

Filtration Efficiency and Pressure Drop Performance of Ceramic Partial Wall Flow Diesel Particulate Filters

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ABSTRACT

A simple 1-dimensional filter model, with symmetric and asymmetric channels, has been developed to investigate the fundamental behavior and performance of ceramic partial diesel particulate filters (PFs). The governing equations of mass and momentum are similar to those of a full DPF [7, 15]. A standard DPF with the plugs at its inlet face removed has been referred to as a 'rear-plugged PF' while, one with the plugs at the outlet face removed has been referred to as a 'front-plugged PF' in the present study. Removal of some of the plugs from a standard ceramic DPF reduces the (i) overall pressure drop (ΔP) across the filter, (ii) filtration efficiency (FE) of the DPF, and (iii) manufacturing cost. Partial filters stand a high chance of being deployed in diesel exhaust after-treatment systems for the emerging markets (Brazil, Russia, India, China) that follow Euro 4 emission regulations. The model has been used for predicting the pressure drop and filtration efficiency of the PF for a variety of situations and practical filter sizes. The effects of filter length, soot load, channel asymmetry, mass flow rate, wall permeability and multi-staging of filters on the pressure drop and filtration efficiency performance of the PF have been studied. An iterative approach to simulating a real-world non-uniform soot layer, instead of the uniform layer commonly assumed in DPF models, has been proposed. A standard DPF, front-plugged PF, and a rear-plugged PF have been modeled and the differences in their fundamental behavior are discussed. The model performance has been demonstrated to be reasonably similar to that of an existing 3-dimensional simulation. The filtration efficiency of and pressure drop across PFs have been predicted at various soot loads for filter sizes typical of light-duty diesel applications. A method to model DPFs with any fraction of the inlet/outlet faces plugged has been proposed.

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1. INTRODUCTION

The emission regulations in the North American and European automotive sectors are mature by now. These markets follow the EPA 2010 and Euro 6 emission regulations, respectively at present. The pace however, is gradually catching up in the emerging markets that follow the Euro 4 and Euro 5 emission regulations that are less stringent than those followed in the developed nations. Diesel particulate filters are the prime path to mitigate particulate matter (PM) emissions from diesel exhaust of both, light-duty and heavy-duty applications. The PM emissions primarily

consist of carbonaceous soot particles, ash, volatile hydrocarbons (HCs), and sulfur compounds that are also described as the soluble organic fraction (SOF). The SOFs are contributed by the un-reacted fuel and lube oil vapors.

A regular wall flow Diesel Particulate Filter (DPF) is a substrate having parallel channels separated by internal porous walls. The inlet and outlet ends of the substrate are selectively plugged in a checkerboard pattern, such that the gas entering the inlet channel is forced through the porous wall due to the backside of the channel being plugged and exits the outlet channel, with the wall retaining part of the

particulate matter from the exhaust stream. It has high filtration efficiency (FE) of 95-99% on a PM mass basis [1, 2]. A DPF is sized considering its maximum soot holding capacity and the soot regeneration intervals that in turn, depend on the automotive controls and desired fuel economy. The Euro 5b regulations for light-duty diesel vehicles (LDDV) not only impose a restriction on the PM mass (0.005 g/km for LDDVs) but also, on the number of PM particles emanated at the exhaust tailpipe. A limit of 6×10^{11} per km on the particle number (PN) for the Euro 5b legislation has been enforced [3]. In situations where the exhaust comprises significant amounts of carbonaceous soot, the volatile sulfates and HCs tend to condense on the existing carbon particle surfaces. In cases where the carbonaceous soot emission count is low, the HCs and sulfates condense independently forming large numbers of nucleation-mode particles [4] thereby, adding to the PN count in the exhaust. A diesel oxidation catalyst (DOC) alone is capable of reducing the PM mass to a significant extent. However, it does not reduce the PN count as it does not impact the soot oxidation appreciably. For example, a heavy-duty retrofit DOC reduced the PM by 20-30% by oxidizing the soluble long-chain HCs [5] but, not the PN. A regular DPF, on the other hand, is designed to meet the requirements of both, the PM and PN reduction. It is capable of achieving >90% reduction efficiency on both fronts [5] and such a filter is aptly used in emission control applications designed to meet the Euro 5b and Euro 6 emission regulations. However, the Euro 4 particulate matter (PM) emission legislation (0.025 g/km for LDDVs) is much less stringent when compared to its 5b counterpart and, dictates no regulation on the PN count. The rigors of a standard DPF, capable of reducing the PM content and PN by more than 90%, are not required to meet the loose Euro 4 PM regulation. A DOC, combined with a partial filter (PF) downstream with the partial filter having some of the plugs removed, demonstrates an approximately 50-60% PM mass reduction performance and is sufficient to cater to this demand for PM emission control. A partial filter is a unique device whose PM filtration efficiency, greater than 50%, falls between that of a flow-through and a regular DPF [5]. A passive regeneration-based after-treatment recipe comprising a DOC with a PF downstream is therefore, becoming increasingly popular in the Euro 4 markets of Brazil, Russia, India, China and Australia. The DOC ensures oxidation of the CO, HCs, SOFs, and some of the NO to NO₂ while, the PF oxidizes the carbonaceous soot appreciably (>50%) with the aid of NO₂.

Removing plugs from either the inlet or the outlet faces of a regular ceramic DPF qualifies the substrate to be called a partial DPF. PFs help lower the axial pressure drop across the filter length and hence, the overall fuel consumption due to a reduced number of plugged channels when compared to a regular DPF. In the present work, a partial filter with plugs removed from the inlet face has been referred to as a 'rear-plugged PF', and one with plugs removed from the outlet

face has been coined 'front-plugged PF'. The 'rear-plugged' and 'front-plugged' PFs are also popularly known as 'outlet-plugged' and 'inlet-plugged' PFs respectively. The first generation of PFs were metallic, PM-METALIT®, that had corrugated stainless steel foil with blades designed to bypass the exhaust stream to sintered fleece layers where the PM was trapped and oxidized through passive regeneration [6]. A 3D flow model coupled with a phenomenological filtration submodel for this PF was developed and used to study the effects of inlet velocity, single and multi-stage configuration, and soot particle size on filtration efficiency. It was suggested that optimization of the blade design would improve the FE [1]. The PM-METALIT technology with a DOC upstream was implemented in a heavy duty retrofit application (see Fig. 1) as a passive PM control device where, the PF relied on the C:NO₂ reaction for continuous passive soot oxidation and filter regeneration. A PM mass reduction efficiency of 77% was achieved [5].



Fig.1. DOC+PF architecture for engines with EGR designed for the Euro 4 emission legislation.

Ceramic wall-flow PFs are a more robust alternative to the PM-METALIT® filters. These next-generation PFs are easy to manufacture, significantly less expensive than their metallic counterpart, and are gradually invoking interest in the scientific community. Besides, the ceramic PFs are closer in design and performance characteristics to the more widely used ceramic fully-plugged DPFs. Practical diesel emission control applications, comprising ceramic PFs, are just entering the after-treatment markets and yet to mature as a technology. However, due to their attractiveness for use in retrofit and Euro 4 markets, it is important to understand their performance as a function of engine operating conditions, filter geometry, plugging pattern and PM loading levels. There has been no simple, lucidly illustrated PF analysis reported in the literature that systematically studies the impact of each of these parameters. Zhang et al. [1] reported a 3D simulation of a rear-plugged PF using a commercial CFD solver. However, the governing equations used and boundary conditions applied at the PF outlet need to be clearly stated [1] and systematic dependence of system performance on filter characteristic and operating conditions was not discussed. On the other hand, a well-defined set of 1D numerical models as well as 3D simulations for a wall-flow standard DPF have emerged over the past three decades. The axial pressure variation across the wall-flow DPF and the active or passive regeneration of soot have been modeled for a pair of adjacent filter channels using 1-dimensional CFD methodologies [2, 7,8,9,10,11,12,13,14,15,16]. Bissett [9] modeled the thermal regeneration of soot and the axial variation of pressure and temperature across a DPF in case of

low exhaust flow rates. The soot deposit layer and the substrate wall were considered a single unit in the model referred to as the 'wall system'. The change in channel geometry due to the soot layer deposit was neglected. The transverse pressure drop across the wall system was computed using a standard equation for porous flow and considering the particle layer and the substrate wall as two porous media in series. Koltsakis et. al. [8] modeled a transient fuel-additive-assisted DPF regeneration process where, the catalytic activity submodel was coupled to the transport phenomena occurring in the filter. Haralampous et. al. [7] developed a 1D DPF model investigating the dependence of soot permeability on the local temperature and pressure. They also accounted for the change in gas density with axial pressure. The advantage of using asymmetric channels for greater ash capacity was also evaluated [17]. Wurzenberger et. al. [10] presented an approach for modeling DPFs having an asymmetric channel geometry, soot and ash deposits. Analytic expressions for predicting the overall pressure drop across a DPF have also been developed [18,19,20]. Bissett et al. [21] have estimated the impact of wall flow Reynolds number on the friction coefficient and heat transfer characteristics of flow in square channels. Konstandopoulous [20] reviewed the most recent advances in DPF simulation discussing topics such as the soot deposit microstructure and compaction, deep-bed filtration, kinetics of passive soot oxidation, ash and soot entrainment phenomena. However, the analytical solutions are zero-dimensional and do not yield the axial variation of the channel velocity or pressure profiles. Information on the channel velocity and pressure profiles primarily helps distinguish between the fundamental behaviors of a regular DPF, a front and rear-plugged PF. It has been demonstrated that predictions from these 1-D models are able to accurately capture the experimental observations over a broad range of operating conditions. None of these models address the filter performance with some of the plugs removed and the filter's sensitivity to the plugging pattern.

Based on this background, in the present work, we have extended the 1D isothermal model for regular wall-flow DPF to predict the filtration and pressure drop behavior of ceramic PFs. The specific objectives of the work are to:

1. Develop a finite-difference based numerical 1D pressure-drop model for PFs and the corresponding boundary conditions. The same conservation equations of mass and momentum have been used as in case of wall-flow DPFs.
2. Study the differences in fundamental performance behavior between a front and rear-plugged ceramic PF.
3. Investigate the effect on ΔP and FE across a PF of the following parameters: (a) cell density or cpsi (cells per square inch), (b) asymmetric geometry and channel widths/ ratios, (c) exhaust mass flow, (d) wall permeability, (e) multi-staging of PFs, and (f) non-linear soot layer thickness.
4. Develop a generic method to model DPFs with any fraction of the faces plugged

Predict representative numbers for the ΔP and FE performance of a PF, across a range of soot loads, for a filter size typical of light-duty diesel applications.

2. THEORY

The PF is modeled using a 1D grid with the cell centers being the computational node points (see Fig.2). The computational domain comprised a representative volume element (RVE) of two adjacent filter channels (see Figures 3,4,5) assuming that all remaining parallel channels of the filter are identical [1, 3]. The key model assumptions are:

- a. The exhaust flow is laminar
- b. The filter is composed of identical adjacent filter channel pairs. The computational domain is a RVE
- c. The transverse pressure drop across the soot layer and substrate wall follows the Darcy law [7, 8]
- d. The formation and deposition of ash have not been considered. The effect of plug length has also been neglected
- e. Secondary flow effects have been neglected

2.1. Governing Equations and Principles of Operation

The resistances offered to the exhaust flow are due to: (i) friction along the channel length, (ii) contraction losses at the filter inlet, (iii) expansion losses at filter outlet, and (iv) transverse pressure losses due to the substrate wall thickness and soot layer. The mass balance equations for the inlet and outlet channels are (see Fig. 2):

$$\frac{d(\rho_1 u_1)}{dz} = -\frac{4\rho_w u_w}{a-2\delta_{sc}} \quad (1)$$

$$\frac{d(\rho_2 u_2)}{dz} = +\frac{4\rho_w u_w}{a} \Delta z \quad (2)$$

The momentum balance equations are:

$$\frac{d(\rho_1 u_1^2)}{dz} = -\frac{dP_1}{dz} - \frac{f\mu u_1}{a^2} \quad (3)$$

$$\frac{d(\rho_2 u_2^2)}{dz} = -\frac{dP_2}{dz} - \frac{f\mu u_2}{a^2} \quad (4)$$

The soot cake and depth layer thicknesses are defined by:

$$\delta_{sc} = \frac{1}{2} \left(a - \sqrt{a^2 - \frac{m_{sc}}{\rho_{sc} V_{DPF} \times 1550 \text{ cpsi}}} \right)$$

$$\delta_{sd} = \frac{m_{sd}}{\rho_{sd} \times 1550 \text{ cpsi}} \frac{1}{4a} \quad (6)$$

The Darcy equation for the wall velocity is [2, 10]:

$$u_w = \frac{P_{in} - P_{out}}{\mu \left(\frac{t_{wall}}{k_{wall}} + \frac{t_{SD}}{k_{SD}} + \frac{a}{2k_{SC}} \ln \left(\frac{a}{a - 2\delta_{sc}} \right) \right)} \quad (7)$$

The PF inlet contraction loss and the outlet expansion loss in pressure are as follows [7]:

$$\Delta P_{entrance} = \left(1.1 - 0.4 \frac{(a - 2\delta_{sc})^2}{2(a + \delta_{sc})^2} \right)^2 \frac{\dot{m}^2}{2\rho_1 N_{inlet}^2 (a - 2\delta_{sc})^2} \quad (8)$$

$$\Delta P_{exit} = \left(1 - \frac{a^2}{2(a + \delta_{sc})^2} \right)^2 \frac{\dot{m}^2}{2\rho_2 N_{inlet}^2 a^2} \quad (9)$$

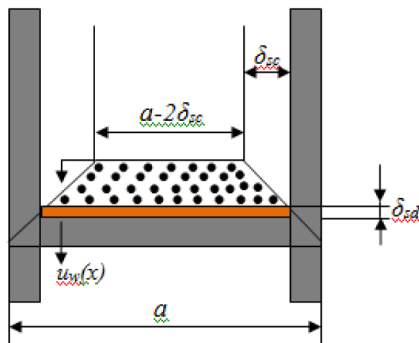


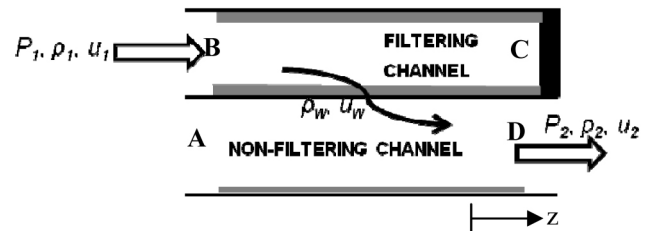
Fig.2. Cross-sectional area of a computational cell

In essence, the governing equations of mass and momentum are the same as those used for modeling wall-plugged regular DPFs. The boundary conditions (BCs) however, are different. The BCs for a rear-plugged PF, front-plugged PF, and a regular DPF are listed below in Eqs. (10), (11), (12), (13), (14), (15). The subscript '1' denotes the inlet or the filtering channel while, the subscript '2' denotes the outlet or the non-filtering channel.

2.1.1. Rear-plugged PF

$$u_1 = 0 \text{ @ } z = L \quad (10)$$

$$P_1 = P_2 \text{ @ } z = 0; P_2 = P_{atm} \text{ @ } z = L \quad (11)$$



1: Inlet/Filtering channel, 2: Outlet/Non-filtering channel

Fig.3. Rear-plugged PF

In Eqs. (10)-(11), u_1 , P_1 and u_2 , P_2 are the axial velocities and static pressures in the inlet and outlet channels (see Fig. 3). The transverse wall velocity, which is a measure of the filtration efficiency, is denoted by u_w . The functioning principle of a rear-plugged PF is illustrated in Ref. [1]. The distribution of exhaust in the inlet and outlet channels depends on the pressure resistance along the flow path. The static pressures are the same at the inlet of the PF (points A and B). The pressure decreases along the length of the outlet channel to hit ambient pressure at its exit, point D. The only losses in total pressure along the inlet channel (B-C) occur due to channel friction and advection. On the other hand, the pressure drop across the path B-D occurs due to advection, channel friction, soot layer and wall pressure drop. Hence, the pressure drop across the inlet (filtering) channel (B-C) is considerably less than that observed across the outlet (non-filtering) channel (A-D). Note that the static pressures at points A and B being the same; the pressure drop across the path B-D is identical to the pressure drop across path A-D. At any axial location therefore, the static pressure in the inlet channel is higher than that in the outlet channel. This pressure difference drives exhaust flow from the inlet to the outlet channel thereby, filtering the PM. The amount of PM filtered also depends on the magnitudes of the resistances offered by the channel friction, substrate wall and the soot layer which in turn, are guided by the channel hydraulic diameter, friction factor, wall and soot permeability, and the soot packing density. The net pressure drop across the partial filter ($P_B - P_D$) increases with an increase in the exhaust flow.

2.1.2. Front-plugged PF

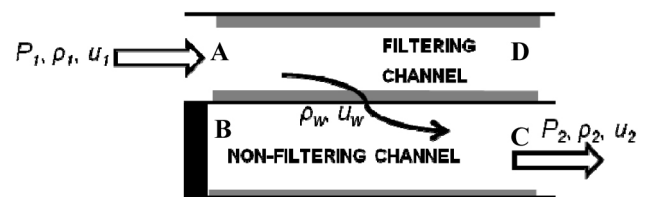


Fig.4. Front-plugged PF

$$u_2 = 0 \text{ @ } z = 0 \quad (12)$$

$$P_1 = P_2 = P_{atm} @ z = L \quad (13)$$

In case of a front-plugged PF (see Fig.4), the static pressures are the same at the outlet of the PF (points C and D). The pressure decreases along the length of the inlet channel (A-D) to hit ambient pressure at its exit, point D. The only loss in pressure along the outlet channel (B-C) occurs due to channel friction. On the other hand, the pressure drop across the path A-C occurs due to channel friction, soot layer and wall pressure drop. Hence, the pressure drop across the outlet channel (B-C) is considerably less than that observed across the outlet channel (A-D). At any axial location therefore, the static pressure in the inlet channel (A-D) is higher than that in the outlet channel. This pressure difference drives exhaust flow from the inlet to the outlet channel thereby, filtering the PM. Note that had none of the exhaust mass flow bled from the inlet to the outlet channel, the static pressure across the entire outlet channel (B-C) would have been at ambient ($P_B = P_C$). The higher static pressure in the inlet channel drives a portion of the exhaust flow, through the soot layer and substrate wall, into the outlet channel thereby, increasing the static pressure in the outlet channel from ambient.

2.1.3. Regular DPF

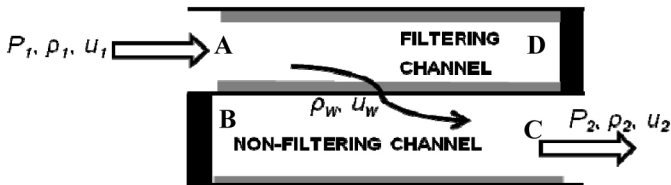


Fig.5. Wall-plugged regular DPF

$$u_2 = 0 @ z = 0; u_1 = 0 @ z = L \quad (14)$$

$$P_2 = P_{atm} @ z = L \quad (15)$$

In case of a regular wall-plugged DPF (see Fig.5), the static pressure is ambient at the filter outlet (point C). The pressure decreases along the length of the outlet channel (B-C) to hit ambient pressure at its exit. The loss in pressure along the outlet channel occurs due to channel friction only. The pressure drop across the path A-C occurs due to channel friction, soot layer and wall pressure drop. The presence of plugs at the ends of both the inlet and outlet channels prevents a sharp fall in static pressure across the lengths of either channel. At any axial location, the static pressure in the inlet channel (A-D) is higher than that in the outlet channel. This pressure difference drives exhaust flow from the inlet to the outlet channel thereby, filtering the PM.

2.1.4. Partially-plugged DPFs

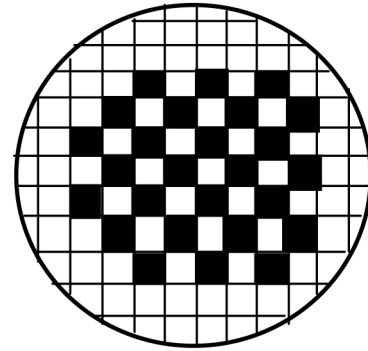


Fig.6a. DPF with a portion of the face (y%) plugged

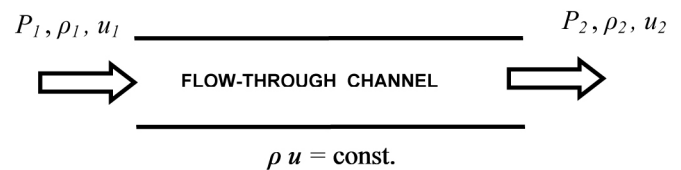


Fig.6b. Modeling flow-through channel like a straight duct

The present model can be adapted to any filter design and plugging pattern. The rear and front-plugged PDPFs are examples of filters that are 50% plugged. A regular DPF is a filter that is fully (100%) plugged. In order to model an unconventional plugging fraction of y% (see Fig. 6a), the number of plugged channel pairs and flow-through channels were computed. The plugged channel pairs were modeled as described in section 2.1.3 and as shown in Fig.5- same governing equations and boundary conditions. The remaining flow-through channels were modeled as square ducts (see Fig. 6b) using Eqs. (16), (17), (18), (19). The boundary conditions for the flow-through channels are as follows:

$$\frac{\partial(\rho u)}{\partial z} = 0 \quad (16)$$

$$u_w = 0 @ z = 0 \text{ to } L \quad (17)$$

$$(P_2)_{\text{flow-through}} = (P_2)_{\text{plugged pair}} = P_{atm} @ z = L \quad (18)$$

$$(P_1)_{\text{flow-through}} = (P_1)_{\text{plugged pair}} @ z = 0 \quad (19)$$

The combined unit of the plugged channel pair and a flow-through channel can be visualized as a parallel electrical circuit with the pressure being analogous to the voltage and the mass flow being analogous to the electrical current. The absolute pressure at the inlet and outlet of the filter is same for both the plugged channel pair as well as the flow-through

channel but, the exhaust masses entering the flow-through channel and plugged channel pair are different depending on their resistance.

2.2. Effect of CPSI and Asymmetric Channels

Two methods of increasing the filtration efficiency of a PF are: (i) by increasing its cpsi and (ii) incorporating asymmetry in the channel geometry. In case of filters that have asymmetric cells, the adjacent cells have different hydraulic diameters (see Fig.7). The asymmetric cell technology (ACT[®]) also helps increase soot and ash accumulation capacity when the larger channels are plugged at the DPF rearoutlet. A typical channel width ratio of 1.3 is used in Corning ACT substrates [17]. The asymmetric cell hydraulic diameters, a_1 and a_2 , could be computed using Eqs. (20), (21), (22) where, a_{nom} is the cell hydraulic diameter had the DPF been symmetric.

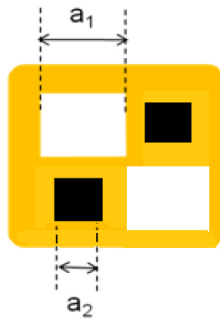


Fig.7. Inlet face of an asymmetric DPF. Channel width ratio = $\frac{a_1}{a_2}$

$$(a_{nom} + t_{wall})^2 \times cpsi \times 1550 = 1 \quad (20)$$

$$2(a_{small} + t_{wall})^2 + 2(a_{big} + t_{wall})^2 = 4(a_{nom} + t_{wall})^2 \quad (21)$$

$$\frac{a_1}{a_2} = 1.3 \quad (22)$$

2.3. Effect of Multi-Stage PFs

Another method of increasing the filtration efficiency of a PF is by using two filters in series (multi-stage) instead of using a single one. The pressure drop as well as the filtration efficiency obtained across the multi-filter arrangement is higher than the single filter because of the additional set of plugs at the back of the first PF. Most of the soot gets trapped in the leading filter because of the plugs at its back. The boundary conditions for this arrangement (see Fig.8) are discussed in Eqs. (23),(24),(25),(26). The pressure losses in the gap between the two filters have not been considered.

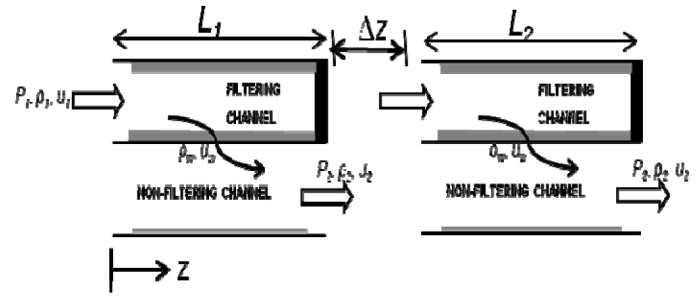


Fig.8. Multi-stage rear-plugged PFs

$$u_1=0 \text{ @ } z=L_1, L_1+L_2 \quad (23)$$

$$P_1=P_2 \text{ @ } z=0, L_1+\Delta z \quad (24)$$

$$P_2(z = L_1) = P_1(z = L_1+\Delta z) \quad (25)$$

$$P_2=P_{atm} \text{ @ } z=L_1+L_2 \quad (26)$$

Note that multi-staging of PFs increases the overall pressure drop across the filter system and may lead to problems in thermal management of soot and sulfur. The above-mentioned boundary conditions are intuitive and have been developed for illustration purposes ignoring the effects of soot oxidation and radial pressure variation.

3. RESULTS AND DISCUSSION

3.1. Model Validation

3.1.1. Comparison of predictions from present model to fully-plugged DPF model

In order to validate the accuracy of our methods presented here, the predictions from the model for the case of a regular fully-plugged DPF have been compared with the results for such a case reported in the literature (see Eqs. 14-15). A silicon carbide DPF having a diameter of 144 mm, length 152 mm, 200 cpsi, wall thickness 0.38 mm and wall permeability of $1.2 \times 10^{-12} \text{ m}^2$ was modeled at an exhaust mass flow of 0.1 kg/s and temperature 200°C as described in Ref. [2]. The channel and wall velocities obtained from the present model (solid lines) and those reported in Ref. [2] (dashed lines) are compared in Figures 9a,b,c. A close match with the Haralampous model results [2] demonstrates that predictions of the present model are reasonably accurate in trend and magnitude.