CFD Analysis of EGR Mixers

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Scope of this paper is investigating different exhaust gas recirculation (EGR) mixers for achieving short route mixed exhaust gas with the intake air for cooled EGR, for turbocharged, intercooled heavy-duty diesel engines. The exhaust gas was carried off upstream the turbine, cooled down and mixed with the intake air upstream the intake manifold. Upstream to the mixer a throttle valve was positioned to provide a positive pressure difference for the EGR circuit.
The mixed exhaust gas fluctuation, the EGR rate at the outlet of the mixer and the pressure drop on the mixer were investigated at several engine operation points. The results were evaluated considering the optimum solution of the three criteria.

A cikk témája hűtött kipufogógáz-visszavezetéssel (EGR), turbófeltöltővel és töltőlevegő-visszahűtővel felszerelt tehergépjármű dízelüzemű motor részére kialakított, különböző kivitelű kipufogógáz-keverők vizsgálata. A kipufogógáz még a turbófeltöltő turbinája előtt, hűtést követően kerül visszavezetésre, a kompresszor, a töltőlevegő visszahűtője, illetve egy fojtószelep után keveredik össze a beszívott friss levegővel. A fojtószelep szerepe a szívócsőben a pozitív nyomáskülönbség biztosítása a kipufogógáz-visszavezetés részére.
A keverő kivezetésén az elkeveredett kipufogógáz eloszlása, a kialakuló EGR-arány, illetve a keverő ki- és belépése között létrejövő nyomáskülönbség került több motormunkapontban vizsgálatra. Az eredmények három alapvető kritérium optimális együttese alapján kerülnek értékelésre.

INTRODUCTION

Over the past 15 to 20 years emissions legislations have continuously severed across the world. The primary target of this legislation for compression ignition engines has been the reduction of the emission of nitrogen-oxides (NOx) and of particulate matter (PM).

Initial engine development efforts aiming for reduced emissions focused on reducing NOx emissions by retarding the fuel injection process. At the end of this development it resulted in an increased PM emission and/or brake specific fuel consumption (BSFC). To avoid this symptom the manufacturers of heavy duty (HD) commercial vehicle diesel engines have introduced increasingly advanced features such as low swirl cylinder heads with 4 valves per cylinder and vertically positioned central injector, shallow centrally located piston bowls, speed and load dependent injection timing control with more and more high pressure fuel injection system with turbocharger and more often with intercooler, occasionally exhaust gas recirculation (EGR) and the Diesel Particulate Filter (DPF) or Selective Catalytic Reduction (SCR) [2, 3].

Through this efforts the above mentioned trade-offs have shifted to much lower NOx and PM emissions levels, while maintaining acceptable fuel consumption levels. With these technologies commercial vehicle diesel engines could meet EURO V legislation even by 2008 (See Fig.1.).

Figure 1. EU and US emission limits of HD engines

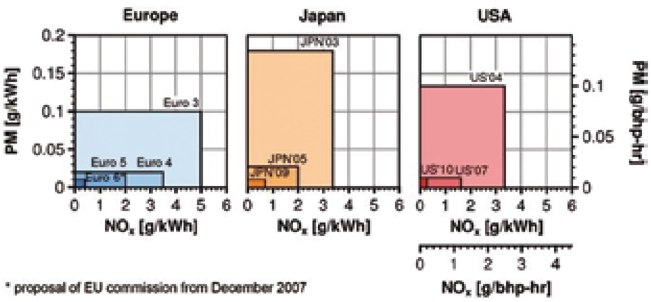

Figure 2. EU, US and Japan emission limits

Earlier it was not expected that optimization of fuel injection and air charge admission of the engine will allow such a substantial reduction of emissions (Fig.2). For meeting latest legal demands, new technologies have to be introduced in the future.

Potential key new technologies have been identified and are summarized in Figure 3. First solutions improve the emissions trade-off by influencing the physical processes of fuel injection and air admission. They no longer aim yet to ameliorate the emissions. The current solutions aim to reduce emissions by directly influencing the chemistry of the combustion process or by catalytic after treatment of the exhaust gases. From the start different solutions were identified in the current generation of HD diesel engines like exhaust gas recirculation (EGR), Diesel Particulate Filter (DPF) or Selective Catalytic Reduction (SCR). The exhaust gas recirculation (EGR) is one of the most interesting candidate technologies for wide spread introduction in the next generation of HD diesel engines.

Preceding researches on EGR-technology for HD diesel engines has shown and confirmed that very low levels of NOx and PM emissions can be achieved with appropriate hardware and a dedicated control strategy while maintaining competitive fuel economy and transient behaviour [3, 4]. The cooled EGR is one of the confirmed potential for achieving future NOx emission levels. The combination of an EGR mixer and an appropriate pressure differential control (e.g. by VGT turbocharger) can give the best results for the very low NOx levels with parallel the lowest PM increasing and fuel consumption penalty.

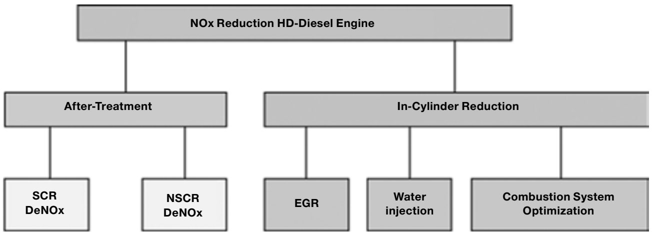

Figure 3. Different technologies for NOx reduction

The more homogeneous EGR gas and air mixture effects lower NOx levels due to evolved uniform cylinder-to-cylinder EGR rate distribution. The mixture homogeneity is an important parameter of the EGR mixer.

This paper focuses on the examination of the efficiency of different EGR mixer designs and intends to elaborate the main design criteria.

EGR CONCEPT

The EGR system of the inspected engine is shown in Figure 4. Exhaust gas is taken from the exhaust side upstream of the turbine and fed to the intake system over an EGR valve and EGR cooler.

At fixed EGR valve position the EGR rate is commensurate with the pressure difference between the exhaust and the intake system. The pressure difference drives the exhaust gas into the intake system. This pressure difference is mainly depending on the applied turbocharger maps and the engine load.

At low loads, when the turbocharger does not supply high boost pressures, the pressure in the exhaust system is usually much lower than the pressure in the intake system, thus a recirculation would not occur. To increase the EGR mass flow rates at small or negative pressure differences, pressure decrease is generated downstream to the EGR feed point in the intake system by throttle valve of the Pneumatic Booster System (PBS). Another option is by generating back pressure downstream to the turbine by the Exhaust Brake (EB).

The EGR rate is defined as the mass fraction of exhaust gas in the total intake charge as:

$$EGR_{rate} = \frac{m_{exhaust}}{m_{fresh\ air} + m_{exhaust}} \qquad (1)$$

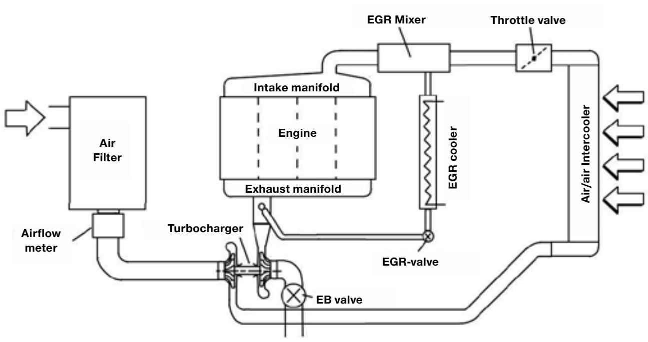

Figure 4. Cooled EGR HD engine layout

MIXER CFD MODELS

The herein presented simulation results are all based on a common CFD model for different mixer geometries, with applying different meshes and subsequent modifications in boundary condition parameters for the various cases. The model utilizes a single-phase, homogeneous, multi-component flow, consists of O2, N2 as fresh air and „Burned" components. The bulk motion of the fluid is modelled using single velocity, pressure, temperature and turbulence fields acquired from 3D Reynolds-averaged Navier-Stokes, shear stress transport turbulence equations as well as from the total energy equation. Continuity equation is also solved with separating the components having their own equations for conservation of corresponding mass. Despite of the fluid is homogeneous (in terms of momentum), a relative mass flux term in the transport equations accounts for differential motion of the individual components. This diffusion-like term involves kinematic diffusity that is defined for all unconstrained components („O2" and „Burned"). Note that, by definition, the sum of component mass fractions over all components is 1.

SIMULATION METHOD

Steady state simulations are aiming to predict mixing capability and pressure loss responses of different EGR mixer designs attached to the same intake manifold system of a 4-cylinder HD engine. The simulated domain consists of charge air and exhaust gas recirculating pipes, throttle valve, EGR mixer, elbow and intake manifold. The boundary conditions and geometry can be seen in Figure 5. Total pressure at inlets and mass flow at outlets are selected as boundary condition pair, while zero heat flux (adiabatic) and hydraulic smoothness is assumed for solid surfaces.

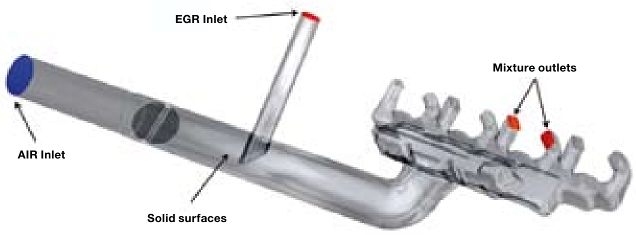

Figure 5. Simulation model with boundaries

Four cases are investigated, each corresponding to a certain steady engine operation with different RPM but almost identical partial load of the described engine.

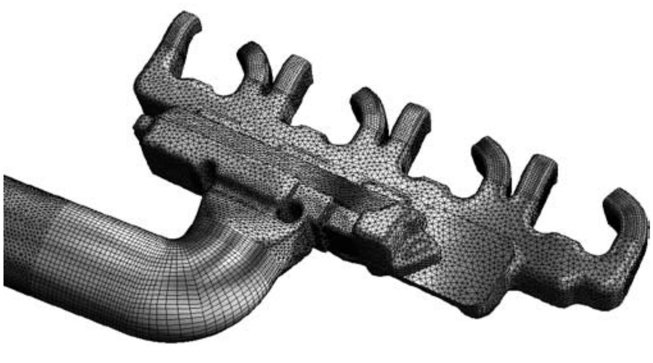

Figure 6. Hybrid-modular mesh

Inlet total pressure and outflow boundary conditions are modified consequently. The composition of inlet flows are 23% of O2 and 77% of N2 at AIR inlet and 100% „Burned" at EGR inlet for

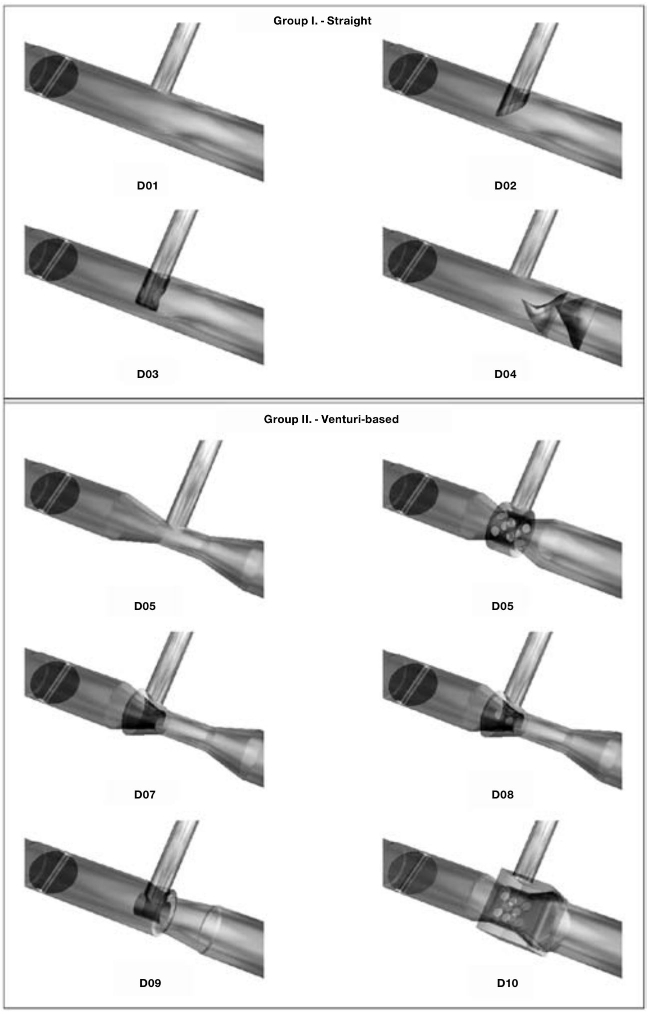

Figure 7. Investigated EGR Mixer concepts

all cases. The computational mesh is built up in modules, making the EGR mixer part interchangeable. For proper comparison the generated meshes feature almost identical number of cells and boundary layer resolution **(Figure 6)**.

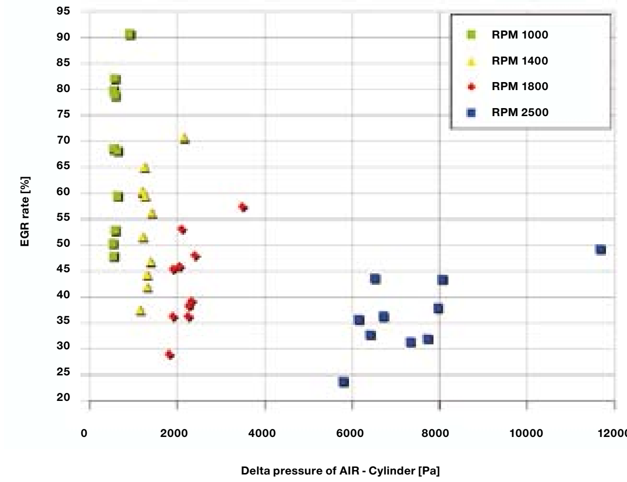

Figure 9. EGR rate as function of pressure differential of AIR – Cylinder for different cases

According to the operation sequence of the engine intake valves, the intake flow domain is highly time-dependent. The charge mass entering the manifold distributes among cylinders in time, whereas for a representative steady state simulation it was assumed that only the third cylinder intake valves are open and the entire charge flows trough them evenly divided.

MIXER DESIGNS

Ten different conceptual EGR mixer geometries are designed and fitted into the above described model shown in the Figure 7. The mixer concepts are separated into two design groups:

– Group I. contains mixers with straight piping. The exhaust gas flows into the AIR path through a simple T-shape connection in the D01. The EGR pipe penetrates into the charge air pipe with a 45 degrees chamfer at the D02. The EGR pipe penetrates into the centre of the inlet pipe with closed end and a bore parallel to the air flow at D03. Just like in case of D01, EGR gas flows

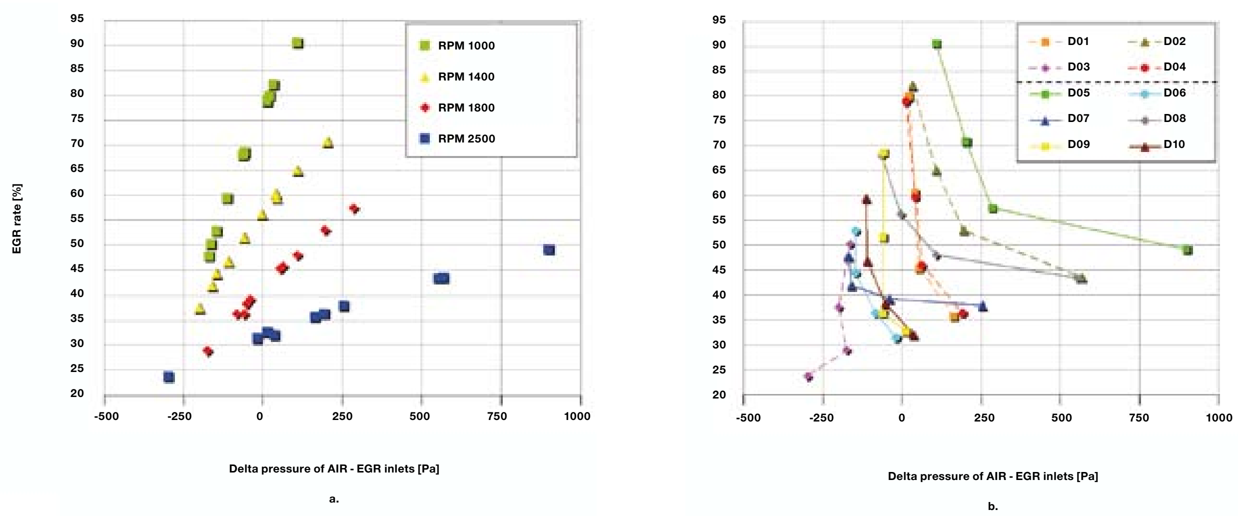

Figure 8. EGR rate as function of pressure differential of AIR – EGR inlets for various cases a) and mixer concepts b)

into the AIR path through a simple pipe connection leading to a static mixer at D04.

– Group II. involves designs utilizing the Venturi effect. D05 is a simple T-shape Venturi pipe connection. D06 features a ring volume where EGR can enter the main pipe through multiple bores. Mixers D07 and D08 are very similar. They sum up the properties of D05 and D06, but D08 has additional bores on the tapered upstream throat. A small coaxial tube in the inlet pipe aids the mixing of the EGR gas at the D09. D10 is similar to D06 concept, but the mixing area cross section is rectangular rather than circular.

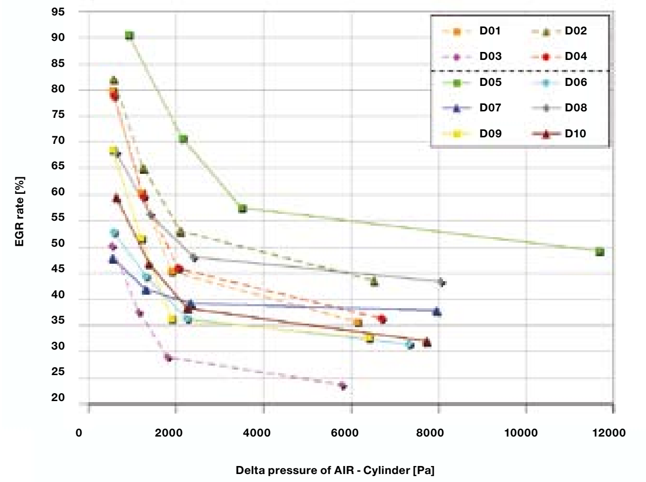

Figure 10. EGR rate as function of pressure differential of AIR – Cylinder for various designs

Figure 11. Mixing evaluation planes

RESULTS

The CFD model specific is defined as:

$$EGR_{rate} = \frac{Burned}{Burned + O_2 + N_2} = Burned \quad Mass \quad Fraction \qquad (2)$$

Although the total inlet pressure level is well defined for each case, the resulting static and dynamic pressure components vary from design-to-design. The mass-flow weighted average static pressure at the inlets, while the EGR rates at the outlet (average for one cylinder, i.e. of two Cylinder intake ports) boundaries are evaluated. It is obvious that the resulting static pressure difference between the two inlets (AIR – EGR) sets the level of EGR rate in the manifold. The higher this pressure differential, the more „Burned" component can enter the manifold. This monotonic tendency can be traced back at each engine operation point (Fig. 8/a).

The highest EGR rate values may seem excessive and pre-

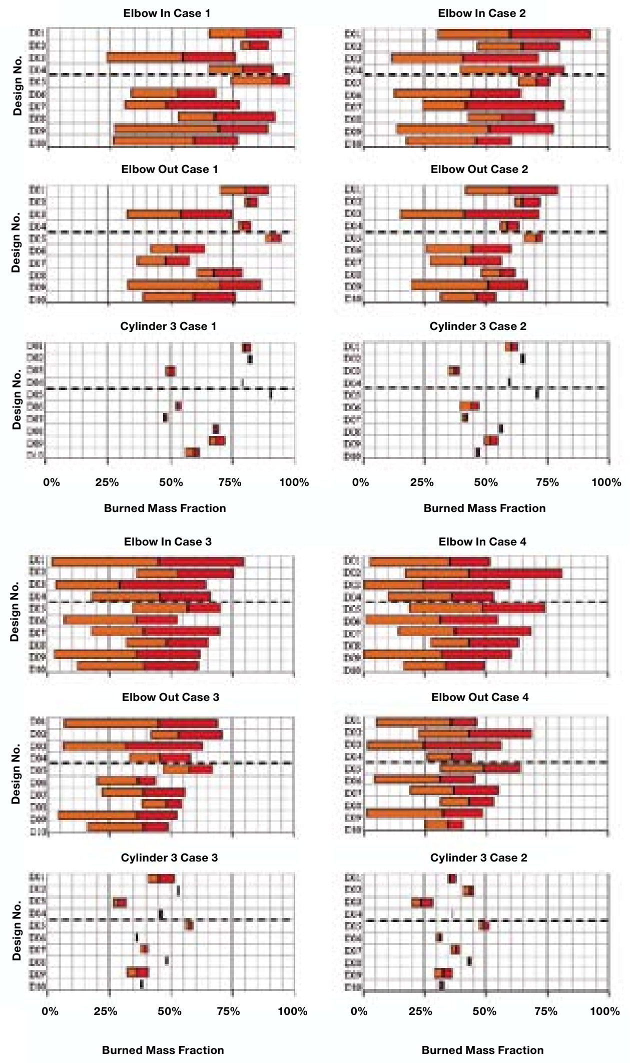

Figure 12. EGR rate fluctuation on evaluation planes

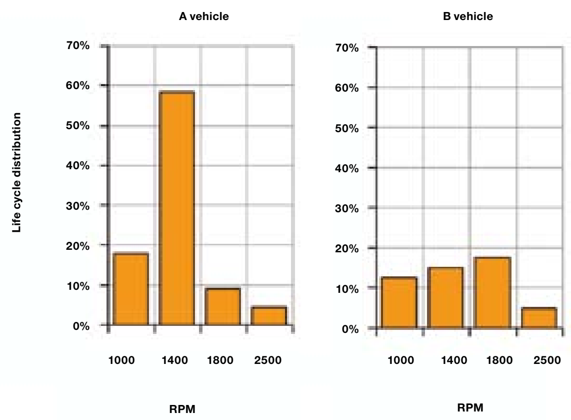

Figure 13. Distribution of investigated engine speeds. A vehicle – Suburban cycle, B vehicle – City cycle

sumably they can not be fulfilled in reality, but still theoretically possible with the given system of boundary conditions.

Besides analyzing the available EGR rates with various mixer geometries, the secondary objective is to expose the pressure loss of the designs in the AIR path, which is crucial in terms of the impact on rated engine performance. Delta pressure of cylinder (average of two ports) and AIR inlet is calculated and plotted once again with resulting EGR rate at the cylinder (Figure 9 and Figure 10).

Pressure loss varies in between 6 and 12 kPa at peak mass flow rate (2500 RPM). This value includes the pressure loss of pipes, open throttle valve, mixer and manifold as well down to the intake valves.

Investigating the „Burned" component concentration downstream the mixer can show more detail of mixing capabilities of the concepts. Three locations are assigned on the CFD model to get data from as depicted in Figure 11.

The minimal, mass-flow-weighted average and maximal values of EGR rate are seen in Fig. 12 corresponding to the four simulated engine operation cases at the three evaluation planes.

As it was observed that once the AIR – EGR inlet side static pressure difference set the average EGR rate level, it persists downstream the mixer, too. As the mixture is flowing towards the engine intake valves the EGR rate fluctuation decreases. By the flow enters the cylinder ports, the mixture homogeneity is well maintained with most of the mixer concepts (less than 5% difference between min an max). Cumulating the fluctuation values of different cases into a single measure seems to be a comprehensive judge of overall mixing effectiveness. The investigated engine RPM cases are ranked in terms of incidence by engine speed spectrums, and used as weighting numbers for cumulative representation (Ri). Two vehicle cycles are distinguished: a suburban (A) and a city (B), see Fig. 13. In addition it was assumed that the former runs at idle in 10% of its life cycle, while the latter in 50 %. Idle RPM case is not simulated, because of nearly zero EGR rates.

For such comparison, the weighted quadratic mean of EGR rate fluctuation averages is used (Figs. 14 to 15). The lower this value, the more homogeneous mixture is composed at the given section for a wide range of engine operation. Considering the negative effect of an uneven cylinder-to-cylinder distribution, the most important indicator from these is the fluctuation at the manifold inlet (Elbow Out section). If an acceptable homogeneity is maintained at this location, the possibility that the cylinders get different concentration of exhaust gas will be minimal.

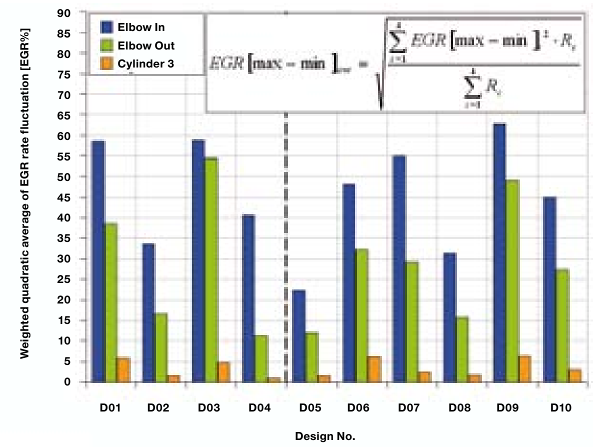

Figure 14. Cumulative EGR rate fluctuation on evaluation planes (A vehicle)

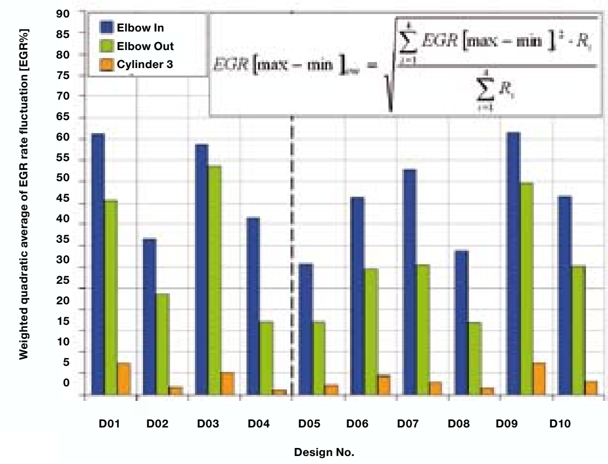

Figure 15. Cumulative EGR rate fluctuation on evaluation planes (B vehicle)

Discretization of the intake path to separate mixing zones can provide additional information on the mixing behaviour (Figs. 16 to 17). One of the two investigated sub-volumes is downstream to the mixer, up to the elbow output and the second volume is the intake manifold itself. Deriving a differential of the average EGR rate fluctuations calculated at these bounding planes can show how much the enclosed volume contributes to mixing process with the certain mixer geometry. The higher the fluctuation difference, the more mixing is performed in the corresponding sub-volume. Quadratic mean average values are calculated for both vehicle cycle types.

The lower "Mixing in Manifold" value is preferred, as this refers to a mixture at the Elbow Out plane that is already mixed well, hence the different cylinders will get the same amount of exhaust gas ratio. It means the major part of mixing occurs in the EGR Mixer or in the piping up to the Elbow output, while the presence of intake manifold contributes the minor part to cylinder-to-cylinder EGR rate distribution.

These relative results refer to the mixing length and should be observed in cope with the cumulative EGR rate fluctuations at the evaluation planes (Figs. 14 to 15) in order to see absolute values for reference.

DISCUSSION

To choose the best mixer one need to inspect each concept pursuant to square points, as possible EGR rate, engine rated performance, mixture homogeneity, and cost. As seen the different concepts are adequate for each of that square point.

The highest EGR rate would be the main goal in point of nitrogen-oxides (NOx) emission reduction view. The best mixer according to this aspect is the D01 and D04, but still good concepts are D02, D08, D09 and D05.

Lowest pressure drop would be optimal to retain the rated engine performance without the EGR system. The intake manifold charger pressure, consequently the engine torque could be the highest with D02 design. However D01, D04 and D08 concepts are also good according to this aspect.

The most homogeneous mixture at manifold inlet is important also to achieve the best reduction of the emission of NOx due to the same EGR rate at the cylinders. Additionally it means the main mixing evolves in EGR Mixer and the Elbow and the effect of the Manifold does not contribute remarkably to the cylinder-to-cylinder EGR distribution. In such a case the EGR mixer efficiency is better due to the more homogeneous mixture. So with the best efficiency mixer was D04, but D08, D05 and D02 are also good.

The lowest production cost is also important. A cheap mixer would be the main goal in point of total engine production cost. The best in this aspect is the D01 design with the simplest construction, but D02, and D05 concepts perform here also well.

A priority of the above criteria is certainly needed to be able choose the right mixer for an engine project. One choice would be the following. The main aspect is the homogeneous mixture, the second and third criteria are the possible EGR rate and the engine performance and the last but important point is the production cost. With this weighting order the proposed best mixer design options are D02 and D05. Based on these proposed concepts one can conclude that even a simple mixer design like D05 can perform well.

CONCLUSIONS

This paper intended to highlight the main questions that must be considered when designing or selecting an EGR mixer for

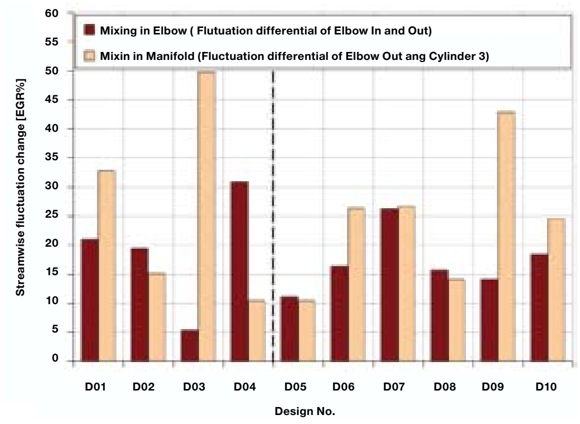

Figure 16. Contribution to EGR rate fluctuation (A vehicle)

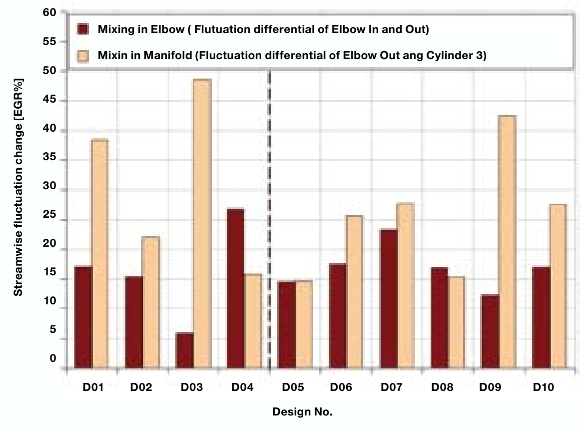

Figure 17. Contribution to EGR rate fluctuation (B vehicle)

HD commercial vehicles. An ideal case would be a mixer that mixes the components in a relative short length while exhibiting a reasonably low pressure drop in the charge air path, enabling a high average EGR level in a wide engine operation range (various mass flow rates).

The results, obtained from static CFD simulations, give an insight into the main behaviour of EGR mixers and help to elaborate the most important criteria, which are required for a successful mixer design selection.

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