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Effect of soot accumulation in a diesel particle filter on the combustion process and gaseous emissions

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ABSTRACT

Particulate filters are massively used in new diesel vehicles to reduce particulate emissions below the regulation limits. The setting of filters in the exhaust lines imposes various changes in the engine working conditions, including increased back-pressure, resulting in some fuel penalty. In this work, the filter loading process was investigated in a modern common-rail diesel engine equipped with diesel oxidation catalyst and diesel particulate filter. The engine was fuelled with EN-590 diesel fuel and run in a highly emissive operating mode selected among those reproducing the New European Driving Cycle. During the test the intake/exhaust conditions, intake charge, injection and combustion timing, fuel consumption and regulated emissions were measured. It was observed that the EGR (exhaust gas recirculation) ratio decreased from 30% down to 21%, shifting the combustion process away from its design parameters. In consequence, nitrogen oxides emissions increased around 60% along the test, compromising the objectives of the emission regulations. A 4% increase in fuel consumption throughout the test was measured, this value being higher than that expected from previous literature. An energy balance revealed that such penalty was caused by increased pumping losses and, especially, by higher energy losses via exhaust gas temperature and engine coolant.

1. Introduction

The continuous evolution of the automobile industry is now accelerating in order to meet the increasingly stringent regulations. Today and future vehicles must i) be adapted to the use of a wide range of biofuels in response to the legislative framework promoting the use of renewable energy in transport [1,2], ii) improve fuel efficiency to fulfil by 2015 the mandatory target of 130 g CO_2/km (on average) for the new car fleet [3] and iii) integrate sophisticated after-treatment techniques for complying with the limits in NO_x emissions and particulate matter (PM) [4]. With regard to the PM issue, most light-duty diesel vehicles are recently equipped with filters (DPF) in the exhaust line. Although these filters are detrimental for the fuel economy [5,6] and thus for the CO_2 emissions, other PM-reducing techniques (increasing the injection pressure as proposed in [7,8] or developing advanced combustion processes such as the premixed charge compression ignition [9]) have already reached a technical limit which impede further reductions.

Depending on the filtration mechanism, DPF (diesel particle filter)'s may be classified as wall-flow and flow-through [6,10]. Flow-through type filters made of ceramic foam, wire mesh or metal wool [10–12] create a smaller pressure drop which in turn benefits the fuel efficiency but at the cost of a lower filtering efficiency, often below 60% [6,10]. In consequence wall-flow type filters are the most commercialized in diesel vehicles to fulfil the current PM regulations. They are honeycomb monoliths made of ceramic materials such as cordierite and silicon carbide [13,14] which consist of a series of parallel channels alternatively plugged at each end to force the exhaust gas flow through the porous filter wall. Thus, the collection efficiency of wall-flow filters depends mainly on the porosity of the substrate, the mean diameter pore and the wall thickness [10,15]. With time the soot collected in the trap will lead to unacceptable values of exhaust back-pressure, therefore a regeneration process (classified as active or passive [16,17]) must be induced.

Wall-flow filters show very high collection efficiencies [6,10] but a fuel penalty is associated to the active regeneration process, which most vehicle manufacturers achieve through late post-injections to increase the exhaust temperature above that required for soot oxidation [18]. Besides, an additional fuel penalty is related to the DPF loading process. This is frequently explained by the higher pumping losses as the exhaust back-pressure increases

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Nomenclature

BET (°C) break-even temperature
bmep (bar) brake mean effective pressure
bsfc (g/kWh) brake specific fuel consumption
DOC diesel oxidation catalyst
DPF diesel particle filter
ECU electronic control unit
EGR exhaust gas recirculation
FAME fatty acid methyl esters
HRF heat released fraction
$\dot{m}_{\mathrm{air}}$ (kg/h) air mass flow rate
$\dot{m}_{EGR}$ (kg/h) EGR mass flow rate
MI main injection
n (min^{-1}) engine speed
n number of moles
NEDC New European Driving Cycle
NO_x nitrogen oxides
p (bar, hPa) pressure
PiI pilot injection
PM particulate matter
SOC_MI (°CA) start of combustion of the main injection
SOC_PiI (°CA) start of combustion of the pilot injection
SOMI (°CA) start of the main injection
SOPiI (°CA) start of the pilot injection
T (°C) temperature
THC total hydrocarbons
V_T (L) engine displacement
W (kg/kmol) molecular weight
x_{CO_2} molar (or volumetric) fraction of CO_2
η_v volumetric efficiency

Subscripts
amb ambient
exh exhaust
int intake

[6,19], and a quick calculation formula is reported [5] based on this assumption. However, such explanation may be oversimplified since the higher back-pressure modifies the pressure and thermal conditions, not only in the exhaust system but also in the intake system since both are communicated via the EGR duct, inducing changes in the cylinder conditions, the combustion process and the engine-out emissions. Also the electronic control unit (ECU) of the engine may introduce further changes in response to the higher temperature and back-pressure. As a result, power output and fuel consumption may differ from the expected ones [20].

Some reports study experimentally the effects of the addition of a DPF, finding a slight increase (generally below 1.5%) in fuel consumption [21–23] and consequently a reduced thermal efficiency and air-to-fuel ratio [21] with respect to the original layout (without DPF). This trend was observed either with naturally aspirated engines [21] and supercharged ones [22,23], and either from engine steady tests [21,23] or from vehicle road operation [22]. Although not stated clearly in these reports, it can be presumed that this penalty corresponds to the case of a relatively clean filter, and that the fuel penalty could get worse as the filter load increases. This is confirmed by other authors [24] who used a model to simulate the effect of a clean DPF and a DPF with a soot load of 5 g/m^2 (this soot mass corresponds to a medium-loaded filter) on a turbocharged engine. They found that such a low soot load could increase up to 2.5% the fuel penalty for some EGR and air-to-fuel ratio conditions. No result was reported about the effect on the combustion timing and the engine-out emissions.

In this work, the transition occurred in a modern diesel engine during a complete loading cycle in the DPF is examined. The experimental approach consisted in fixing the engine power while monitoring (not controlling) the other variables, particularly the EGR valve and the injection timing. The changes in the engine intake/exhaust conditions are shown first, followed by the injection and combustion process. All these are finally linked to the engine-out emissions and performance.

2. Experimental

2.1. Test engine and instrumentation

A scheme of the experimental installation is presented in Fig. 1. The engine tested is a 4-cylinder, 4-stroke, turbocharged and intercooled 2.0 L Nissan diesel engine (model M1D) with 110 kW of rated power. This engine utilizes a common-rail injection system with piezoelectric injectors and cooled EGR for NO_x reduction. Its after-treatment devices comprise a diesel oxidation catalyst (DOC) and a catalysed wall-flow diesel particle filter (DPF). The engine was also equipped with the necessary instrumentation for the measurement and monitoring of the main operating temperatures and pressures (intake air, fuel, lube oil, and at different locations along the exhaust system). The main characteristics of the engine are shown in Table 1, and those of the DPF are listed in Table 2.

The engine was coupled to an asynchronous electric brake Schenck Dynas III LI 250. The brake control system permitted to measure and control the engine speed (with 1 rpm accuracy), throttle position and effective torque (with 0.1% accuracy). The INCA PC software and the ETAS ES 591.1 hardware were used for the communication and management of the electronic control unit (ECU) of the engine. This allowed real-time recording of the injection pressure and timing, among other variables.

Fuel consumption was measured with a gravimetric fuel balance AVL 733-S (with uncertainty 0.12%), and fresh-air flow rate with a hot-wire sensor Siemens 5WK9628. Carbon monoxide and carbon dioxide concentrations were determined with a non-dispersive infrared analyser MIR2M (with noise below 0.5%). CO_2 measurements were used to calculate the EGR ratio through Equation (1). The EGR flow rate was obtained by substituting the EGR ratio and the air flow rate in Equation (2). It can be demonstrated that both equations are equivalent when the molecular weight of the intake air and the exhaust gas are assumed equal (see Appendix).

$$\mathrm{EGR} = \frac{x_{CO_2,\mathrm{int}} - x_{CO_2,\mathrm{amb}}}{x_{CO_2,\mathrm{exh}} - x_{CO_2,\mathrm{amb}}} \cdot 100\% \quad (1)$$

$$\mathrm{EGR} = \frac{\dot{m}_{\mathrm{EGR}}}{\dot{m}_{\mathrm{EGR}} + \dot{m}_{\mathrm{air}}} \cdot 100\% \quad (2)$$

The total hydrocarbon emissions (THC) were sampled through a heated line, pump and filter (190 °C), and were measured with a flame ionization detector Graphite 52M-D (with repeatability 1%). Finally, the NO_x emissions were measured using a chemiluminescence Topaze 3000 analyzer (also with repeatability 1%), able to distinguish between NO and NO_2. All the emission analysers are integrated in a modular system supplied by Environnement, along with the electro-valves and software necessary to commute between sample gas, zero gas and calibration gases. As indicated in Fig. 1, the exhaust gas was sampled from up- and down-stream of