NUMERICAL ANALYSIS OF THE AIR AND EGR GAS MIXTURE IN THE INTAKE MANIFOLD

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Abstract. *The standards which define the limits of the pollutant emissions produced by the automotive vehicles have become increasingly restricted, specialty for diesel engines. To meet these standards, the automotive companies have developed several technologies to reduce and control the pollutant emissions resulting from the combustion process in the engines. Among these technologies we can find the exhaust gas recirculation system, which is usually known in the industry as EGR system. This system is very functional; however the success of its operation is quite sensitive to the boundary conditions which surround its operating. One of these boundary conditions is the homogeneity of the air and EGR gas mixture which is provided to the engine cylinders. The more homogeneous were this mixture, more efficient will be the EGR system operation and lower will be the pollutant emissions generated by the engine. In this investigation aimed to evaluate numerically the mixture process. It was used the Computational Fluid Dynamics (CFD) techniques to develop the analyses. The computational code used is the FLUENT™ 6.3.26, which is a code based on the finite volume method to solve the transport equations of the fluid mechanics. The boundary conditions of the problems were defined with basis on the test cycles which the engine is submitted to be evaluated by the regulatory agencies. As it was adopted the permanent regime to make the system numerical analyses, it's presented qualitative results of the flow and the mixture process of the air and EGR gas (in the real phenomena the boundary conditions vary according to the engine operation – transient regime).*

Keywords*: diesel engine, CFD, EGR, emissions,*

1. INTRODUCTION

In 1892 the engineer Rudolf Christian Karl Diesel registered a patent for a new internal combustion engine conception. His concept was based on start the combustion with the injection of liquid fuel into a chamber filled with heated air, where the air was heated through a compression process. It allowed this new engine to reach an efficiency twice times bigger than the engines of his time, Heywood (1988). This thermal machine has been called Diesel Engine, and due its robustness, durability and fuel economy, it has been used in great part of the water and land transportation, and it's responsible to generate electrical power for farms, constructions and industrial activities, Lloyd and Cackette (2001).

Although the positive characteristics described above, the diesel engine has some negative issues. One of those issues is common for all internal combustion engines: the pollutant emissions resultant from the combustion process (Loureiro, 2005; Onursal and Gautam, 1997). As the use of this kind of engine was growing quickly, some countries started to control the pollutant emissions.

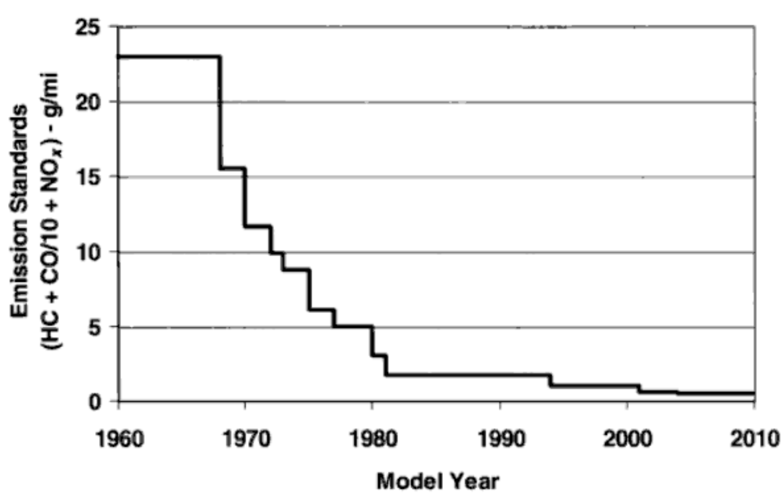

Figure 1. Evolution of the standard emissions for light duty vehicles in the USA

Figure 1 show the evolution on the American emission standards for light duty vehicles. On this chart HC (unburned hydro-carbon), CO (carbon monoxide) and NOx (nitrogen oxides) are combined only for illustration.

Among the all compounds resultant from the diesel combustion which have its limits controlled by the emission standards, the nitrogen oxides are the ones which had the biggest restriction. Figure 2 show the evolution on the American emission standards to control the NOx and the soot produced by heavy duty vehicles.

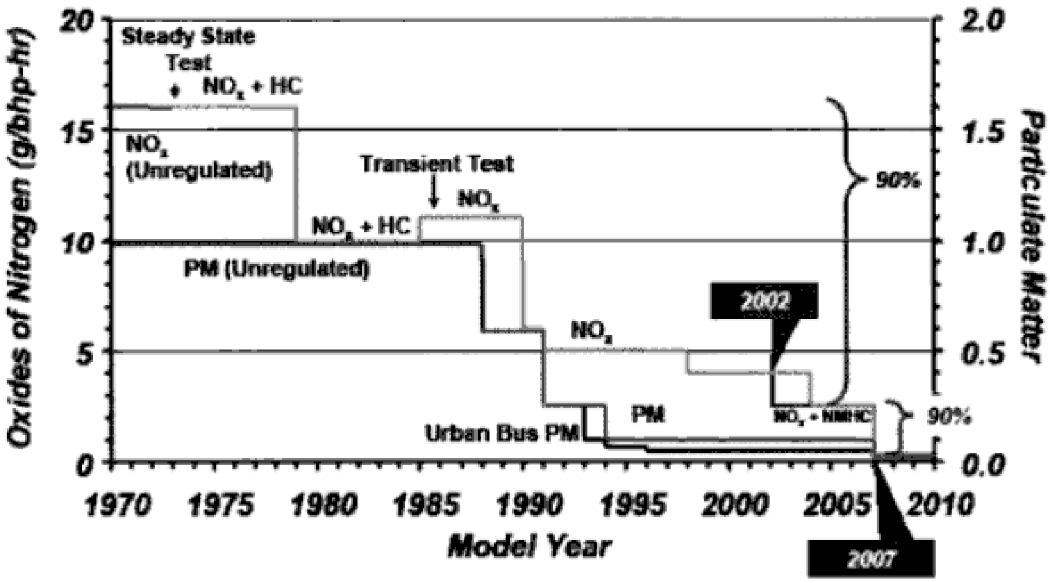

Figure 2. Evolution of the control for NOx and Soot produced by heavy duty vehicles in the USA

To follow these standards, some technologies were developed in order to reduce the NOx produced by the engine. One of them is the Exhaust Gas Recirculation, usually called EGR. This system works substituting a fraction of the air inducted into the engine with exhaust gases from the combustion. The exhaust gas mass fraction is introduced into induction system through a connection between the exhaust manifold and the intake manifold, and then is mixed with the main air flow. After that the mixture flows to the cylinders, Bosh (2004). Figure 3 show a schematic drawing of the EGR system.

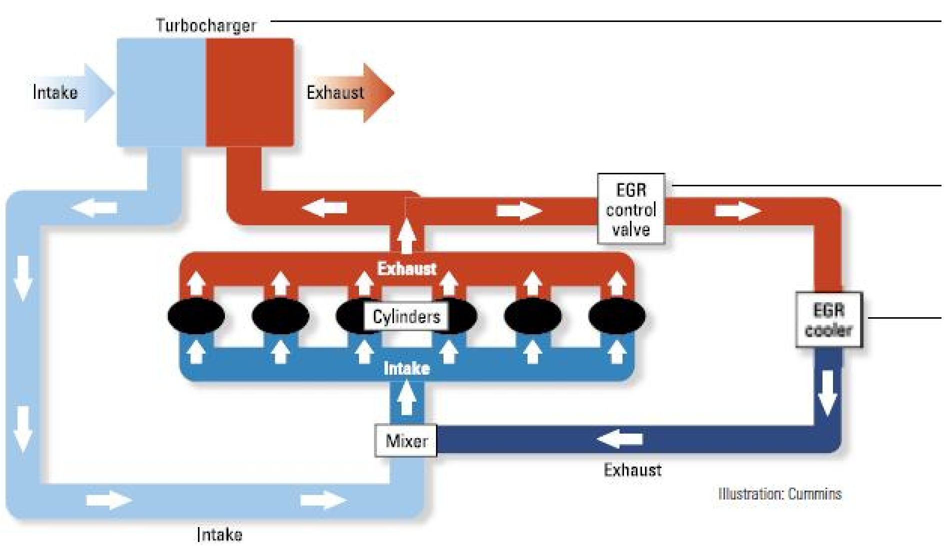

Figure 3. Schematic drawing of the EGR system

The significant increasing of thee EGR rate results on great NOx reductions. Unfortunately, big quantities of EGR mass increase the emissions of unburned hydro-carbons, CO and the formation of soot due the reduction of the oxygen in the chamber (Sher, 1998; Siewert et al., 2001; Zheng et al., 2003). Other problem regarding the use of EGR technology is to ensure the air and EGR mixture will be distributed to the cylinders in a homogeneity way. The low homogeneity mixture will cause the increase of the NOx emissions and soot. These problems are caused due the chemical composition variation and temperature variation of air and EGR mixture that flows to the cylinders (Maiboon et al., 2009). In order to facilitate the mixture process and avoid the problems described above, it's normally used a mixture device (called EGR mixer) on the engines.

This work aims to numerically evaluate the air/EGR mixture into the intake manifold with pre-chamber mixer.

2. DESCRIPTION PROBLEM

The present study focuses on the mixture Air/EGR in the intake manifold. This problem is solved considering compressible, turbulent, constant properties fluid flow where gravity and viscous dissipation effects were neglected. Based on these assumptions, numerical results of a 3D simulation are obtained for the domain showed in Fig. 4.

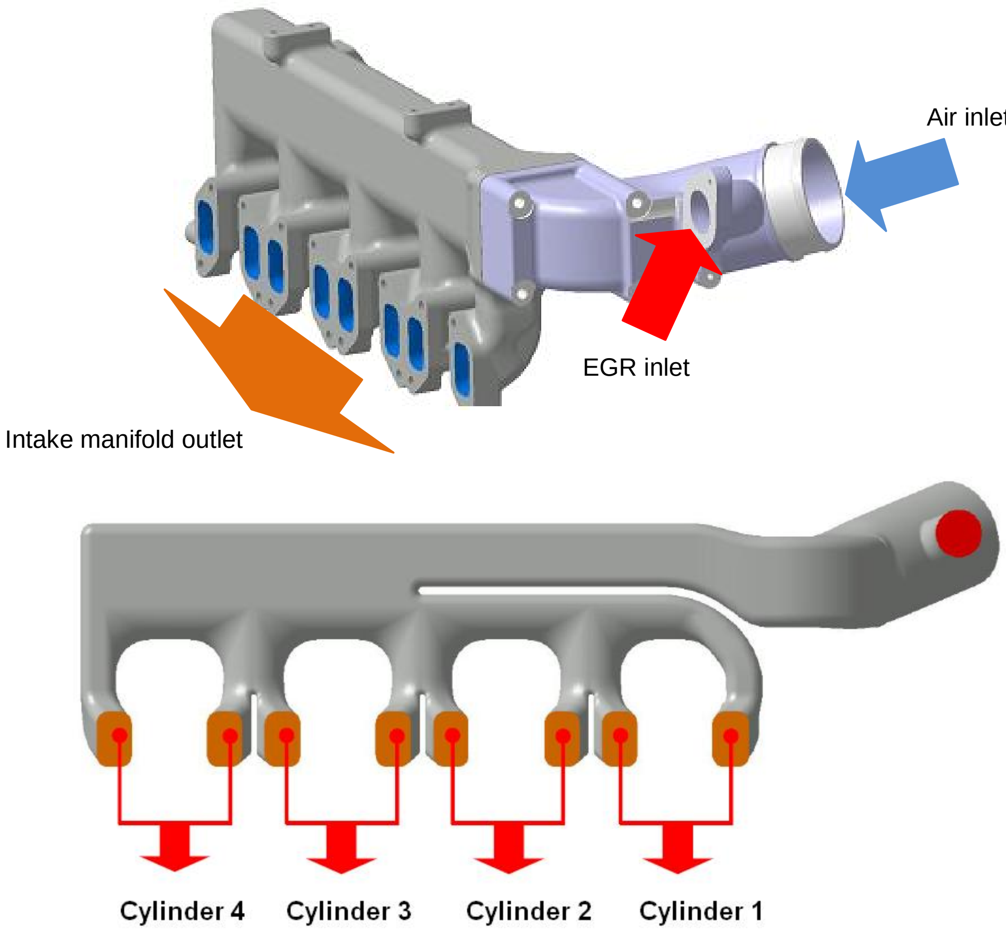

Figure 4 – Imposed boundary conditions

3. MATHEMATICAL FORMULATION

The steady-state Navier-Stokes equations for an compressible turbulent flow with constant properties fluid are presented in the three-dimensional Cylindrical coordinate system (r,θ, z). It is assumed that the viscous heating and gravity effects are negligigle.

Continuity equation:

$$\frac{1}{r}\frac{\partial(r\rho v_r)}{\partial r}+\frac{1}{r}\frac{\partial(\rho v_\theta)}{\partial \theta}+\frac{\partial(\rho v_z)}{\partial z}=0 \quad (1)$$

Momentum equation in *r*-direction:

$$\rho\left(\frac{Dv_r}{Dt}-\frac{{v_\theta}^2}{r}\right)=-\frac{\partial p}{\partial r}+2\frac{\partial}{\partial r}\left(\mu_{ef}\frac{\partial v_r}{\partial r}\right)+\frac{1}{r}\frac{\partial}{\partial\theta}\left[\mu_{ef}\left(\frac{1}{r}\frac{\partial v_r}{\partial\theta}+\frac{\partial v_\theta}{\partial r}-\frac{v_\theta}{r}\right)\right]+\frac{2\mu_{ef}}{r}\left(\frac{\partial v_r}{\partial r}-\frac{1}{r}\frac{\partial v_\theta}{\partial\theta}-\frac{v_r}{r}\right)$$

$$+\frac{\partial}{\partial z}\left[\mu_{ef}\left(\frac{\partial v_r}{\partial z}+\frac{\partial v_z}{\partial r}\right)\right] \tag{2}$$

Momentum equation in θ-direction:

$$\rho\left(\frac{Dv_\theta}{Dt}-\frac{v_\theta v_r}{r}\right)=-\frac{1}{r}\frac{\partial p}{\partial r}+2\frac{\partial}{\partial r}\left(\frac{\mu_{ef}}{r^2}\frac{\partial v_\theta}{\partial\theta}\right)+\frac{\partial}{\partial r}\left[\mu_{ef}\left(\frac{1}{r}\frac{\partial v_r}{\partial\theta}+\frac{\partial v_\theta}{\partial r}-\frac{v_\theta}{r}\right)\right]+\frac{2\mu_{ef}}{r}\left(\frac{1}{r}\frac{\partial v_r}{\partial\theta}+\frac{\partial v_\theta}{\partial r}-\frac{v_\theta}{r}\right)$$

$$+\frac{\partial}{\partial z}\left[\mu_{ef}\left(\frac{1}{r}\frac{\partial v_z}{\partial\theta}+\frac{\partial v_\theta}{\partial z}\right)\right] \tag{3}$$

Momentum equation in z-coordinate:

$$\rho\frac{Dv_z}{Dt}=-\frac{\partial p}{\partial z}+2\frac{\partial}{\partial z}\left(\mu_{ef}\frac{\partial v_z}{\partial z}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left[\mu_{ef}r\left(\frac{\partial v_r}{\partial z}+\frac{\partial v_z}{\partial r}\right)\right]+\frac{1}{r}\frac{\partial}{\partial\theta}\left[\mu_{ef}\left(\frac{1}{r}\frac{\partial v_z}{\partial\theta}+\frac{\partial v_\theta}{\partial z}\right)\right] \tag{4}$$

Energy equation:

$$\rho\frac{De}{Dt}+p\frac{D}{Dt}\left(\frac{1}{\rho}\right)=\frac{1}{r}\frac{\partial}{\partial r}\left(rk_{ef}\frac{\partial T}{\partial r}\right)+\frac{1}{r^2}\frac{\partial}{\partial\theta}\left(k_{ef}\frac{\partial T}{\partial\theta}\right)+\frac{\partial}{\partial z}\left(k_{ef}\frac{\partial T}{\partial z}\right) \tag{5}$$

Where μ_{ef} and k_{ef} are the effective (molecular +turbulent) viscosity and thermal conductivity.

State equation:

$$p=\rho RT \tag{6}$$

The standard k-ε model is a semi-empirical model based on model transport equations for the turbulent kinetic energy (k) and its dissipation rate (ε). The model transport equation for k is obtained modeling the exact equation, while the model transport equation for ε was obtained using physical reasoning and bears little resemblance to its mathematically exact counterpart, Launder and Spalding (1974).

In the k-ε model it was assumed that the flow is fully turbulent, and the effects of molecular viscosity are negligible away from solid walls.

Transport equations for the Standard k-ε model are (Nakayama, 1995):

$$\rho\frac{Dk}{Dt}=\frac{\partial}{\partial x_i}\left[\left(\mu+\frac{\mu_T}{\sigma_k}\right)\frac{\partial k}{\partial x_i}\right]+G_k-\rho_\varepsilon \tag{7}$$

$$\rho\frac{D\varepsilon}{Dt}=\frac{\partial}{\partial x_i}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x_i}\right]+C_{1\varepsilon}\frac{\varepsilon}{k}(G_k)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{8}$$

where:

G_k - represents the generation of turbulent kinetic energy due mean velocity gradients

$C_{1\varepsilon}, C_{2\varepsilon}, C_{3\varepsilon}$ - are constants

σ_k - turbulent Prandtl number for k

σ_ε - turbulent Prandtl number for ε

The turbulent viscosity, μ_t, is computed by combining k and ε:

$$\mu_t = \rho c_\mu \frac{k^2}{\varepsilon} \tag{9}$$

The boundary conditions prescribed at each domain surface are summarized in Tab. 1:

Table 1 – Imposed boundary conditions in the model

Surface	Imposed boundary condition
Inlet Air	Pressure
Inlet EGR	Mass Flow
Wall	Adiabatic
Outlet	Pressure

The values used in the simulation were obtained in the dynamometer test for the three operation conditions, Tab.2:

Table 2 – Dynamometer test data

Engine operation point	Engine speed (RPM)	Engine Load (kgf)	Inlet EGR flow (kg/s)	Inlet air pressure (Pa)	Outlet
1	2010	90,5	$3{,}05 \cdot 10^{-2}$	155045,87	Atmospheric pressure
2	1690	99,3	$2{,}44 \cdot 10^{-2}$	134169,19	
3	1369	25,6	$1{,}14 \cdot 10^{-2}$	18882,09	

With the numerical analysis concluded, it was calculated the EGR rate based on the EGR flow available for each cylinder. The EGR rate is calculated through the following equation:

$$EGR(\%) = \left(\frac{\dot{m}_{EGR}}{\dot{m}_{EGR} - \dot{m}_{AIR}} \right) x100 \tag{10}$$

where $\dot{m}_{EGR}$ and $\dot{m}_{AIR}$ are mass flow of the EGR and air, respectively.

4. COMPUTATIONAL METHODOLOGY

The numerical simulations were carried out using the commercial CFD software, FLUENT™ (2007), version 6.3.26. The finite volume scheme (Fletcher, 1998), which involves integrating the governing equations about each control volume, was applied. The governing equations were discretized using the first order upwind scheme for convective terms and solved by a density-based formulation, using the SIMPLE approach for pressure-velocity coupling.

An analysis of grid independence was performed using five different meshes. Table 3 summarizes the number of elements for each mesh.

Table 3 – Mesh

Mesh	*Elements number*
Mesh 1	57,845
Mesh 2	116,984
Mesh 3	214,636
Mesh 4	332,209
Mesh 5	529,478

The simulations were performed using a mesh with hybrid elements tetrahedral, Fig. 5.

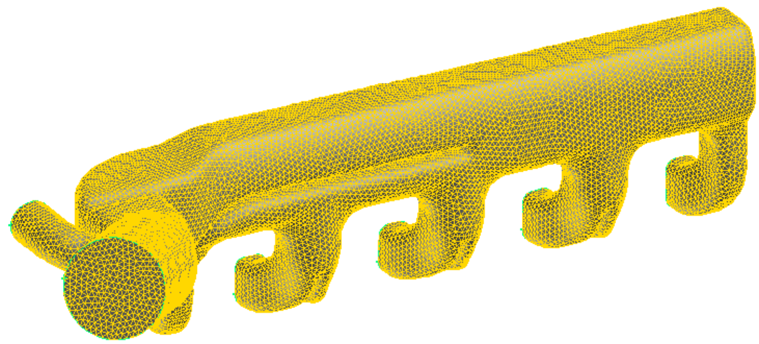

Figure 5 – Example of mesh used in the simulations.

5. RESULTS

Figure 6 shows the results of the grid independence analysis. Based on that, we can conclude there are no significant variations on EGR rate received by each cylinder for meshes with more than 214,000 cells.

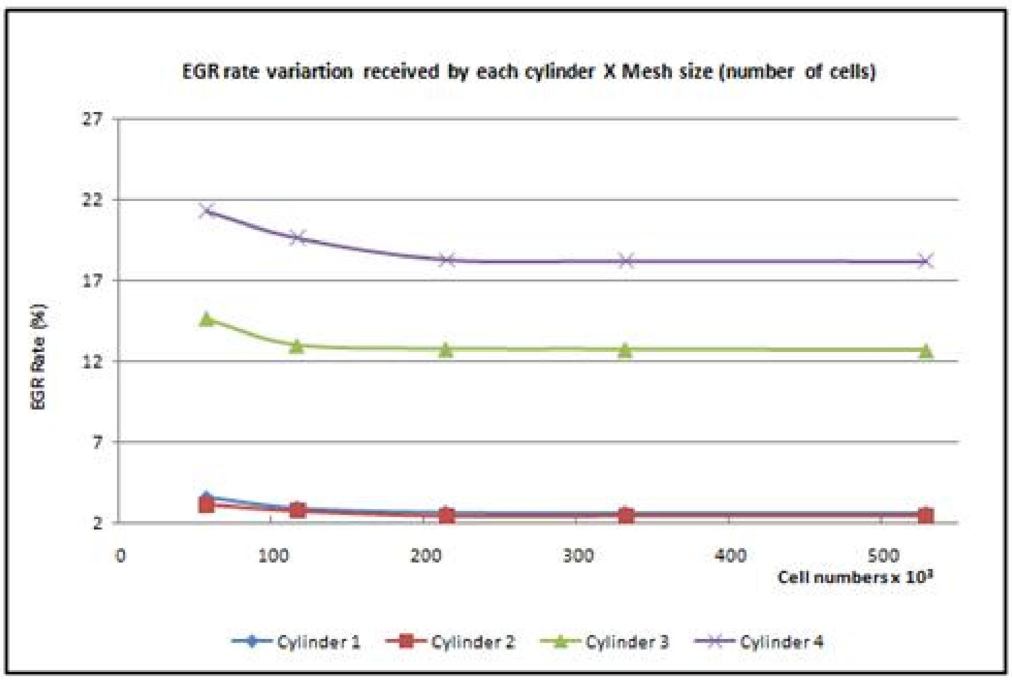

Figure 6. Grid independence analysis results

According to the methodology described and the mesh size defined by the grid independence analysis it was analyzed the EGR system configuration intake manifold with pre-chamber. Figure 7 shows the results of the intake manifold with pre-chamber mixer analysis. As show the results, there are big discrepancies on the EGR rates values among the cylinders. As mentioned, this condition can increase the NOx emissions and soot formation.

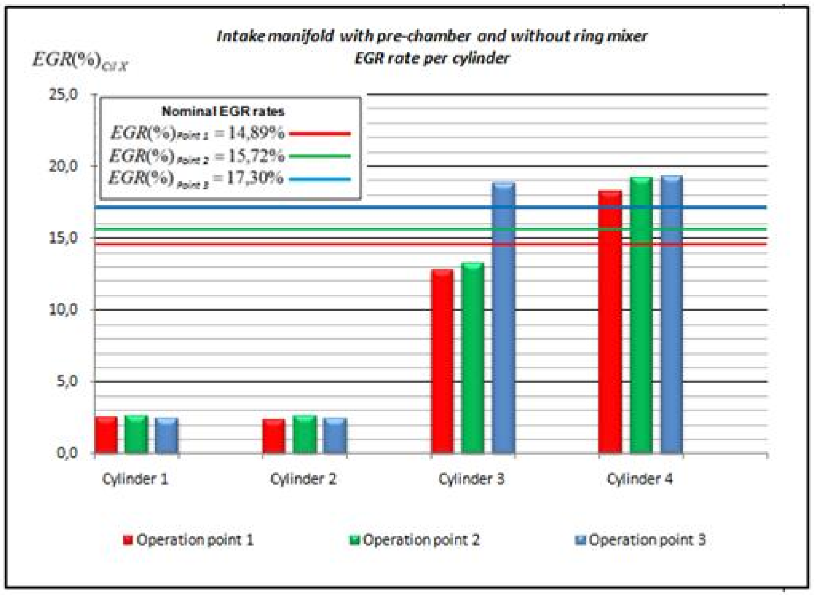

Figure 7. EGR rate per cylinder – Pre-chamber mixer

Figure 8 shows the EGR mass fraction contours. It's possible to check the mixture provided to the cylinders 3 and 4 is much richer in EGR than the mixture provided to the first two cylinders.

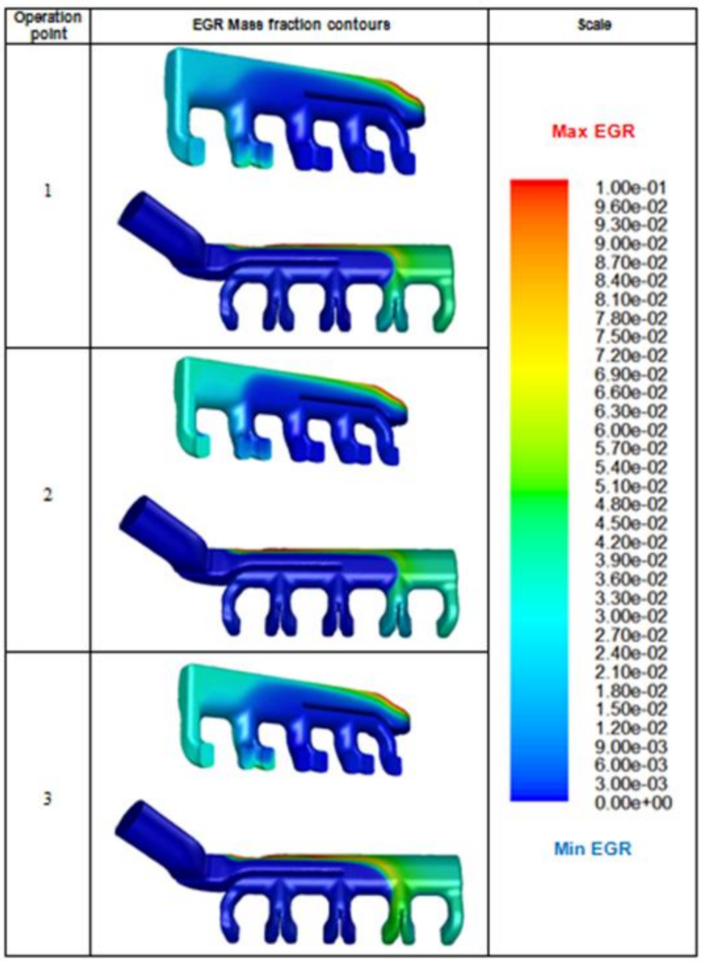

Figure 8. EGR mass fraction contours

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The presented results don't consider the opening and closing of the valves due to complexity of the phenomena. For the qualitative analysis, the results show that the mal-distribution of the mixture is mainly caused by intake manifold geometry. On the real phenomena, with the opening and closing of the valves, the distribution of the mixture would be more homogeneous on the intake manifold outlet than on considered situation.

6. FINAL COMMENTS

Even with the simplifications considered on the analysis and knowing that on the real phenomena the opening and closing of the valves facilitate the mixture process, it´s possible to conclude that intake manifold geometry is not the optimal design for the engine. Due the internal wall, which divides the intake manifold in two volumes, the drop pressure imposed to the flow gets to the cylinders 1 and 2 is much bigger than the drop pressure imposed to the flow gets to the cylinders 3 and 4. As the mixture doesn´t get homogenous enough on the pre-chamber (before the outlet ports), and with the drop pressure discrepancies among the cylinders, the cylinders 1 and 2 receive a poor EGR mixture and the cylinders 3 and 4 receive a rich EGR mixer.

For a qualitative analysis and to guide the designing process of the intake manifold, the results obtained are very satisfactory. If quantitative results were required, it´ll be necessary to couple the CFD code with a 1D code of the engine in order to transform the stationery boundary condition in transient. With this the opening and closing of the valves will be considered on the flow, resulting in a situation closer to the real phenomena. The 1D code must be developed and calibrated based on dynamometer bench test data.

7. ACKNOWLEDGEMENTS

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