The details of the investigated fuels can be found in **Table 2**.

The soot emissions from engine one were measured using an AVL Micro Soot

Table 1: Specifications of the two test engines

Parameter	Engine 1	Engine 2	Units
Number of Cylinders	4	4	–
Displacement	2.15	1.97	l
Compression Ratio	19:1	16:5	–
Stroke	88.4	95.5	mm
Bore	88	81	mm
Number of Valves	4	3 (2 Intake, 1 Exhaust)	–
Control System	EDC 15	EDC 16	–
Turbocharger	VTG	VTG	–
EGR	Yes	Yes	–
Max. Injection Pressure	1350	1800	bar
Soot Instrumentation	AVL Micro Soot Sensor	Filter Smoke Number	–
Application	Steady-State / Transient	Steady-State	–

Table 2: Comparison of the three implemented fuels; the reference fuel corresponds to a regular diesel fuel

Parameter	Reference	Fuel 2	Fuel 3	Units
Density	829	776	833	kg/m^3
Evaporation Temperature	336	226	347	°C
Cetane Number	51	43	54	–
Sulfur Content	9.5	< 10.0	8.0	mg/kg
Aromatic Content (Mass)	18.6	1.9	18.5	%
Viscosity (at 40 °C)	2.33	1.07	2.60	mm^2/s
Application	Engine 1 (stat./trans.)	Engine 1 (stat.)	Engine 2 (stat.)	–

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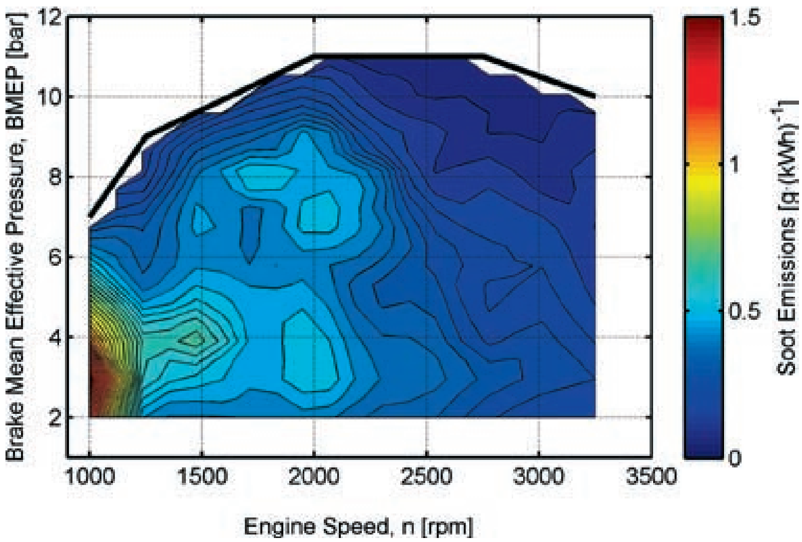

Figure 1: Measured steady-state soot emissions map (engine one, reference fuel)

Sensor (photoacoustic measurement principle [1]) coupled with a Dekati Fine Particle Sampler. For the transient measurements the gas transport times (approximately 2.5 s) and dynamic response (T = 0.5 s) of the soot measurement system were characterized, similar to [2]. The steady soot emissions from engine two were measured using a Filter Smoke Number (FSN) system (AVL 415) and converted to a soot concentration using an empirical correlation [3] so that they could be compared to the calculated soot values.

3 Mean Value Soot Model

The mean value soot model was developed through the reduction of an existing crank angle resolved soot model [4, 5] to one with a combustion cycle resolution. Because of this reduction, the mean value model no longer considers temporally resolved cylinder pressures, injection rates, or mass fraction burned profiles as model inputs, but rather their representative average values. Consequently, the calculation time of the model has been reduced from 1 s to 10 ms per combustion cycle. The general form of the mean value soot model is given in Eq. (1), where * refers to a representative average value [6]. The soot formation process is assumed to take place at the local flame temperature at a pressure representative of that during combustion. The detailed soot formation chemistry is simplified and described using a soot formation function, f, which considers a representative air fuel ratio and formation temperature [7]. The oxidation process is described using a characteristic pressure and global mixture temperature during expansion. Using these representative state parameters and parameters from the engine control unit, it is possible to determine the cycle averaged soot formation and oxidation rates.

Rather than integrating the soot formation and oxidation processes over an entire combustion cycle, as is done in the crank angle resolved model, the mean value model considers a representative soot formation and oxidation duration. The soot formation duration is assumed to be given by the diffusion combustion duration (taken to be proportional to the main injection duration), while the oxidation duration is taken as the time from the beginning of the diffusion combustion to the time at which the temperature in the combustion chamber falls below a specified value. Using these representative formation and oxidation rates and durations, the cycle resolved soot emissions can be determined.

The developed mean value soot model contains 16 parameters to describe engine, combustion and fuel parameters and must be determined for every new fuel and engine combination. The mean value soot model was parameterized using evolutionary algorithms with the goal of maximizing the correlation between the calculated and measured soot emissions [8, 9].

4 Steady State Validation

The mean value soot model was parameterized and validated based on the soot measurements from the two aforementioned engines using different fuels. Shown in **Figure 1** is the steady state soot emissions map for engine one operating with the reference fuel. **Figure 2** shows the operating point specific soot emissions from engine two (fuel three) for variations in the engine speed, load, in-

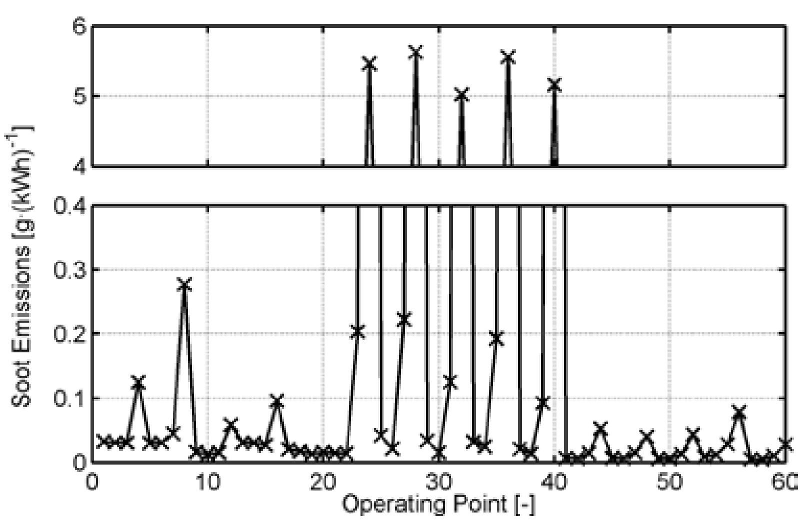

Figure 2: Measured steady-state soot emissions during parameter variations (engine two, fuel three)

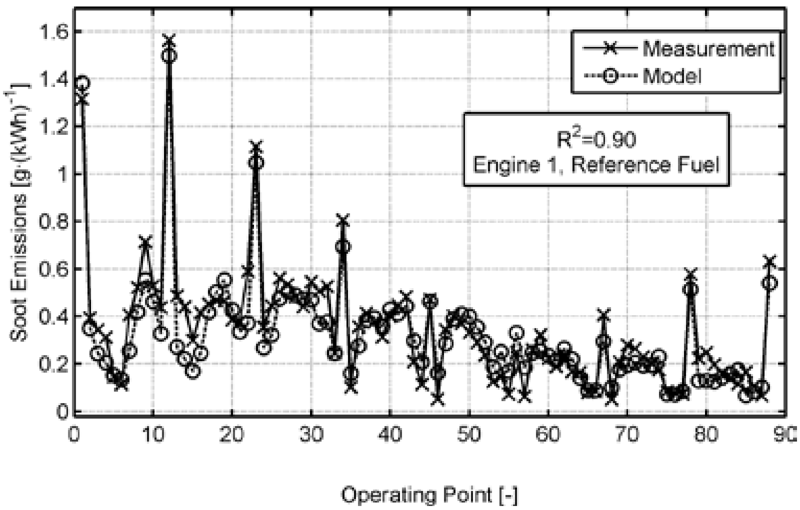

Figure 3: Comparison of the measured soot emissions and those calculated using the parameterized mean value soot model (engine one, reference fuel)

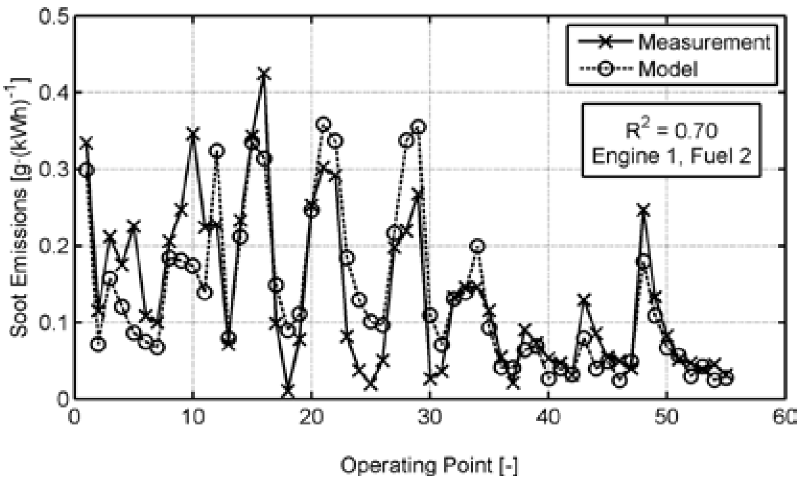

Figure 4: Comparison of the measured soot emissions and those calculated using the parameterized mean value soot model (engine one, fuel two)

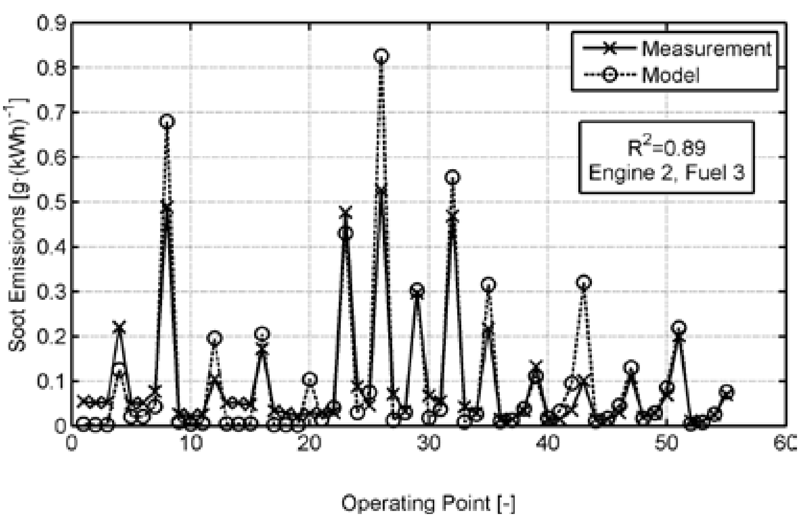

Figure 5: Comparison of the measured soot emissions and those calculated using the parameterized mean value soot model (engine two, fuel three); the operating points with extremely high soot emissions in Figure 2 are not included

jection timing, injection pressure, pre-injection fuel quantity, and EGR rate. The operating points with extremely high soot emissions shown in Figure 2 correspond to operating points at 2000 rpm with 40 % EGR. While these points are not representative of typical operating points, they are used to investigate the capability of the model to describe the soot emissions in extreme cases.

The basis parameterization of the mean value soot model was carried out for engine one with the reference fuel, for which all model parameters were optimized using evolutionary algorithms. To consider operation of engine one with fuel two, only the (eight) model parameters influenced by the fuel were varied, as the engine as such was not changed. The capability of the model to calculate the soot emissions for engine one with the reference fuel and fuel two are shown in **Figure 3** and **Figure 4**, respectively, where it can be seen that the model can reproduce the quantitative and qualitative soot emissions trends for operation with both fuels. The somewhat reduced performance of the mean value model for operation with the second fuel can likely be attributed to subtle changes in the combustion process itself (premixed combustion fraction, for example) which are difficult to predict without the benefit of in-cylinder pressure measurements.

To parameterize the model for engine two, all model parameters were once again optimized as both a new engine and fuel were being considered. The operating points with extremely high soot emissions in Figure 2 were not considered for the model parameterization, the results of which are shown in **Figure 5**. If the extreme operating points are considered, the correlation between the measured and calculated soot emissions drops to R^2 = 0.79, which indicates that the model is capable of reproducing the soot emission trends for operating points which were not considered during the parameterization.

5 Transient Soot Emission Measurement

In order to validate the capability of the mean value soot model to reproduce transient soot emissions, the soot emis-

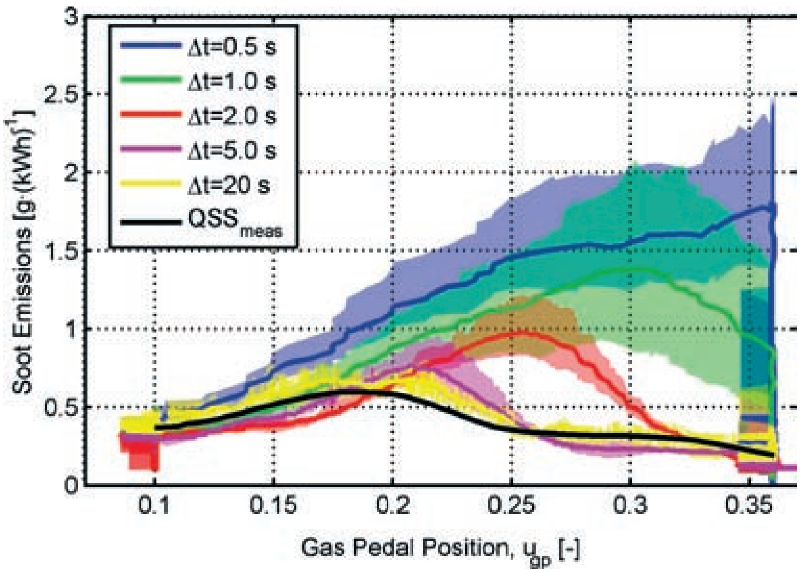

Figure 6: Measured soot emissions during tip-in transients at 1250 rpm and their respective steady-state values (QSS); the shaded areas represent the uncertainty in the measured soot emissions

sions were measured during tip-in and acceleration transients. An acceleration transient refers to an increase in the engine speed at a constant load (1250 rpm to 3000 rpm at 7 bar BMEP, with varying transient durations, Δt), while a tip-in transient refers to an increase in the engine load at a constant engine speed (1.5 to 7.5 bar BMEP at 1250 and 2000 rpm, with varying transient durations, Δt). Prior to presenting transient model validation, the measured soot emissions, oxygen availability and intake temperature measured during the transients will be compared with steady-state values using a Quasi Steady State Approximation (QSS) to gain understanding of the factors influencing the transient soot emissions.

The QSS approximation provides a tool by which parameters (for example soot emissions or lambda) during transient and steady state operation can be directly compared [10]. At each time point during a measured transient, the engine speed and load are used to interpolate the corresponding steady state parameter value from its steady state map. It should be noted that the QSS approximation bases this comparison only on the engine speed and load, and hence changes in other engine operating parameters are not considered.

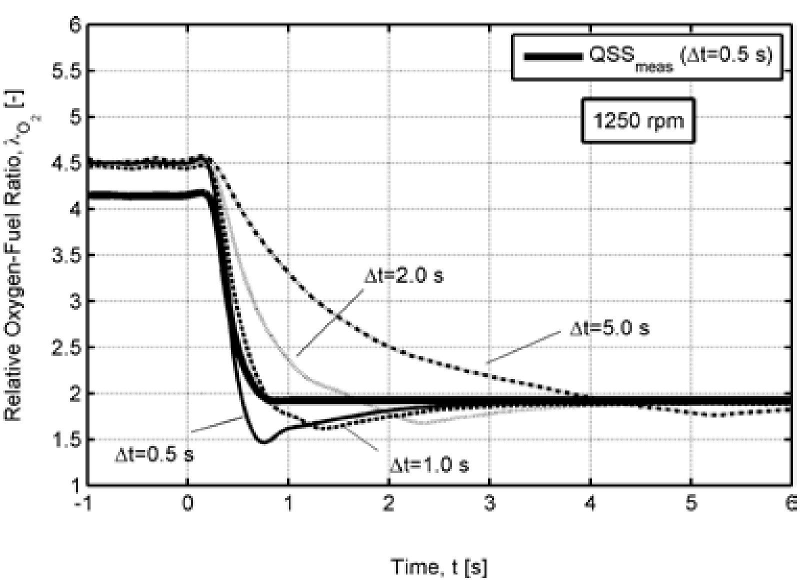

Figure 7: Relative oxygen availability (Eq. (2)) during the tip-in transients at 1250 rpm and corresponding steady-state values (QSS) for the 0.5 s transient

5.1 Important Parameters for Transient Soot Emissions

From the steady-state investigations and a sensitivity analysis, the oxygen availability and intake temperature were found to have a strong influence on the calculated soot emissions. Thus, the determination of these parameters during the measured transients was given particular attention.

The relative oxygen availability, λ_{O2}, in the combustion chamber is defined by the EGR Rate, the intake charge pressure, and the injected fuel quantity as defined in Eq. (2) [11]. However, in order to determine the relative oxygen availability during a transient, temporally resolved knowledge of the EGR rate is required. This was determined using the intake CO_2 concentration measured using a fast mass spectrometer (V&F airsense.net) and the exhaust stream oxygen concentration measured using a lambda sensor.

The mixture temperature at intake valve closing is used by the mean value model to characterize the overall thermodynamic state in the combustion chamber. Among other things, this temperature influences the fuel evaporation, wall heat transfer, premixed combustion fraction, and soot formation and oxidation processes. In general, an increase in the intake valve closing temperature corresponds to an increase in the soot emissions. During steady state operation, it is possible to estimate the temperature at intake valve closing based on the temperature measured in the intake manifold, though under transient operation this measurement is too slow. The intake temperature is therefore estimated using the ideal gas law and an engine specific correction factor given in Eq. (3) [11]. With the exception of the EGR temperature all parameters in Eq. (3) can be measured sufficiently fast and are generally available from the engine control unit.

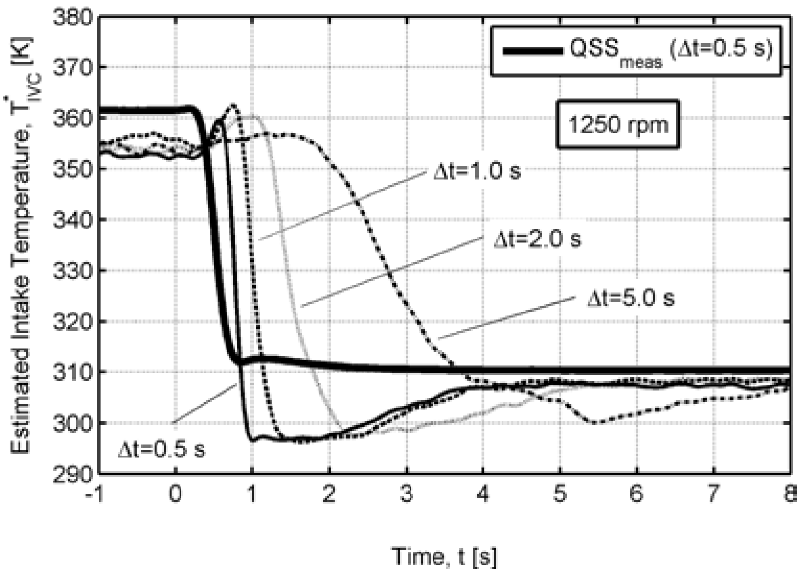

Figure 8: Estimated temperature at intake valve closing during the tip-in transients at 1250 rpm, compared to the corresponding steady-state values (QSS) for the 0.5 s transient

5.2 Measured Transient Soot Emissions

When the measured soot emissions during a tip-in transient at 1250 rpm are compared with their respective steady-state values (QSS), as shown in **Figure 6**, an increase in the soot emissions, particularly during fast transients is seen. The phenomena is also seen for tip-in transients at 2000 rpm, though the increase in soot emissions is not as extreme. Furthermore, for the investigated acceleration transients no significant increase in the soot emissions could be seen during transient operation in comparison to steady-state operation. For this reason, the focus of the discussion presented here will lie on the tip-in transients.

During the tip-in transients, a short term deficit in the relative oxygen availability compared with steady state operation was identified, which, as shown in **Figure 7** for the 1250 rpm transients, becomes more severe with decreasing transient duration. The oxygen deficit is attributed to a slow closing of the EGR valve and slow buildup of the intake charge pressure, with a simultaneous increase in the injected fuel quantity. The tip-in transients at 2000 rpm also have a slight oxygen deficit, though because of the overall higher charge pressure before and after the transient, it is not as severe. It is assumed that the increased soot emissions during the tip-in transients compared to steady-state operation can be attributed to the oxygen deficit, which both promotes soot formation and inhibits its oxidation.

If the temperature at intake valve closing during the transients is compared with the steady state temperatures, as shown in **Figure 8**, a brief increase in the temperature is seen at the beginning of the transient. After the transient however, the temperature is lower than and only slowly returns to the steady-state value. This decreased temperature results in a decrease in the soot emissions compared to steady-state operation and is discussed in further detail below (see Figure 10, for example).

6 Transient Validation of the Mean Value Soot Model

For the implementation of the mean value model for transient operation, no changes were made to the model itself or its parameterization. However, the temporally resolved measured EGR rate and estimated intake valve closing temperature replaced the slow measurements used during the steady-state model validation. The estimated intake valve closing temperature was corrected to compensate for the slow EGR temperature measurement through the use of a weighted average of the steady-state intake temperature (QSS) and the intake valve closing temperature calculated based on transient measurements, as described in Eq. (4) [11]. It was found that a weighting factor of 0.3 is required in order guarantee adequate model performance for all investigated transients.

To evaluate the ability of the mean value soot model to predict the increase in soot emissions during transient operation compared to steady state operation, the steady state calculated soot emissions (QSS) are compared with those cal-

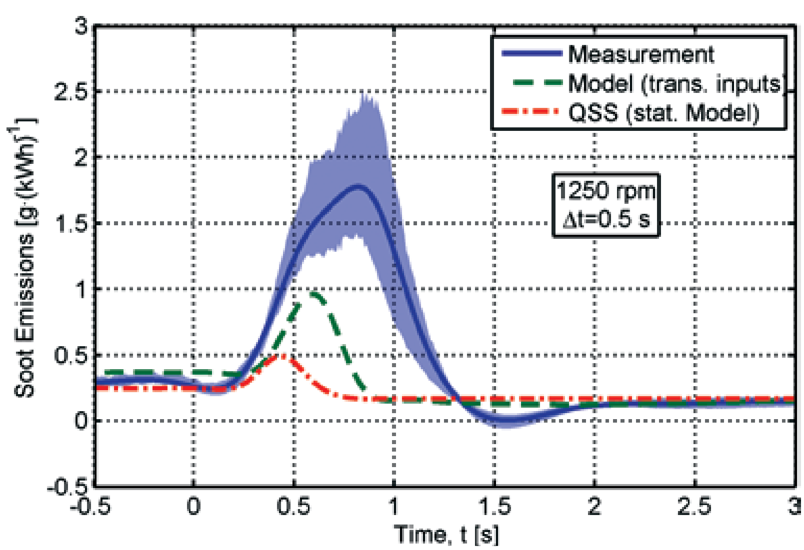

Figure 9: Comparison of the measured soot emissions and those calculated using the mean value soot model with temporally resolved inputs and its QSS approximation; tip-in transient at 1250 rpm, Δt = 0.5 s

culated using temporally resolved model inputs and the measured transient soot emissions. This comparison is shown in **Figure 9** and **Figure 10** for the 0.5 s tip-in transients at 1250 rpm and 2000 rpm, respectively. In both cases, the model with temporally resolved inputs provides an improvement compared to its steady-state implementation (QSS). For the 1250 rpm transient, the peak soot emissions are under-predicted, while at 2000 rpm they are over-predicted.

Additionally, at 2000 rpm, the model was able to reproduce further slow soot emission phenomena. As shown in Figure 10, the measured soot emissions immediately after the transient are lower than and only slowly increase to their corresponding steady state value. When temporally resolved inputs are used, the model is able to reproduce this phenomena attributed to the short term reduction in intake valve closing temperature.

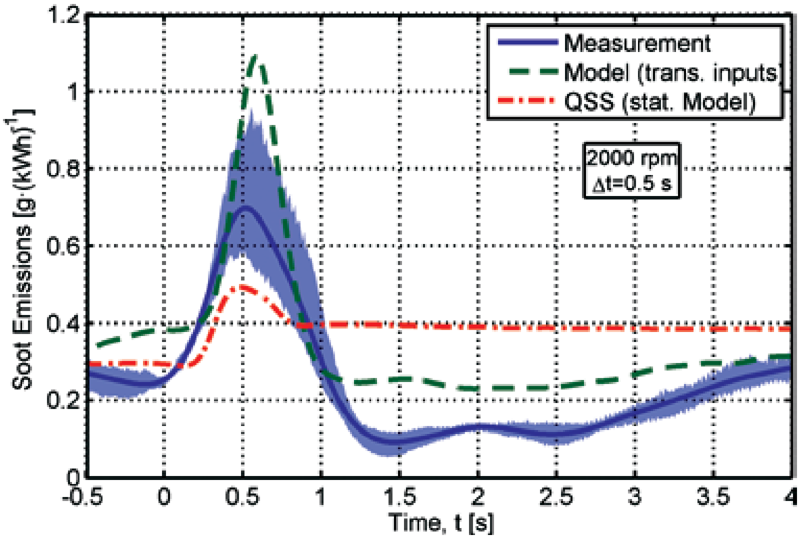

Figure 10: Comparison of the measured soot emissions and those calculated using the mean value soot model with temporally resolved inputs and its QSS approximation; tip-in transient at 2000 rpm, Δt = 0.5 s

7 Summary and Outlook

A mean value soot model was developed and validated by the Aerothermochemistry and Combustion Systems Laboratory at the ETH Zurich (Switzerland) to calculate the steady-state and transient soot emissions from a common-rail diesel engine in real time (10 ms calculation time per operating point), using only parameters available from the engine control unit. The model was parameterized for various different engine-fuel combinations using evolutionary algorithms, after which the model was capable of calculating the soot emissions for each of the respective engines and fuels.

It was found that the EGR rate, the intake charge pressure, and the temperature at intake valve closing, which is taken as representative for the thermodynamic state in the combustion chamber during combustion, must be well characterized for acceptable model performance. As such, the intake valve closing temperature was estimated for transient operation, while the EGR rate was measured using fast instruments.

From the measured soot emissions during tip-in transients, it was found that the soot emissions are higher than during steady-state operation due to a short-term oxygen deficit. Additionally, due to a reduced intake valve closing temperature, lower soot emissions compared to steady-state operation are seen immediately after the transient. No significant difference was noted between the steady-state soot emissions and those measured during the considered acceleration transients.

Equations

$$m_s = \frac{dm_{s,form}}{dt}\Big|^{*} \cdot \tau_{form} - \frac{dm_{s,ox}}{dt}\Big|^{*} \cdot \tau_{ox} \qquad \text{Eq. (1)}$$

$$\frac{dm_{s,form}}{dt}\Big|^{*} = A_{form} \cdot \left(\frac{m_{fuel\,\zeta diff}}{\tau^{*}_{char}}\right) \cdot \left(\frac{p^{*}_{form}}{p_{ref}}\right)^{n_1} \cdot f_{soot}\left(T^{*}_{form}, \lambda_{form}\right)$$

$$\frac{dm_{s,ox}}{dt}\Big|^{*} = A_{ox}\,\frac{1}{\tau^{*}_{char}}\, m_{s,form} \left(\frac{p^{*}_{O_2}}{p_{O_2,ref}}\right)^{n_4} exp\left(-\frac{T_{A,ox}}{T^{*}_{ox}}\right)$$

$$\tau_{form} = \tau_{diff} = C_{diff} \cdot \tau_{inj}$$

$$\tau_{ox} = \frac{\varphi_{freeze} - \varphi_{b,diff}}{6 \cdot n}$$

$$\lambda_{O_2} = \frac{m_{O_2} / m_{fuel}}{\left(m_{O_2} / m_{fuel}\right)_{stoich}} = \frac{Y_{O_2} \cdot \frac{p_{int} V_{IVC}}{R_u T_{int}} \cdot M_{O_2} / m_{fuel}}{\left(m_{O_2} / m_{fuel}\right)_{stoich}} \qquad \text{Eq. (2)}$$

$$T^{*}_{IVC} = \frac{p_{IVC} \cdot V_{IVC}}{R_m \cdot m_{cyl,IVC}} \cdot f_{corr}\left(p_{IVC}, T_{EGR}, n\right)$$

$$m_{cyl,IVC} = m_{air}\left(1 + \frac{EGR}{1 + EGR}\right) \qquad \text{Eq. (3)}$$

$$T^{*}_{IVC,est} = \zeta \cdot T^{stat}_{IVC,est} + \left(1 - \zeta\right) T^{trans}_{IVC,est} \qquad \text{Eq. (4)}$$

Given that accurate, cycle resolved estimates or measurements of the model inputs, in particular the EGR rate, intake charge pressure, and intake valve closing temperature are available, the model is capable of also reproducing the soot emissions trends during transient operation. There are however, still differences between the measured and calculated soot emissions, likely due to differences in the combustion and soot processes which cannot be adequately captured by the model. So that the model can consider these differences, further information of the in-cylinder processes through pressure measurements is required.

In a subsequent FVV Project (No M1107) the mean value soot model will, among other things, be further developed. In particular, in-cylinder pressure measurements will be used to characterize the combustion process such that they can be appropriately considered. In addition, optical measurements of the in-cylinder soot concentration and temperature will be used to further understand the soot formation and oxidation processes, as well as validate the model's description thereof.

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