

Energy Efficiency Analysis of Monolith and Pellet Emission Control Systems in Unidirectional and Reverse-Flow Designs

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ABSTRACT

The work aims at analysing the energetic performances of monolith and pellet emission control systems using unidirectional and reverse-flow design (passive and active flow control respectively). To this purpose a one-dimensional transient model has been developed and the cooling process of different system configurations has been studied. The influence of the engine operating conditions on the system performances has been analysed and the fuel saving capability of the several arrangements has been investigated. The analysis showed that the system with active reverse flow and pellet packed bed design presents higher heat retention capability. Moreover, the numerical model put in evidence the large influence of the exhaust gas temperature on the energy efficiency of the emission control systems and the significant effect of unburned hydrocarbons concentration.

INTRODUCTION

In the last few decades a host of experimental and numerical investigations have been carried out to improve the effectiveness of emission control systems and reduce engine tailpipe emissions [1-3]. In fact, to meet the increasingly stringent environmental regulations, the automotive industry and research community have designed and developed more and more efficient aftertreatment systems [4-6].

For stoichiometric spark ignition engines, three-way catalytic converters (TWCs) represent a proper and efficient method to control the pollutant emissions [7-9]. Conversely, for the lean burn engine, energy efficient technologies to simultaneously remove NO_x , particulate matters, CO and HC are not available, and an integration of different systems is often necessary [1,10,11].

Furthermore, the emission control systems often require supplemental fuel to assure the proper thermal level for reliable processes and, as a consequence, a non-negligible fuel penalty and a deleterious impact on engine efficiency are produced [10,11]. A profound understanding of the thermal behaviour of aftertreatment apparatus is thus requested for developing and optimizing modern internal combustion engines (ICEs). In particular, in the last few years, an innovative flow control has been proposed to reduce the fuel penalty (active control) [11,12]. It is based on reversed flow converters with cyclic inversion of the exhaust gas between the two system ends (Figure 1). A cycle consists of forward and backward operations. The cycle is defined as symmetric reverse flow if the two consecutive processes last the same time, otherwise the cycle is termed as asymmetric reverse flow. Conversely, passive flow control represents the technical solution largely adopted in practice with unidirectional flow within the aftertreatment system.

The present work aims to analyse the energetic performances of active and passive emission control systems and to compare monolith and pelletized configurations. The first system consists of structured monolith beds with several square cross-section thin channels. The second design employs beds of randomly packed spherical pellets (unstructured system).

A computational one-dimensional transient model has been developed to assess the heat exchange between the solid and the exhaust gas, to estimate the energy effectiveness of the emission control systems and to evaluate the fuel saving capability of the active system with respect to the unidirectional operation.

Furthermore, the response of the system after sudden variations in engine load and the effect of the engine operating conditions on the energetic performances of emission control systems have been investigated. Specifically, the influence of the exhaust gas temperature and unburned hydrocarbons concentration have been analysed.

Although the behaviour of the aftertreatment systems depends also on different practical aspects, such as pulsating flow, engine vibration, road shock, proper location of the converter, the evaluation of their influence goes beyond the scope of this paper.

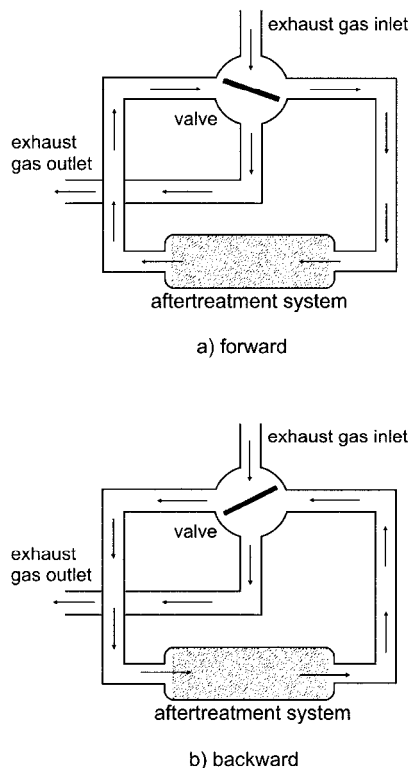


Figure 1 Active emission control system: forward (a) and backward (b) operations.

NUMERICAL MODEL

A one-dimensional transient model was developed to simulate the thermal exchange between the solid and exhaust gas phase. The basic model assumptions are the following:

- working fluid as ideal mixture;
- constant mass flow rate;
- negligible thermal accumulation of the gas in the system;
- negligible conductive and radiative energy exchange mechanisms;
- adiabatic system towards the surroundings.

In accordance with the literature [15], the composition of the exhaust gas was calculated from the combustion of $C_{12}H_{24}$ with excess air. The oxidation of the unburned hydrocarbons HC present in the exhaust gas was treated by assuming an equivalent mass flow rate of methane, that is representative of the combined exhaust HC and CO in average Diesel and lean burn gaseous fuel engines, in accordance to literature [11].

Two emission control configurations were studied: monolith structured and pellet unstructured design.

MONOLITH STRUCTURED DESIGN

A typical cross section of the monolith bed is shown in Figure 2, while the dotted square represents the control volume adopted in the model. The structured monolith usually consists of blocks having longitudinal square holes with about 1 mm x 1 mm channel flow area [1].

Referring to the control volume, the thermal exchange can be described as follows:

$$\frac{\partial T}{\partial x} = \frac{h \cdot (P / A_c) \cdot (T_s - T)}{G \cdot c_p} \quad (1)$$

$$\rho_s \cdot c \cdot S \cdot dx \cdot \frac{\partial T_s}{\partial t} = \dot{q}_{ex} \cdot P \cdot dx - G \cdot A_c \cdot c_p \cdot dT \quad (2)$$

where h is the heat transfer coefficient;
 x is the longitudinal coordinate;
 P is the channel perimeter;
 A_c is the cross area of a single channel;
 T is the gas temperature;
 T_s is the solid temperature;
 G is the mass flow rate per unit area A_c ;
 c_p is the specific heat of the working fluid;
 c is the specific heat of the solid phase;
 ρ_s is the solid density;
 t is the generic time;
 $\dot{q}_{ex}$ is the exothermic energy generation rate per unit area, related to the oxidation of the equivalent methane.

The ratio (P/A_c) is the exchange surface per unit volume of the bed and $S \cdot dx$ is the volume of solid that exchanges thermal energy with the gas:

$$S = w \cdot (2b + w) \quad (3)$$

where b and w are shown in Figure 2.

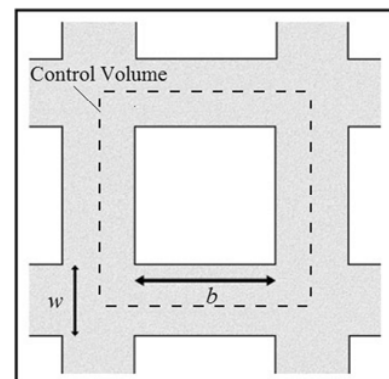


Figure 2 Typical cross section of a monolith converter.

The exothermic reaction of the working fluid was treated as surface heat generation and no assumptions of light off temperature for the catalyst were assumed. A purely thermal calculation was adopted in the model and the energetic contribution of oxidation was equally distributed over the whole channel:

$$\dot{q}_{ex} = \frac{\dot{m}_{CH_4Eq} \cdot LHV}{P \cdot L} \quad (4)$$

where LHV is the lower heating value;
 L is the monolith length;
 $\dot{m}_{CH_4Eq}$ is the equivalent methane mass flow rate.

For the complete resolution of the problem, the further following equations were considered:

$$\frac{\partial(\rho v)}{\partial x} = 0 \quad (5)$$

$$\frac{\partial p}{\partial x} = -\frac{1}{2 \cdot D_{eq}} \cdot f \cdot \rho \cdot v^2 \quad (6)$$

$$p = \rho \cdot R \cdot T \quad (7)$$

where v is the gas velocity;
 ρ is the gas density;
 p is the gas pressure;
 D_{eq} is the channel hydraulic diameter.

The momentum balance equation for a fixed bed is taken into account through the equation 6. It depends on the Fanning coefficient f , equal to $64/Re$ for laminar flow ($Re < 2000$), as occurs in the operative conditions of this paper.

The equations were solved with a finite difference scheme. More detail on the model and on the relative validation, is reported in the literature [16,17]. As far as the heat transfer coefficient is concerned, various correlations are present in the literature. Specifically, there are correlations for laminar or turbulent flow, for constant heat flux or constant solid temperature [18,19]. The heat transfer coefficient h was obtained as a function of Reynolds (Re) and Prandtl (Pr) numbers, using the correlation attributed to Hausen [20] and adopted by Rafidi and Blasiak [21] to study the thermal exchange within a structured monolith:

$$Nu = h \cdot \frac{D_{eq}}{k} = 3.61 + \frac{0.0668 \cdot (D_{eq}/L) Re Pr}{1 + 0.04 \cdot [(D_{eq}/L) Re Pr]^{2/3}} \quad (8)$$

The value of Nusselt number for the operating condition used in the present work is near to 3.65, while 3.66 is the Nu constant value largely adopted in literature for square cross section and constant heat flux with a fully developed laminar flow [18,19].

PELLET UNSTRUCTURED DESIGN

The control volume for pelletized emission control system is represented in Figure 3. The converter consists of a bed of spherical pellets with typically 3 mm diameter [1].

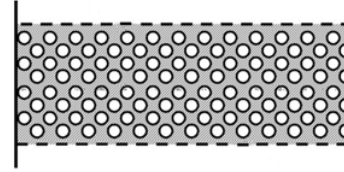


Figure 3 Typical cross section of pellet emission control system.

Referring to the control volume, the thermal exchange in the packed bed with forced fluid flow was described through the following equations:

$$\frac{\partial T}{\partial x} = \frac{h \cdot a \cdot (T_s - T)}{G \cdot c_p} \quad (9)$$

$$\rho_s c_s A dx (1 - \varepsilon) \cdot \frac{\partial T_s}{\partial t} = \dot{q}_{ex} a A dx - G A c_p dT \quad (10)$$

where a is the wetted surface of the packed bed per unit volume;
 ε is the porosity, assumed to be uniform within the bed;
 A is the cross section of the pellet bed system.

The energetic contribution of oxidation was evaluated as following:

$$\dot{q}_{ex} = \frac{\dot{m}_{CH_4Eq} \cdot LHV}{a \cdot A \cdot L} \quad (11)$$

while the wetted surface of the packed bed per unit volume was expressed by:

$$a = \frac{6(1 - \varepsilon)}{\varphi_s d_s} \quad (12)$$

where d_s is the sphere diameter and φ_s is the sphericity of particles, which is the ratio of the sphere surface to the real surface of the particle. Therefore, sphericity is equal to 1 only for spherical particles, while it is lower for packing units of different shape.

Pressure losses were evaluated by the Ergun equation [18], which relate pressure to superficial velocity in a porous medium:

$$\frac{\partial p}{\partial x} = -\frac{150 \cdot (1 - \varepsilon)^2 \cdot \mu \cdot v}{\varphi_s^2 \cdot d_s^2 \cdot \varepsilon^3} - \frac{1.75 \cdot (1 - \varepsilon)}{\varphi_s \cdot d_s \cdot \varepsilon^3} \cdot \rho \cdot v^2 \quad (13)$$