

# Performance Analysis of a Turbocharged Heavy Duty Diesel Engine with a Pre-turbo Diesel Particulate Filter Configuration

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## ABSTRACT

The use of particulate filters (DPF) has become in recent years the state of the art technology for the reduction of soot aerosol emissions for light, medium and heavy duty Diesel vehicles. However, the effect of the system location on engine performance is a key aspect that should be studied. In the present work a numerical study has been carried out with the objective to analyze the effect on the engine performance of an innovative DPF placement upstream of the turbine. This study has been performed by means of the gas dynamic simulation of a two-stage turbocharged heavy duty Diesel engine, which has been previously modeled from experimental data obtained under steady state conditions. The original DPF has been divided into two monoliths for the case of the pre-turbo DPF configuration. Three cylinders discharge in each of these monoliths and after the filtration the flow is driven towards the high-pressure turbine and the EGR system.

The engine response is compared with a pre-turbo DPF configuration under steady state conditions against the common DPF placement downstream of the turbine. The study covers a parametric study based on EGR and wastegate valves opening. It has allowed comparing the variations on the engine operation as function of the EGR rate and the air-to-fuel ratio settings. Different soot loadings have been considered in order to extend the study over a wider range of operating conditions. Finally, the location of the DOC has been also considered as an additional parameter (upstream of

the DPF or downstream of the turbines in the pre-turbo configuration).

Results allow identifying the phenomena controlling the engine response when the DPF is placed upstream of the turbines. Among them highlight the increase of the thermal level that favors the passive regeneration, the DPF pressure drop reduction and the benefits that the use of two parallel monoliths brings out regarding the attenuation of the wave interference between cylinders during exhaust valve opening, which leads to make unnecessary the use of twin turbines. The analysis of the interaction among all the involved phenomena provides a comprehensive understanding of the proposed architecture and its potential regarding both the engine and the DPF behavior.

## INTRODUCTION

One of the most difficult problems that engineers and manufacturers has to face up during the Diesel engine development is the reduction of pollutant emissions to fulfill the stringent levels set by current and oncoming regulations [1]. This fact has derived in an extensive implantation of aftertreatment systems in the modern Diesel engines. Among these systems, the Diesel particulate filter (DPF) is the most extended aftertreatment device for trapping soot particles contained in Diesel engine exhaust gas [2]. A wide variety of DPFs involving different materials, filter media and geometric configurations has been developed due to an intensive research during the last decades [3].

In this context, the fact that the placement of the DPF in the exhaust line involves a flow restriction has been a key issue. As the DPF is loaded, this restriction is higher and leads to the increase of the engine back-pressure. To prevent unacceptable levels of exhaust pressure on the engine, it is required frequent oxidation of the accumulated soot (regeneration) [4]. The main future research in this area will focus on the optimization of this process in order to limit the use of active regeneration as much as possible.

Traditionally, the DPF is located after the Diesel oxidation catalyst (DOC) and both downstream of the turbine. The DOC increases the amount of  $\text{NO}_2$ , which favors the oxidation of the particulates in the presence of catalysts inside the DPF. This is the base concept for the CRT system (Continuously Regeneration Trap) development [5][6].

However, several configurations regarding the location of the DPF upstream of the turbine have been proposed in recent years. Some of these configurations offer the possibility of dividing the original DPF into several monoliths [7]. Each of these monoliths is communicated with the different sets of combustion cylinders, improving the aftertreatment system packaging. Other novel multifunctional configurations of the exhaust line has even proposed including the monoliths in the outlet of each exhaust port [8][9], what gives rise to lower exhaust manifold volume.

The pre-turbo DPF placement offers also the possibility to develop a clean high-pressure exhaust gas recirculation system. The exhaust gases can be recirculated after crossing the DPF monoliths being free of particles and avoiding soot depositions through the EGR path [10], so that the useful life of the EGR system may be extended. The high-pressure EGR may even return to the intake manifold through a double wall of the exhaust manifold. It would act as a thermal insulator of the aftertreatment devices and of the exhaust manifold, as it has been proposed in [11] for insulating the catalyst of petrol engines.

In this way, Payri et al. [12] propose a novel configuration of the exhaust manifold including the pre-turbo multi-monolith DPF placement, clean high-pressure EGR and thermal insulation. The EGR free of particles flows through a gap conformed by a double sheet of the manifold. The objective is to obtain an effective thermal insulation of the exhaust gases upstream of the DPF but at the same time to promote the pre-cooling of the clean high-pressure EGR, because of the heat transfer towards the environment through an extensive surface.

Under this context, the purpose of the present work is to quantify the effect of a pre-turbo DPF architecture of the exhaust line on the engine performance of a two-stage heavy duty Diesel engine. The analyzed architecture is the one

proposed by Payri et al. [12], but in this paper the thermal insulation is only based on an air gap, without any EGR flowing through it. The results allow discussing the advantages and drawbacks of placing the aftertreatment system upstream of the turbines.

The understanding of the involved phenomena is achieved by means of the use of modeling techniques based on computer thermo-and fluid-dynamic calculation. It is important to note that most of the wall-flow DPF models assume quasi-steady flow [13] as a consequence of the common placement of the DPF downstream of the turbine. Nevertheless, the use of a wall-flow DPF model formulated for non-homentropic, one-dimensional, unsteady, and compressible flow is necessary to carry out a rigorous study regarding the DPF location. This wall-flow DPF model is integrated into OpenWAM™ [14] [15], an open source gas dynamic software for internal combustion engine modeling developed at CMT-Motores Térmicos. This software is applied to simulate the engine operation under steady state conditions. These operating conditions are given by a parametric study which is defined by the EGR and waste-gate valves opening. It allows analyzing the effect of the proposed architecture in a wide range of EGR rate and air-to-fuel ratio (AFR) settings.

One of the most noticeable conclusions resulting of this work is the important increase of the thermal level at the inlet of the DPF, which would improve the conditions for passive regeneration. This fact would involve a positive impact on the fuel consumption, reducing the active regeneration events and the soot loading levels (minimization of the pumping losses). In this way, the results show that the engine performance when the aftertreatment system is placed upstream of the turbine is influenced by the effect of the pressure drop and the heat losses across the DPF, as well as the destruction of kinetic energy associated to the pulses across the DPF. Finally, the division of the DPF into several monoliths allows attenuating the wave interference between cylinders and reduces the pumping losses [16] providing a similar effect than a twin turbine.

## ENGINE MODEL

The numerical study has been carried out under full-load steady state conditions with OpenWAM™ [14][15], an open source gas dynamic software for internal combustion engine modeling developed at authors' institution. The engine selected for this work has been a double-stage turbocharged heavy-duty Diesel engine. It is an in-line six-cylinder engine, which is equipped with a waste-gate HP single entry turbine and a LP fixed geometry turbine. The EGR circuit is divided into two parallel circuits that are controlled by two valves and two reed valves, which avoid the circulation from the intake to the exhaust manifold. The specifications of the engine are summarized in [Table 1](#).

Previously to the analysis of the DPF placement influence, the engine model was tuned up under steady state operating conditions at several engine speeds. Experimental data obtained in an engine test bench were taken as reference. The engine was tested without DPF installed in the exhaust line, as shown in [Figure 1](#), which represents the 1D scheme of the original engine in OpenWAM's interface.

[Table 1](#) shows the comparison between experimental data and modeled results at full-load steady state conditions at 1200 rpm (maximum torque), 1500 rpm and 1800 rpm (maximum power). The parametric study is focused at 1500 rpm, which is the most often engine speed at driving conditions. The maximum difference between the experiments and the model is limited to  $\pm 3\%$  for all magnitudes and operating points. Additionally, the good agreement between instantaneous magnitudes, which are shown in [Figure 2](#) for steady state conditions at full-load 1500 rpm, evidences that the model provides good accuracy and consistency in the prediction of the different aspects describing the engine operation.

**Table 1. Engine specifications.**

|                           |                                      |
|---------------------------|--------------------------------------|
| Type of engine            | Double-stage turbocharged HDD engine |
| Engine swept volume       | 10836 cm <sup>3</sup>                |
| Number of cylinders       | 6                                    |
| Stroke                    | 152 mm                               |
| Bore                      | 123 mm                               |
| Connecting Rod            | 255 mm                               |
| Compression ratio         | 16:1                                 |
| Maximum torque (at speed) | 2000 Nm (1200 rpm)                   |
| Turbocharger              | WG HP turbine + LP turbine           |

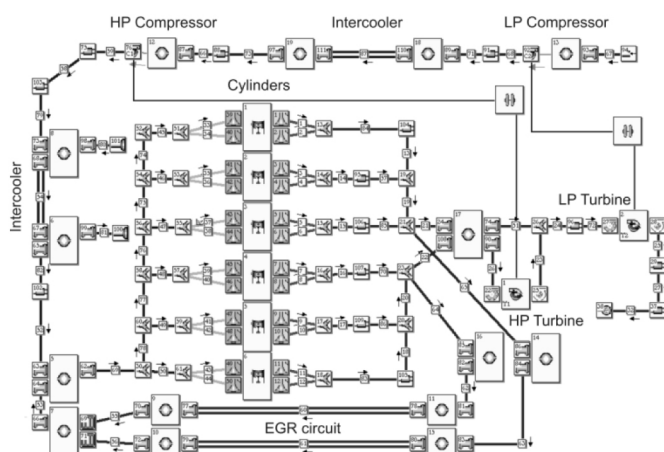


**Figure 1. Engine scheme in OpenWAM's interface.**

The one-dimensional model for wall-flow DPF used in this study is able to deal with unsteady flow, which is characteristic of the proposed placement upstream of the turbine. In this way, the model solves the governing

equations for non-homentropic, one-dimensional, unsteady, and compressible flow inside the monolith channels. As any other element within a gas dynamic model, the wall-flow DPF is geometrically represented as a proper combination of 1D elements, which are solved in OpenWAM™ by means of finite difference schemes, 0D elements, which are solved by means of filling-and emptying models, and boundary conditions, which are solved by the method of characteristics. The DPF model formulation is described in detail in [\[17\]](#).

In this work, the DPF behavior was assumed to be axisymmetric, so that the thermo-and fluid-dynamic of the flow inside a channel is only a function of its radial location. For that reason, the monolith is discretized into concentric channel beams. It is assumed that all the pairs of inlet and outlet channels in the same channel beam have the same behavior. Therefore, only one of them has to be solved by channel beam. The solution of the governing equations in the internal nodes of the channels has been performed with the two-step Lax&Wendroff method [\[18\]](#). Its formulation was adapted to couple the solution of the governing equations in the inlet and outlet channels by means of the Darcy's law [\[17\]](#). The boundary conditions of the monolith channels were solved applying the method of characteristics derived for flow in porous wall channels [\[19\]](#).

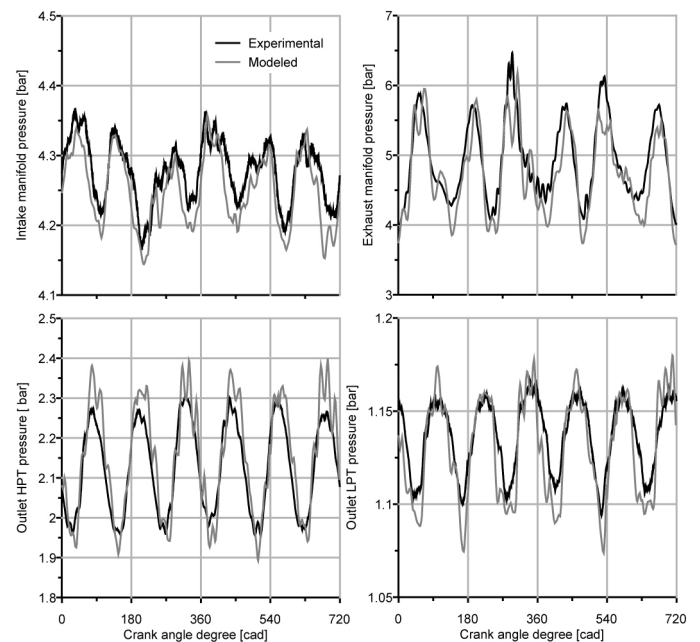


**Figure 2. Comparison of instantaneous variables between experimental data and modeled results under steady state conditions at full-load and 1500 rpm.**

The solution of the flow inside the monolith channels provides the pressure drop due to non-inertial mechanisms, i.e. flow through the porous medium and flow-wall friction. The remaining pressure drop contributions across the DPF, which are inertial, are modeled by boundary conditions

**Table 1. Comparison between experimental data and modeled results at full-load steady state conditions at 1200 rpm, 1500 rpm and 1800 rpm.**

|                         | 1200 RPM |         |           | 1500 RPM |         |           | 1800 RPM |         |           |
|-------------------------|----------|---------|-----------|----------|---------|-----------|----------|---------|-----------|
|                         | Measured | Modeled | Error [%] | Measured | Modeled | Error [%] | Measured | Modeled | Error [%] |
| Torque [Nm]             | 2000     | 2022.13 | 1.11      | 1950     | 1992    | 2.15      | 1650     | 1698.49 | 2.94      |
| Air mass flow [kg/s]    | 0.276    | 0.283   | 2.54      | 0.363    | 0.365   | 0.72      | 0.395    | 0.388   | -1.63     |
| EGR rate [%]            | 24.61    | 23.92   | -2.78     | 29.152   | 28.460  | -2.35     | 33.59    | 33.70   | 0.32      |
| Intake pressure [bar]   | 3.55     | 3.50    | -1.22     | 4.278    | 4.25    | -0.55     | 4.39     | 4.44    | 1.02      |
| Exhaust pressure [bar]  | 3.84     | 3.88    | 1.09      | 5.92     | 4.94    | 0.32      | 5.35     | 5.34    | -0.12     |
| Intake temperature [K]  | 322.18   | 324.51  | 0.73      | 359.30   | 354.99  | -1.20     | 379.58   | 385.53  | 1.57      |
| Exhaust temperature [K] | 809.36   | 782.73  | -3.29     | 837.92   | 809.7   | -3.37     | 822.76   | 813.89  | -1.08     |
| HP turbospeed [rpm]     | 101320   | 102839  | 1.50      | 107727   | 107613  | -0.11     | 108933   | 109306  | 0.34      |
| LP turbospeed [rpm]     | 63262    | 61397   | -2.95     | 74547    | 76041   | 2.00      | 76976    | 74680   | -2.98     |

connecting OD and 1D elements. By means of these boundaries, the pressure drop at the inlet and outlet of every DPF volume is calculated defining discharge coefficients based on the Benson's proposal [20].

This wall-flow DPF model was applied to the characterization of the DPF used in the study, which corresponds to a Corning EX-80 filter. Figure 3 shows the good agreement between modeled results and experimental data at 0.284 kg/s and 533 K at different soot loading conditions. The experimental characterization and the DPF geometrical specifications are published in the literature [21].

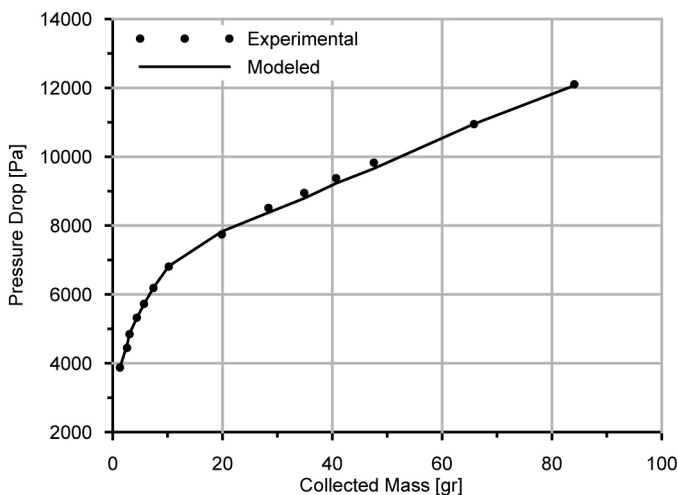


**Figure 3. Comparison of DPF pressure drop at different soot loading conditions between modeled results and experimental data from [21].**

Additionally, the DPF model includes a nodal heat transfer sub-model to consider heat losses across the monolith [22] [23]. The model solves the heat transfer equation both in the axial and radial direction. The discretization of the monolith in several channel beams allows taking into account the heat transfer between two contiguous channel beams, as well as the heat transfer from the monolith to the metal can and to the environment. Due to the fact that the study has been performed at steady state operating conditions, the heat capacity of the different materials have been neglected to speeds up the convergence of the heat transfer model [24].

For the sake of simplicity, chemical reactions in the aftertreatment systems have not been considered in this study. Therefore, the effect of the corresponding heat release on the turbine inlet temperature has been neglected. It is expected that this assumption in the modeling slightly under predicts the DPF outlet temperature and therefore the energy recovered in the turbine, being always in the security side when discussing the obtained results.

## METHODOLOGY

The first stage of the work has comprised the optimization of the exhaust line for the pre-turbo DPF configuration. Figure 4 shows the engine schemes for every of the exhaust line architectures modeled. Figure 4(a) shows the engine scheme for the case of the traditional post-turbo DOC and DPF placement whereas plots (b) and (c) present alternatives for the case of the pre-turbo DPF configuration. In these two cases, the original DPF has been divided into two monoliths and integrated as part of the exhaust manifold. Three cylinders discharge into each of these monoliths and the flow is conducted to the HPT after crossing the DPFs. This configuration allows attenuating the wave interference

between cylinders outlet and reduces the pumping losses. This solution avoids using a twin turbine, which has lower efficiency than conventional turbines [25], because of the filtration effect on the flow pulses.

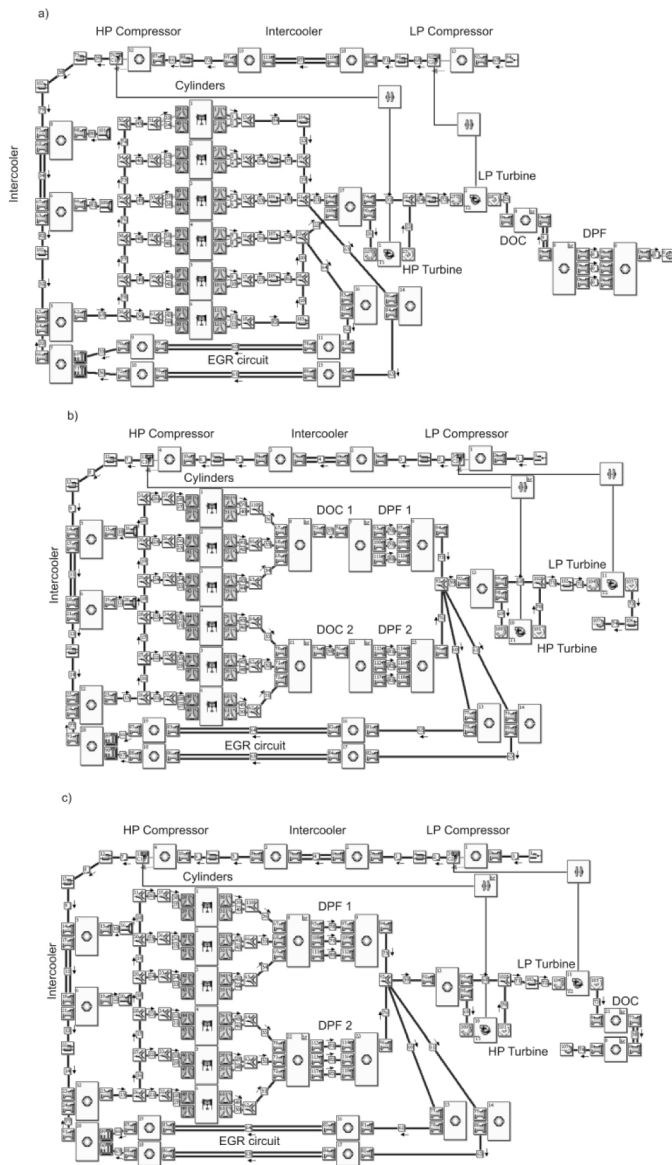


**Figure 4. Engine schemes in OpenWAM's interface: a) Post-turbo DOC and DPF configuration; b) Pre-turbo DOC and DPF configuration; c) Pre-turbo DPF configuration.**

Figure 4(b) shows the scheme of the engine for its modeling with a pre-turbo DOC and DPF placement. In this approach the DOC is placed upstream of the DPF monolith, so that the regeneration by  $\text{NO}_2$  would be promoted. Furthermore, its presence upstream of the turbine would improve its light-off during warm-up operation. On the contrary, this placement increases the mass, and then the thermal inertia, of the exhaust manifold, so that it may damage the engine

performance during the warm-up. Another drawback is related to the additional increase of the volume between the exhaust valves and the HPT. For that reason, the placement of the DOC downstream of the LPT has been also considered, as it is represented in Figure 4(c). This proposal has as main drawback that increases the back-pressure of the exhaust line, mainly if it is taken into account that the pressure drop is higher downstream than upstream of the turbines for the same operating conditions [23][26].

The influence of the exhaust manifold volume on turbine isentropic power is quantified in Figure 5(a). The x-coordinate shows the ratio between the exhaust manifold volume (before the HPT) and the engine swept volume whereas the y-coordinate represents the available power at the inlet of the HPT if the expansion were isentropic up to the ambient pressure. All the results shown in this figure have been obtained for the same operating conditions, i.e. the same engine speed (1500 rpm), the same mass fuel (full-load equivalent), the same EGR rate (21%) and the same AFR (22). As known from classic books [25], the increase of the exhaust manifold volume damages the power available for the expansion in the turbine. This behavior is due to the destruction of the kinetic energy associated to the exhaust pressure pulses. The results represented in Figure 5(a) allow extending this statement to the analysis of the pre-turbo DOC and DPF placement.

Point A in Figure 5(a) represents the original post-turbo DOC and DPF configuration, which is described in Figure 4(a). If the DOC and the DPF were placed upstream of the turbine without any geometry optimization, the isentropic power at the inlet of the HPT would be that represented by point B. As explained, this result states that the increase of the volume damages the turbine available energy with respect to point A, and then the engine performance.

However, it is important to note that the pre-turbo DOC and DPF placement provides higher power at the inlet of the HPT for the same volume than the traditional post-turbo DOC and DPF architecture. The reason is due to the influence of the aftertreatment pressure drop location. This phenomenon was already explained in [23] and will be recalled later when discussing the results of the paper.

The results of the study that will be further presented in this work have been obtained after the optimization of the exhaust manifold volume (this task will be not presented here by brevity reasons) combined with the division of the exhaust manifold into two branches. The new configuration, which is shown in Figure 5(b), keeps constant both the catalyst and filtration areas of the DOC and the DPF respectively. The result of this geometry is given by point C in Figure 5(a). It is also interesting to underline that the valve overlap applied to this new geometry is the original one, in spite of the fact that the interferences among the exhaust processes of the

cylinders have been removed. An optimization of the valves overlap may lead even to better results than these shown in following sections.

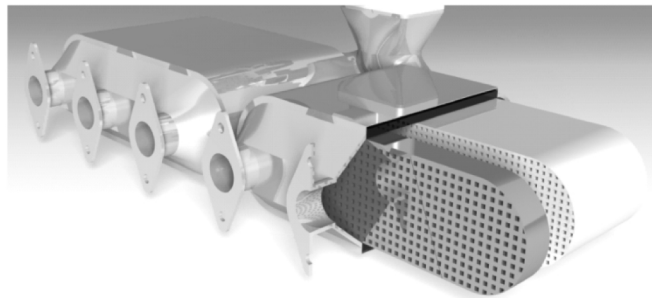
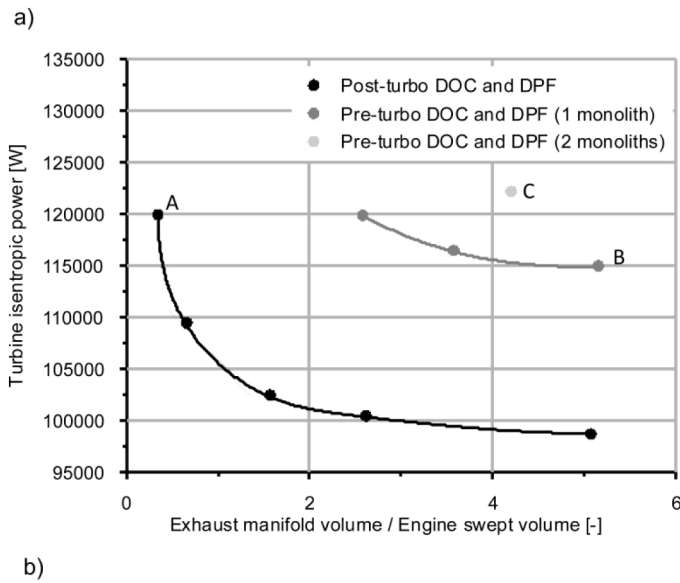


**Figure 5. a) Available isentropic power at the inlet of the HPT as function of the exhaust manifold volume and architecture at full-load and 1500 rpm. b) Pre-design of a prototype of exhaust manifold with pre-turbo DOC and DPF placement.**

The pre-turbo DPF configuration, which is represented in [Figure 4\(b\)](#) and [\(c\)](#), enables the development of a clean high-pressure exhaust gas recirculation system placing the probe of the EGR line after the DPF monoliths. This system offers some important technological advantages:

1. No soot depositions at the EGR components (valve, line or EGR cooler), extending the useful life of the EGR system and avoiding the effect of the soot on the efficiency and the pressure drop across the EGR cooler.
2. Overcoming of any  $p_3$ - $p_2$  drawback of using long route EGR from a point downstream of the DPF up to a point downstream of the compressor with the possibility of obtaining very high EGR rates.
3. Possibility of exploring the viability of the clean EGR discharge upstream of the intake intercooler, avoiding the use

of any specific EGR cooler and improving the mixing between fresh air and EGR before the intake manifold.

The last step in the optimization of the pre-turbo DPF configuration regards the additional heat loss to the environment across the DPF due to the higher surface and residence time of the gas between the exhaust manifold and the turbine inlet. This phenomenon involves less available energy at the inlet of the turbines than in the case of the post-turbo DPF placement for the same engine settings. As a consequence, the WG valve must be at a closer position (EGR valve opening changes its value to keep the EGR rate) in order to recover energy for the turbines. This strategy increases the pumping losses. With the objective to minimize this effect, a thermal insulation is proposed for the exhaust manifold only for the case when the aftertreatment systems are placed upstream of the turbines ([Figure 4\(b\)](#) and [\(c\)](#)). The specifications of the thermal insulation are summarized in [Table 3](#), where the number of the different layers grows from the internal to the external ones.

**Table 3. Properties at 293 K of the materials composing the exhaust manifold (from internal to external layers).**

| Layer | Material                           | t [mm] | $\rho$ [kg/m <sup>3</sup> ] | k [W/mK] | $c_p$ [J/kgK] |
|-------|------------------------------------|--------|-----------------------------|----------|---------------|
| 1     | Wrought iron, 0.5 % C [27]         | 1      | 7849                        | 58       | 460           |
| 2     | Ceramic (MgZrO <sub>3</sub> ) [28] | 1      | 5600                        | 0.8      | 650           |
| 3     | Air [27]                           | 3      | 1.295                       | 0.024    | 1055          |
| 4     | Steel 18 % Cr, 8 % Ni [27]         | 3      | 7817                        | 16.3     | 460           |

The numerical study performed in this work covers the comparison of the engine performance between the pre-turbo and post-turbo DOC and DPF configurations through the analysis of the thermodynamic phenomena controlling the engine and DPF response as well as its interaction.

The parametric study has been carried out under steady state operating conditions. The opening of EGR and WG valves has been varied from 0 % to 100 % and the injected fuel mass has been kept constant. This strategy allows comparing the variations of the engine performance in a wide range of EGR rate and AFR and identifying the main trends on control settings for the pre-turbo DPF architecture.

[Figure 6](#) shows the results of the parametric study at full-load and 1500 rpm under clean DPF conditions. The variations of AFR and EGR rate are shown as function of the EGR and WG valves opening. The \*bsfc, which is back-drawn in grey scale, is the percentage of bsfc variation relative to that obtained in the reference operating point. Such an operating point is defined by an EGR rate and an AFR equal to 25 %