CFD ANALYSIS OF THE EFFECT OF THE EXHAUST MANIFOLD DESIGN ON THE CLOSE-COUPLED CATALYTIC CONVERTER PERFORMANCE

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Abstract

It is a common practice to mount a catalytic converter directly at the outlet of the exhaust manifold in order to reduce the cold start emissions from the automotive engines by improving the light-off time. The so called Close-Coupled Converter, because of its nearness to the engine, is more exposed to non-uniform fluid flows coming from the individual manifold runners, what frequently yields a stream of gases flowing mainly through a section of the monolith causing the aging of the catalyst and low conversion efficiency, among other negative effects. Computational Fluid Dynamics (CFD) has became a very useful and widely used tool to analyze and optimize this kind of exhaust after-treatment systems in a relatively fast and accurate way for design purposes. There are several designs of exhaust manifolds whose shape and dimensions are mainly restricted not only by the engine characteristics but by the space constraints of the particular vehicles they are designed for. In this work the commercial CFD code ANSYS FLUENT was used to evaluate and compare the effect of three shapes of exhaust manifolds on the fluid flow uniformity at the entrance of the Close-Coupled Converter. Pressure drop in the entire Manifold-Converter device is also an important parameter considered for the design evaluation. The manifolds investigated are of type cast, 4-2-1 and L-shaped, which are commonly used in the automotive industry.

Keywords: *Close-Coupled Catalytic Converter, Exhaust Manifold, Flow Uniformity, Pressure Drop*

1. Introduction

Nowadays, as the emission regulations become more stringent, there are many investigations which search the fuel economy and better emission performance. The catalytic converter has been adopted since some years ago as a very effective device which allows the conversion from harmful compounds to less damageable ones, mainly the conversion from Nitrogen Oxides, hydrocarbons and Carbon Monoxide to Nitrogen, water and Carbon Dioxide. Such a device, generally used for gasoline engines, is called Three Way Catalyst because of the treatment of these three compounds.

Recently with the introduction of the more stringent emission regulations, many improvements have been studied in order to achieve better efficiencies in the reduction of pollutant emissions. A very important way for improving the exhaust system emission performance is found by decreasing the light-off time of the catalytic converter, i.e. the time which takes for the monolith temperature raises the optimum temperature, and so reducing the cold start emissions. One measure to accomplish that is by heating the substrate either electrically or by means of post-cylinder combustion [3, 7], but a drawback of this is the additional energy supply which is required. Other considerable approach to increase the converter temperature quickly is to mount it directly to the exhaust manifold [3, 6, 7], with the main restrictions being the lack of space around the engine and the challenging and substantial necessity of having a well distributed fluid flow at the catalyst entrance, otherwise the conversion efficiency and durability of the catalytic converter may be compromised.

W. Guojiang and T. Song [4] as well as J. Windmann et al. [11] investigated the effect of the inlet flow distribution on the catalytic converter performance and observed that the lower flow uniformity index results in longer light-off time and lower conversion efficiencies for all pollutants. Furthermore, the non-uniform flow distribution, a catalyst located very close to the cylinder head, and direct pulsating flow with a certain angle of incidence could all impact on the brick damage of a Close-Coupled thin wall catalyst [8].

Among the most important factors involved on the flow uniformity at the inlet of the Close-Coupled Converter are those related to the geometric features of the exhaust manifold, mixing pipe and inlet cone designs as well as the distance between the junction of the runners and the monolith. These factors also influence the pressure drop through the exhaust system, which represents one of the main parameters to be optimized because of its impact on the engine output power; engine output power is reversely proportional to exhaust back pressure [12]. Hyo Gyu Chang et al [2] performed a transient CFD analysis, for a 4-1 type exhaust manifold and CCC, varying the length of the mixing pipe. Their results revealed that long junctions can achieve better flow uniformity than those ones with smaller length because exhaust systems with longer mixing zones have enough time and space for mixing gases from the exhaust manifold. Sophie Salasc et al [8] compared the flow distribution of three manifold-CC catalytic converter system designs by using hot wire anemometry and CFD calculations. The results confirmed the flow distribution is more uniform by adding a pre-mixing chamber between the manifold and the catalyst, moreover, the flow distribution gets more affected by the flow rate for a 4-1 type manifold compared with an integrated catalyst manifold with and without pre-mix chamber. In Goo Hwang *et al.* [5] analyzed two kind of exhaust manifold by experimental tests and unsteady CFD simulations. The study showed conventional cast type manifold yields higher flow uniformity and lower back pressure than the bending type.

This paper presents results of steady state 3D CFD simulations which were used to compare the flow uniformity and pressure drop through three different exhaust manifold designs; the conventional cast manifold, a 4-2-1 type manifold and an *L*-shaped manifold, all of them with four runners.

2. Numerical Analysis

COMPUTATIONAL DOMAIN AND MESH

Computational meshes for the three different designs of manifold – CCC are depicted in the Fig. 1. Hybrid meshes (hexahedral-tetrahedral) are constituted by 474773, 708648 and 565811 cells for the cast manifold, the 4-2-1 manifold and the *L*-shaped manifold, respectively. In order to evaluate the effect of the manifold geometry on the flow distribution and the overall pressure drop, the three designs shared the same geometry from the inlet cone of the converter to the outlet tube. Also, the distance from the junction of the manifold runners to the inlet cone, i.e. the length of the mixing pipe, for the three cases is quite similar although the *L*-shaped manifold design results a bit different. In the conventional cast manifold the four runners are joined firstly two by two and then the two couples got joined softly, while for the 4-2-1 manifold two couples of runners are joined as two parallel streams converging to a mixing pipe. The here called *L*-shaped manifold is quite different compared to the other two configurations since it is basically composed by a wide straight pipe with four runners connected obliquely. Fig. 1 shows the manifold runners which have been identified with names.

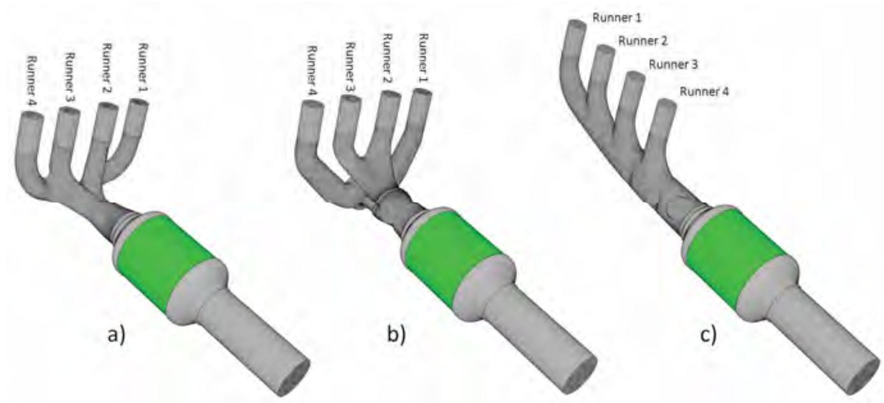

Fig. 1. Computational domain and meshes for a) cast manifold, b) 4-2-1 manifold and c) L-shaped manifold

MODEL

The commercial software ANSYS FLUENT was used to perform the calculations of the three-dimensional, turbulent and non-reacting flow. Four individual steady state simulations considering one manifold runner working at time were carried out, using the total engine flow rate at the runner inlet in order to study four characteristic stages of the real transient flow, which are expected to yield the worst possible flow distributions at the inlet of the monolith [9]. For the analysis the assumed working fluid is air treated as an ideal gas:

$$\rho = \frac{p_{op} + p}{\frac{R_u}{M} T}, \tag{1}$$

where:

ρ - density,
p_{op} - the operating pressure,
p - the local static pressure relative to the operating pressure,
R_u - the universal gas constant,
M - the molecular weight,
T - the local temperature.

The three coefficients Sutherland's law was employed to consider the temperature dependence of viscosity:

$$\mu = \mu_0 \left(\frac{T}{T_0} \right)^{3/2} \frac{T_0 + S}{T + S}, \tag{2}$$

where:

μ - the viscosity,
μ_0 - the reference viscosity,
T_0 - the reference temperature,
S - the Sutherland constant.

The coefficients had the following values for air [10]:

$$T_0 = 273\,\mathrm{K},\ \mu_0 = 1.716 \times 10^{-5}\ \mathrm{N \cdot s/m^2},\ S = 111\,\mathrm{K}.$$

The continuity, momentum and energy equations were solved through the commercial code:

$$\nabla \cdot (\rho \vec{v}) = 0, \tag{3}$$

$$\nabla \cdot \left(\rho \vec{\upsilon} \vec{\upsilon}\right) = -\nabla p + \nabla \cdot \left(\overline{\overline{\tau}}\right) + S_V \ , \tag{4}$$

$$\nabla \cdot \left(\vec{\upsilon}\left(\rho E + p\right)\right) = \nabla \cdot \left(k_{eff} \nabla T\right) + S_H \ , \tag{5}$$

where:

$\vec{\upsilon}$ - the vector velocity,
$\overline{\overline{\tau}}$ - the stress tensor,
S_v - a momentum source term,
E - energy,
k_{eff} - the effective conductivity,
S_H - energy source term.

The turbulent effects were included using the standard κ–ε model with standard wall function treatment, which includes the equations for the turbulence kinetic energy and the turbulence dissipation rate:

$$\frac{\partial}{\partial x_i}\left(\rho \kappa u_i\right) = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial \kappa}{\partial x_j}\right] + G_\kappa + G_b - \rho\varepsilon - Y_M + S_\kappa \ , \tag{6}$$

$$\frac{\partial}{\partial x_i}\left(\rho \varepsilon u_i\right) = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_{1\varepsilon}\frac{\varepsilon}{\kappa}\left(G_\kappa + C_{3\varepsilon}G_b\right) - C_{2\varepsilon}\rho\frac{\varepsilon^2}{\kappa} + S_\varepsilon \ , \tag{7}$$

where:

κ - the turbulence kinetic energy,
ε - the turbulence dissipation rate,
G_k - the generation of turbulence kinetic energy due to the mean velocity gradients,
G_b - the generation of turbulence kinetic energy due to buoyancy,
Y_M - the contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate,
μ_t - the eddy viscosity,
σ_κ - the turbulent Prandtl number for κ, σ_ε - the turbulent Prandtl number for ε, S_κ and S_ε are source terms,
$C_{1\varepsilon}$, $C_{2\varepsilon}$, $C_{3\varepsilon}$ - constants.

The model constants had the following default values which have been found to work fairly well for a wide range of wall-bounded and free shear flows [1]:

$$C_{1\varepsilon} = 1.44, \quad C_{2\varepsilon} = 1.92, \quad C_\mu = 0.09, \quad \sigma_\kappa = 1.0, \quad \sigma_\varepsilon = 1.3.$$

The fluid flow within the domain of the catalytic converter substrate is treated as laminar flow in porous medium because of the low Reynolds number occurring in this zone due to the small hydraulic diameter of the monolith passages. The relationship between the pressure drop and the velocity is settled according to the following equation [1]:

$$\frac{\partial p}{\partial x_i} = -\left(\frac{\mu}{\alpha}\upsilon_i + C_2\frac{1}{2}\rho\left|\upsilon\right|\upsilon_i\right), \tag{8}$$

The first term on the right-hand side of equation (8) comes from the Darcy's law, whereas the second term represents the inertial resistance and this latter becomes important for high velocities. The coefficients $1/\alpha$ (the reciprocal of permeability) and C_2 are characteristic values determined by the specific substrate nature. Since in a real substrate the fluid flow is mainly axial inside the monolith, the axial resistance coefficients were calculated for the specific conditions here studied and the values corresponding to the radial direction were fixed three orders of magnitude higher than the axial ones to ensure a negligible radial flow.

The numerical calculations were carried out using double precision format, standard interpolation scheme for pressure and second order Upwind scheme for all convection terms. The SIMPLE algorithm was employed for the linking of pressure and velocity. As convergence criterion, all the residuals must drop around 1×10^{-3}.

BOUNDARY CONDITIONS

Mass flow rate of 90 g/s at 900°C of total temperature is prescribed at the inlet of the operating manifold runner in each simulation (one runner at a time), which corresponds to the maximum charge of a typical 4 cylinder engine. At the outlet, a relative static pressure of 30 kPa is settled, what is typically expected in this part of the exhaust system. Heat transfer from the converter to the environment can be considered negligible as a first approximation since this is not our main concern, thus adiabatic condition was used for all the walls.

Resistance coefficients for a typical 600 cpsi/4.3 mil, 0.9 L catalytic converter were used in this study: $1/\alpha = 8.3 \times 10^{7}$ m^{-2} and $C_2 = 9.5$ m^{-1} for the axial direction. As commented before, three orders of magnitude higher values were established for the radial direction.

EVALUATION OF THE FLOW UNIFORMITY

As mentioned above, the fluid flow distribution across the inlet face of the catalytic converter has significant effect on the exhaust emission conversion efficiency and the durability of the catalyst. A quantitative parameter named *flow uniformity index*, usually represented as γ, is widely used in the automotive industry to evaluate the flow distribution at the catalyst inlet. Based on the definition originally performed by Herman Weltens [9], and using area-weighted averages to remove the effect of the non-uniform mesh while computing mean values, the flow uniformity index can be expressed as:

$$\gamma = 1 - \frac{1}{2}\frac{\sum_{i=1}^{n}\left|w_i - w_{mean}\right|A_i}{A w_{mean}}, \tag{9}$$

where:
w_i - the local velocity,
A_i - local area,
A - the cross area where γ is evaluated,
and the mean velocity is computed as:

$$w_{mean} = \frac{1}{A}\sum_{i=1}^{n} w_i A_i , \tag{10}$$

with *i* as the local grid cell and *n* as the number of grid cells within the cross-section plane.

According to the above definition, γ could be as high as 1 when the flow at the cross area is completely uniform. Generally, modern exhaust manifolds have high uniformity index over 0.90, nevertheless, higher uniformity index over 0.95 is required to meet the stringent vehicle emission regulation (e.g. PZEV) [5]. In this analysis, flow uniformity index has been measured on a cross-sectional plane located inside the substrate at 10 mm from its inlet.

3. Results

CAST MANIFOLD

Figure 2 shows the velocity distribution normalized with the area-weighted average velocity (w_i/w_{mean}) on a cross-sectional plane located inside the catalyst at 10 mm from its entrance when each runner of the cast type manifold is operating individually. Also, the flow uniformity indexes are indicated beside their corresponding pictures in Fig. 2. High flow uniformity indexes, above 0.96, for all the runners are encountered, although it can be clearly observed the existence of peak velocities whose position depends on the geometry of the open runner. The bends of a runner cause the flow to be discharged asymmetrically, so the relative location of the manifold defines the way the flow is discharged over the catalyst front face even though the concentric cone and the straight outlet tube used for this analysis promote a main flow through the centre of the substrate cross-section. As it is shown in the contours of normalized velocities, the velocity changes are lower than 20% of the mean velocity for all the runners.

Figure 3 shows contours of total pressure for the cast manifold. The total pressure distribution is preferred in this paper to illustrate the aerodynamics within the system because it allows evaluating the pressure which cannot be recovered. It can be appreciated the main pressure drops occur within the manifold and the catalyst; catalyst yields a pressure drop about 10 kPa, whereas the manifold generates total pressure drops around 14.6, 12.2, 10.8 and 14.3 kPa for runners 1 to 4, respectively. Such computed values elucidate the similarity and symmetry between opposed runners, e.g. runner 1 and runner 4. The overall total pressure drop is between 21 and 25 kPa for this design.

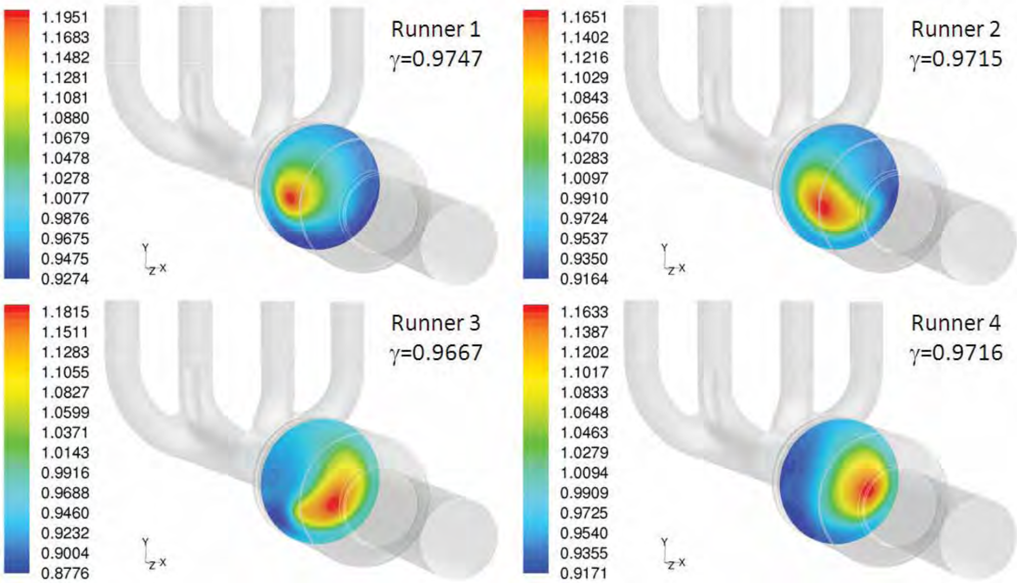

Fig. 2. Contours of normalized velocity (w_i/w_{mean}) for cast type manifold

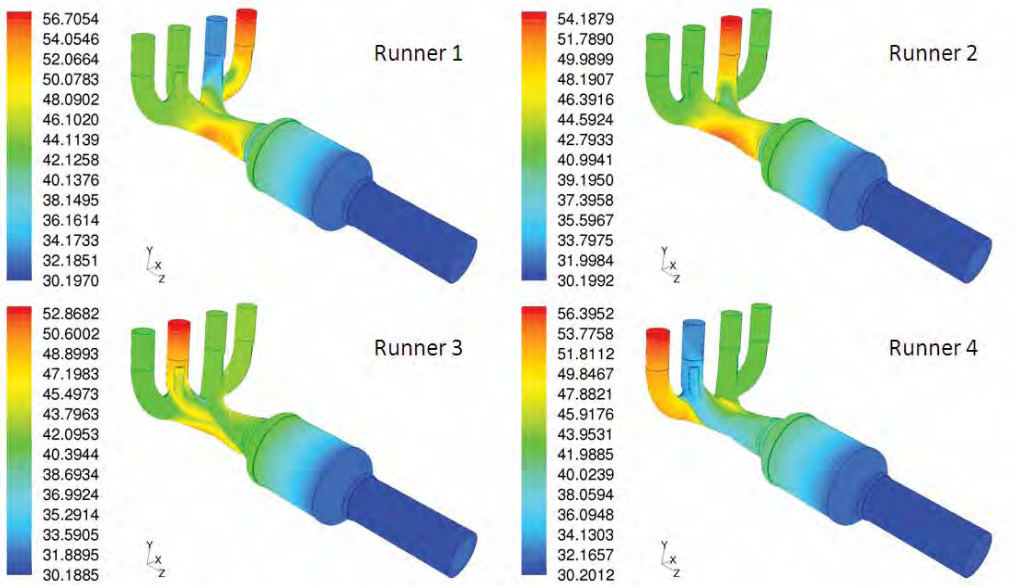

Fig. 3. Contours of total pressure (kPa) for cast type manifold

4-2-1 MANIFOLD

Normalized velocity distributions for the 4-2-1 manifold are depicted in the Fig. 4. Flow uniformities about 0.98 exist for this manifold design, what agrees with a good flow distribution with a narrow velocity range with variations lesser than 11% around the mean value. On the other hand the 4-2-1 manifold is less symmetric compared with the cast manifold design what yields velocity distributions with more than a couple of distribution patterns.

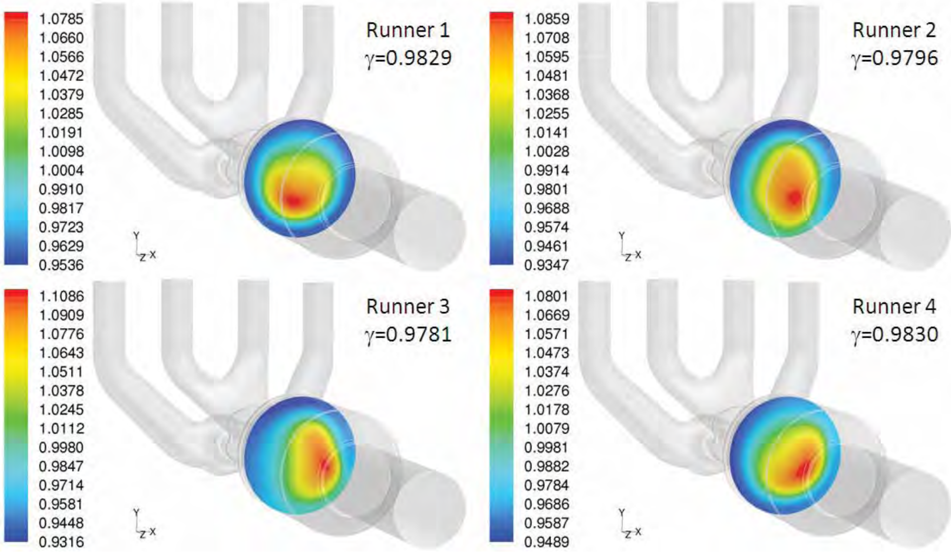

Fig. 4. Contours of normalized velocity (wi/wmean) for 4-2-1 type manifold

Contours of total pressure for the 4-2-1 type are shown in Fig. 5. Pressure drop in the catalyst is not significantly influenced by the manifold design due to the fact that the mass flow rate flowing across the substrate is the same for the three analyzed designs and there are not pressure differences between them that could originate important changes in the average velocity inside the converter.

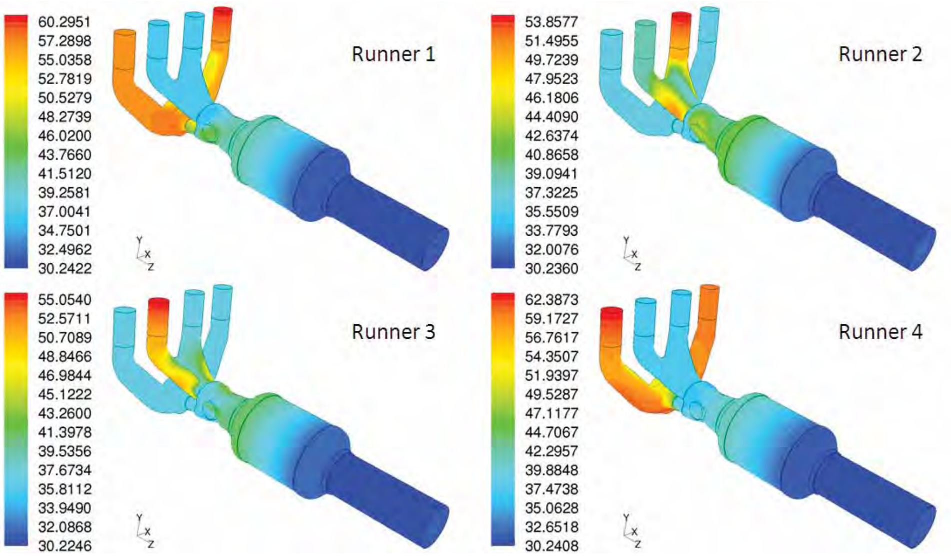

Fig. 5. Contours of total pressure (kPa) for 4-2-1 type manifold

Total pressure drops are 18.0, 11.7, 13.0 and 20.3 kPa for runners 1 to 4, respectively. As it can be appreciated in the figures of this particular design, the geometries of the outer runners, i.e. runners 1 and 4, are not appropriate to guide the flow towards the mixing pipe and cause a stagnation zone against each other, and this issue could be relevant since the pressure drop for these runners is bigger than those corresponding to the inner ones. Moreover, the abrupt expansion of the stream at the junction of the runners yields a significant pressure drop. Overall total pressure drop between 22 and 31 kPa exists for this design.

L-SHAPED MANIFOLD

Figure 6 shows the normalized velocity distribution for the *L*-shaped type manifold. The velocity contours indicate the flow distribution gets a little bit better as the distance from the open runner to the substrate increases. The peak velocity is approximately 11% higher than the average velocity for runner 4, while the velocity variation is below 8% for each one of the other runners. The flow from runner 1 is guided by the elbow and the flow from runner 2 behaves pretty similar, on the other hand runners 3 and 4 got more intense changes on the flow direction. In spite of the velocity variations, high flow uniformity indexes are encountered for all runners, especially those which are closer to the catalyst. This manifold design results with flow uniformity indexes about 0.98.

Figure 7 shows contours of total pressure for the *L*-shaped manifold configuration. The total pressure drop for this manifold is 9.7, 9.4, 9.5 and 9.2 kPa for the runners 1 to 4, respectively. Overall pressure drop for this *L*-shaped manifold is almost the same for all the runners with values in the range of 20 to 20.6 kPa.

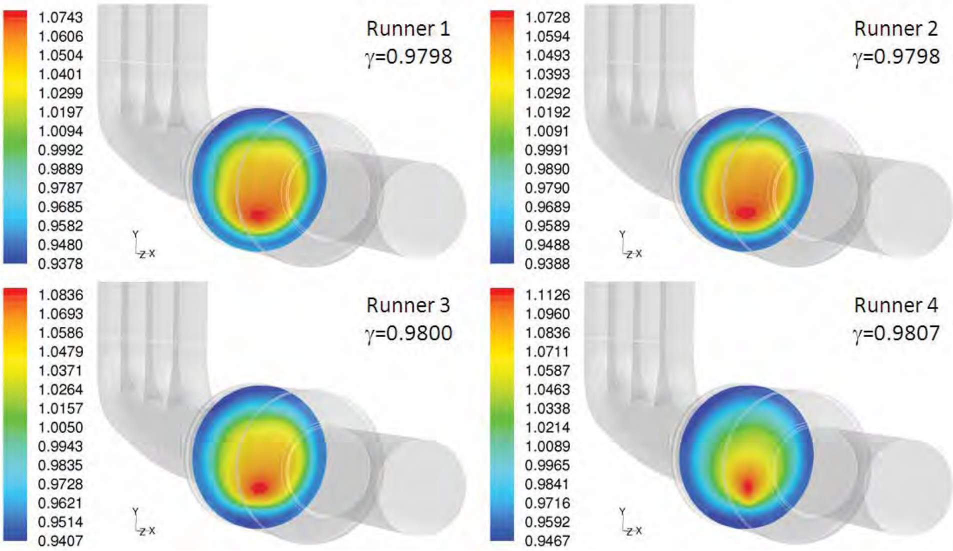

Fig. 6. Contours of normalized velocity (wi/wmean) for L-shaped type manifold

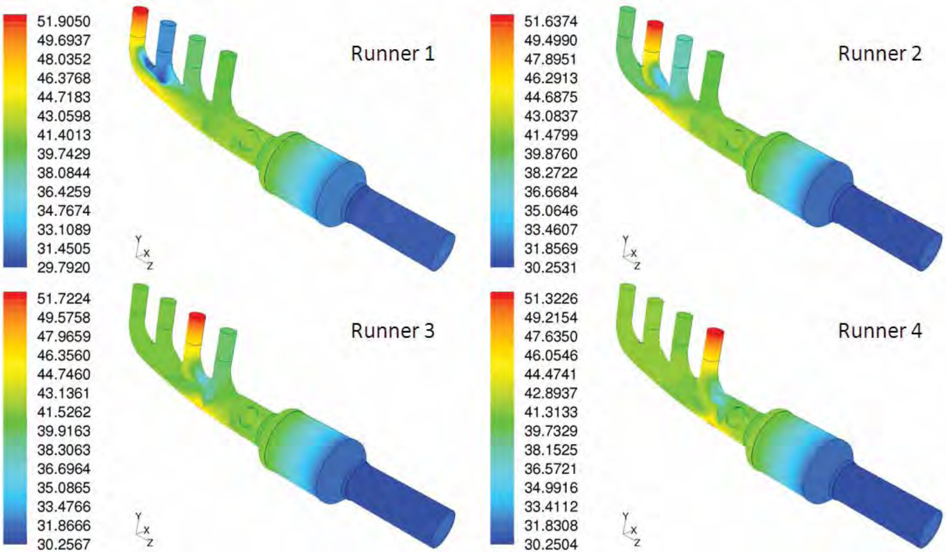

Fig. 7. Contours of total pressure (kPa) for L-shaped type manifold

4. Conclusions

In this study, steady state CFD simulations were performed in order to compare three different exhaust manifold designs, evaluating the relationship between their particular geometry and the performance of a Close-Coupled Converter. The quantitative parameters used for such a comparison were a flow uniformity index at 10 mm from the catalyst entrance and the overall pressure drop. The results show high flow uniformity indexes above 0.96 for all the manifold types: cast, 4-2-1 and the here called as *L*-shaped. Regarding the flow distribution, both 4-2-1 and *L*-shaped types were found to be slightly superior to the cast type, with flow uniformity indexes around 0.98 and tighter ranges for velocity variation, i.e. flatter velocity distribution. It is known that different locations for the peak velocity from each manifold runner, as found for cast and 4-2-1 types, could improve the monolith warm-up, nevertheless such a condition could avoid the interaction between flows coming from cylinders with different air/fuel mixture. This last issue could yield decrement of conversion efficiencies. Moreover, the system with the *L*-shaped manifold has the lowest total pressure drop, followed by the cast manifold. The worse case regarding the total pressure drop is the 4-2-1 type with up to 55% higher total pressure drop compared to *L*-shaped type.

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