

Improvement of SiC DPF Control Strategies for Uncontrolled Regenerations with the Aid of Quasi 3D DPF Model

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ABSTRACT

With more and more stringent standards concerning soot mass and number at the tailpipe of Diesel vehicles, the control of Diesel Particulate Filter regenerations is becoming a key feature for car manufacturers. When regulations on particle number will appear, even low cracking of the DPF could lead to the non-conformity of the filter with respect to the regulations. Strategies of regeneration control are often developed for the most critical regeneration conditions for DPF durability, the so-called "Back to Idle" tests. The development of those strategies is time-consuming, since the influence of the engine tuning on regeneration control means to load DPF for each engine tuning tested. Moreover, as the best engine tuning strategy is determined relatively to other strategies results, one needs very repeatable tests conditions and soot loading in terms of distribution and soot reactivity. This repeatability is very hard to get in real conditions. In this study, we used both experiments on a diesel vehicle with empty DPF and a 3D DPF regeneration model to simulate the effect of several engine tuning strategies to decrease the maximum internal temperature in the worse "Back to Idle" conditions.

INTRODUCTION

With more and more stringent standards concerning soot mass at the tailpipe of diesel vehicles, DPF is becoming mandatory on diesel vehicles. If standards on particle number appear in the future, risks concerning the cracking of DPF during regeneration will be a key feature in the design process of filters, since the filtration efficiency strongly decreases above a certain volume of cracks, leading to the non-conformity of the filter regarding to regulations. Those cracks occur during the regeneration phase of the filter at high temperatures, due to thermo-mechanical constraints in the substrate. Therefore, one must pay attention to the maximum soot mass loaded in the filter (Soot Mass Limit) to avoid reaching high

temperatures during DPF regeneration. SML is widely determined on so-called "Back to Idle" tests, the most severe conditions for the DPF: engine is turned back to idle, so that the energy from the combustion of the soot is no more evacuated by the exhaust mass flow, leading to very high temperatures in the filter.

One objective of car manufacturers is to increase SML, to gain time between two consecutive regenerations and so over-consumption and oil dilution or to reduce costs by decreasing the DPF volume. SML is then a major issue for the DPF performance and influences the whole control strategies, to prevent DPF from overheating and then to ensure DPF durability over its lifetime.

Two different substrates can be used for DPF: SiC or Cordierite. These two materials have very different properties, as for heat capacity and thermal stress resistance that could lead to different control strategies. This study is focused on SiC substrates.

Uncontrolled regenerations have been the topic of many studies as uncontrolled regeneration may lead to material irreversible damages like cracking or melting [1-5].

P. Florchinger *et al.* [1] have worked on a strategy to avoid uncontrolled regeneration during idle for cordierite filters. They find a strategy of control: introducing strong EGR rate (without engine instabilities) for O₂ rate minimization and increasing the idle speed to have more airflow in the exhaust line.

G. C. Koltsakis *et al.* [2] used a 3D DPF model to develop a strategy for peak temperature limitation in DPF: they concluded on the necessity to decrease O₂ rate while maximizing the air flow rate in order to protect DPF from overheating.

R. Zhan, *et al.* [3] studied three situations leading to uncontrolled regeneration with Cordierite filters. For

the “Back to Idle” case, they concluded to the necessity to decrease the O_2 rate by increasing the EGR rate.

Finally, D. Ricoult [4] worked with cordierite substrate and also demonstrated that controlling the O_2 rate by EGR and intake throttle is an efficient way of controlling the peak temperature in the DPF during uncontrolled regeneration.

Mizutani *et al.* [5] came to the conclusion that a low gas flow rate and a high O_2 concentration bring to a high maximum temperature, based on simulation.

From a previous study, we determined the worse conditions for a “Back to Idle” test, leading to an over-temperature in the DPF during regeneration. The purpose of this work is to tune engine strategies to decrease the maximum internal temperature in the worse “Back to Idle” conditions, in order to increase the SML at constant volume, or decrease the DPF volume at constant initial soot mass.

As this study would have been time and cost consuming with a classical experimental campaign, we developed a coupled experimental and numerical methodology, based on specific thermal measures at the DPF inlet and on the use of a 3D DPF regeneration model.

DPF MODELING

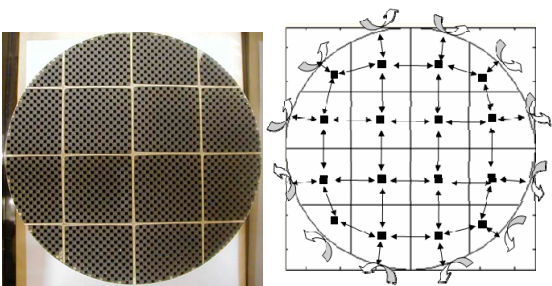
The model we used in the present study is a quasi 3D model of segmented DPF, written in C and running in a Matlab environment.

The model is a set of 1D-model interacting directly through thermal conduction, as it has been done for axi-symetrical modeling by other authors [5, 6] and 3D modeling by Mizutani et al. [7].

NUMBER OF REPRESENTATIVE CHANNELS

The set of channels was chosen regarding to the DPF macro geometry. As the real filter is a segmented filter, made of 16 segments, we chose to model it as a set of 16 representative channels, the behavior of each channel accounting for the one of a whole segment (Figure 1).

Even if the code allows the use of a greater number of channels, 16 has been chosen as it is a good compromise between calculation time and accuracy. Moreover, this assumption is supported by the high thermal conductivity of the material, which lies between 30 to 150 W/m/K depending on the temperature and is considered to lead to the homogenization of the temperature at a short length of the filter.



- Representative channel
- ↔ Thermal conduction
- ↶ Thermal loss through convection & radiation

Figure 1: Set of representative channels

SINGLE CHANNEL MODELING

The single channel model is close to the one developed by Mizutani *et al.* [5].

Based on symmetry considerations, one defines the set shown in Figure 2 as the representative element of the filter. It consists in a quarter of the inlet channel, the porous layer (substrate and soot) and a quarter of the outlet channel.

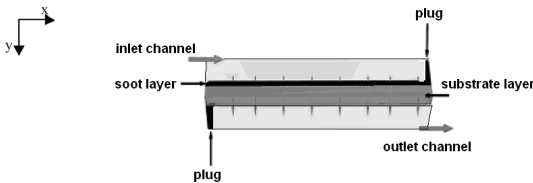


Figure 2: Zone taken as the representative element of the filter

This zone is spatially subdivided in both x and y directions, so that the element is numerically described as a set of 0D models being either in the inlet channel (i), in the porous layer (PL: soot + substrate) or in the outlet channel (o) (Figure 3).

i	i	i	i	i	i	i	i	i	
P	PI	PI	PI	PI	PI	PI	PI	PI	
o	o	o	o	o	o	o	o	o	

Figure 3: Elementary Model

FLOW FIELD

Flow field is computed using the assumption taken by Mizutani *et al.*: the conservation of momentum is not calculated, and the flow is solved considering that pressure drop along the filter is negligible as compared to the pressure drop due to filtration through porous media.

Local filtering mass flow is then given by:

$$Qf(x) = \frac{Qm}{R(x) \cdot \sum \frac{1}{R(x)}} \quad (1)$$

Where $R(x)$ represents the local resistance of the porous media to the flow. $R(x)$ is derived from the classical Darcy's law applied to the substrate and trapezoidal soot layers, for a compressible gas:

$$R(x) = \frac{\mu(x)}{k_w} \cdot w_{wall} \cdot \frac{RT(x)}{A_f(x)} + \frac{\mu(x)}{k_s} \cdot \frac{RT(x)}{A_f(x)} \cdot w_{soot} \cdot \ln\left(\frac{d}{d - 2 \cdot w_{soot}}\right) \quad (2)$$

Flow field in the inlet and outlet channels are then computed using the conservation of mass.

$$\begin{aligned} Q_{inlet}(x + dx) &= Q_{inlet}(x) - Q_{filtration}(x) \\ Q_{outlet}(x + dx) &= Q_{outlet}(x) + Q_{filtration}(x) \end{aligned} \quad (3)$$

Besides this quasi-steady state assumption taken for flow field calculation, one uses a transient computation concerning energy balance, soot mass balance and O_2 mass transfer.

ENERGY BALANCE

For time calculation concerns, wall and gas temperatures are the same in the porous layer. This assumption has also been taken by other authors and was explained by the very high gas to surface contact in the substrate [5, 6, 8].

Equations governing energy balance are as follows.

Inlet Channel (Gas)

$$\begin{aligned} (\rho \cdot Cp \cdot V)_I \cdot \frac{dT_{I(x)}}{dt} &= (Q \cdot Cp \cdot T)_{I(x-dx)} \\ &- (Q \cdot Cp \cdot T)_{I(x)} + (H \cdot S)_I \cdot (T_{W(x)} - T_{I(x)}) \end{aligned} \quad (4)$$

Wall

As conductive heat exchange with neighbors is only put on the wall, we exceptionally note in the following equation $T_{W(x)}^n$ the wall temperature at the computed location and $T_{W(x)}^N$ the neighbored channel wall temperature.

$$\begin{aligned} (\rho \cdot Cp \cdot V)_W \cdot \frac{dT_{W(x)}^n}{dt} &= (Q \cdot Cp \cdot T)_{I(x)}^n - (Q \cdot Cp \cdot T)_{W(x)}^n \\ &+ \sum_{X=I,O} (H \cdot S)_X \cdot (T_{X(x)}^n - T_{W(x)}^n) + \frac{\lambda_x \cdot S_x}{\Delta x} \cdot (T_{W(x-dx)}^n - 2 \cdot T_{W(x)}^n + T_{W(x+dx)}^n) \\ &+ \sum_{N=neighbors} \frac{\lambda_y \cdot S_{(n-N)}}{\Delta y_{(n-N)}} \cdot \frac{1}{N_{elements}(n)} \cdot (T_{W(x)}^N - T_{W(x)}^n) \\ &+ \frac{dm}{dt} \cdot \Delta H_{soot \rightarrow CO_2} \end{aligned} \quad (5)$$

If the segment is on the filter periphery, heat exchange with the outside occurs through convection and radiation terms:

$$\begin{aligned} \text{ConvectiveHeat loss} &= (H \cdot S)_N \cdot (T_{outside} - T_{W(x)}^n) \\ \text{RadiativeHeat loss} &= (\varepsilon \cdot \sigma \cdot S)_N \cdot (T_{outside}^4 - T_{W(x)}^n{}^4) \end{aligned} \quad (6)$$

Convective heat loss parameter H is function of the vehicle speed.

Convective and radiative parameters are put in an equivalent heat resistance.

Outlet Channel (Gas)

$$\begin{aligned} (\rho \cdot Cp \cdot V)_O \cdot \frac{dT_{O(x,t)}}{dt} &= (Q \cdot Cp \cdot T)_{O(x-dx,t)} \\ &+ (Q \cdot Cp \cdot T)_{W(x-dx,t)} - (Q \cdot Cp \cdot T)_{O(x,t)} + (H \cdot S)_O \cdot (T_{W(x,t)} - T_{O(x,t)}) \end{aligned} \quad (7)$$

SOOT COMBUSTION AND CHEMISTRY

As the chosen substrate has a low porosity and a low PM loading, a single layer model has been retained (as compared to the two layers model proposed by Konstandopoulos *et al.*[6]).

The contact between soot and catalyst coating being "loose" as defined by Neeft *et al.*[9,10], we use a not-catalyzed reaction rate, given by **equation 8**.

$$\frac{dm}{dt} = -k_0 \cdot e^{-\frac{Ea}{RT}} \cdot m_0 \cdot \left(\frac{m}{m_0}\right)^\alpha \cdot x_{O_2}^\beta \quad (8)$$

The mass dependence of this reaction rate allows both to take into account the effect of initial soot mass and the evolution of reaction rate with conversion, as shown in [11].

Accumulation of soot is not taken into account, since regeneration step is quite short and few soot mass is emitted by the engine during this period.

Even if the PM loading of the DPF is low, it is sufficient to oxidize completely CO to CO_2 . Therefore, one consider that the enthalpy of soot combustion is a function of carbon to CO_2 enthalpy.

γ accounts for the fact that soot is not made of pure carbon (for example, Diesel soot characterized in [11] showed 87% carbon content). γ is calibrated considering regeneration results, to obtain both peak temperatures and soot regeneration efficiency.

As the filter is placed behind a 2.2L Diesel Oxidation Catalyst (DOC), HC and CO conversions are not considered, concentrations of those two species being very low during regeneration steps (<100 ppm C_i each).

Though passive regeneration can cause heterogeneities of soot mass in the filter and then influence the initialization of soot mass distribution in the DPF, it is not taken into account in this model. The assumption of homogeneous loading in the filter at the beginning of the regeneration step is taken for the calculations.

O₂ MASS TRANSFER

O₂ transfer through convective and diffusive transport is taken into account.

Inlet Channel (Gas)

$$(\rho \cdot V)_I \cdot \frac{dx_{O_2, I(x)}}{dt} = (Q \cdot x_{O_2})_{I(x-dx)} - (Q \cdot x_{O_2})_{I(x)} + \frac{S_x}{\Delta x} (D_{O_2, I(x-dx)} \cdot x_{O_2, I(x-dx)} - 2 \cdot D_{O_2, I(x)} \cdot x_{O_2, I(x)} + D_{O_2, I(x+dx)} \cdot x_{O_2, I(x+dx)}) + \frac{S_y}{\Delta y} (D_{O_2, W(x)} \cdot x_{O_2, W(x)} - D_{O_2, I(x)} \cdot x_{O_2, I(x)}) \quad (10)$$

Wall

$$(\rho \cdot V)_W \cdot \frac{dx_{O_2, W(x)}}{dt} = (Q \cdot x_{O_2})_{I(x)} - (Q \cdot x_{O_2})_{W(x)} + \frac{S_y}{\Delta y} (D_{O_2, O(x)} \cdot x_{O_2, O(x)} - 2 \cdot D_{O_2, W(x)} \cdot x_{O_2, W(x)} + D_{O_2, I(x)} \cdot x_{O_2, I(x)}) + \frac{dm}{dt} \cdot \frac{32}{12} \quad (11)$$

Outlet Channel (Gas)

$$(\rho \cdot V)_O \cdot \frac{dx_{O_2, O(x)}}{dt} = (Q \cdot x_{O_2})_{O(x-dx)} - (Q \cdot x_{O_2})_{O(x)} + \frac{S_x}{\Delta x} (D_{O_2, O(x-dx)} \cdot x_{O_2, O(x-dx)} - 2 \cdot D_{O_2, O(x)} \cdot x_{O_2, O(x)} + D_{O_2, O(x+dx)} \cdot x_{O_2, O(x+dx)}) + \frac{S_y}{\Delta y} (D_{O_2, W(x)} \cdot x_{O_2, W(x)} - D_{O_2, O(x)} \cdot x_{O_2, O(x)}) \quad (12)$$

INLETS AND OUTLETS OF THE MODEL

This model needs first, exhaust mass flow and oxygen concentration upstream of the filter. Vehicle speed is needed for heat losses computation. Pressure downstream of the filter allows pressure loss calculation.

Finally, one needs temperature distribution in front of the filter. In this study we used a specific set of thermocouples to know this inlet as proposed by Miyairi et al. [12], instead of making CFD calculations [13] or assumptions [14].

Though this choice leads to heavy experimental set-up (see part 2), it has many advantages as compared to CFD solution, the more important being a better accuracy than CFD codes. Actually, those codes are very sensitive to boundary conditions on the upstream exhaust line, which are tricky to apply specially in under floor conditions. As we look for precise regeneration efficiency and peak temperatures simulations, this solution seems to be the best for the moment.

As any regeneration model, it returns soot mass and temperatures distribution in the filter, pressure loss and O₂ concentration downstream DPF.

EXPERIMENTAL SET-UP

ENGINE AND DPF CHARACTERISTICS

For this study, we used a vehicle equipped with a 4-cylinder common rail diesel engine, with the following characteristics:

Engine	Engine type	L4, 16 valves
	Displacement	1995 cc
	Bore*stroke	84*90 mm
	Comp. ratio	15.7/1
	Rated Power	127 kW
	Common rail system	piezo injectors 1600 bar max
Exhaust line	Air handling system	Variable Geometry Turbocharger, Air to air intercooler
	Oxidation Catalyst volume (close coupled)	2.2 L
	DPF Volume (underfloor)	3.7 L
Vehicle	DPF material	SiC
	Inertia	1665 Kg

Table 1: Engine and exhaust line description

DPF Regeneration is conducted in a specific mode (regeneration mode, in opposite to normal mode), with two late post injections and inlet air throttling that allows a DPF inlet temperature above 600°C.

The SiC material has the following limits:

- 1200°C as maximum temperature
- 180°C / cm as maximum radial and longitudinal gradient.