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NUMERICAL ANALYSIS OF THE FUEL FLOW IN A COLD START SYSTEM EQUIPPED WITH AN ELECTRONIC FUEL INJECTOR

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Abstract. *Cold start systems provide the introduction of gasoline inside the engine intake system during cold start and warm-up acceleration, as ethanol fuel shows engine operation difficulties at low temperatures. Despite the technological evolution, traditional cold start systems may present problems mainly related to flow control and gasoline distribution to the engine cylinders hindering cold start and warm-up operation with the increasing of emissions levels and fuel consumption. The cold start system used allows for simultaneous gasoline introduction through calibrated orifices in the four conduits of the intake system. Adjustment of gasoline flow is made by an electronic fuel injector. The objective of this work is to analyze numerically the gasoline flow in an auxiliary cold start system for engines fuelled by ethanol or ethanol-gasoline blends (flex fuel). Computational fluid dynamics made possible to determine the calibrated hole diameters for more uniform distribution of gasoline to the engine cylinders.*

Keywords*: cold start, ethanol, fuel injection, internal combustion engine,numerical method*

1. INTRODUCTION

From the beginning of the 21st century, there was an increasing interest in using ethanol as fuel, due to high petrol price, which has long been the main energy source, together with the need to decrease global warming due to high level of pollutant air emission. The world has also seen a growing use of flexible fuel vehicles, which allows for the use of ethanol or ethanol-gasoline blends. Those vehicles are equipped with a cold start system that introduces the gasoline in the intake system in order to promote start-up and good performance at low temperature, especially when the vehicle is fueled by ethanol only or by a high ethanol proportion blend. In such case use of gasoline at cold start and warm-up is extremely necessary, considering that ethanol presents high latent heat of vaporization and ethanol distillation starts at the temperature around 78° C. Under normal conditions, an ethanol fueled engine will not start at a temperature below 13° C (Silva and Sodré, 2000).

Despite all technological developments related to cold start and solutions that minimize the problems when ethanol fuel is used at low temperature, the cold start systems are still object of research, aiming at the improvement of vehicle performance and exhaust emissions. At present automakers have been employing an auxiliary system of gasoline supply consisted of a small gasoline tank, an electrical pump and an electrical valve, which adds gasoline into the air intake system through a tube with one or more calibrated holes. Gasoline is added during cold start and warm up acceleration. The amount of injected gasoline is controlled by the engine electronic control module by means of the electrical pump and the electrical valve. The control of the quantity of injected gasoline into the intake manifold and the distribution among the cylinders influences cold start and warm up acceleration, exhaust emissions and fuel consumption.

The objective of this work is to analyze numerically gasoline flow in an auxiliary cold start system for engines fuelled by ethanol and ethanol-gasoline blends, called flexible fuel. The cold start system analyzed presents an electrical injector (nozzle) that controls the gasoline flow in a fuel distributor, which can be connected to an intake manifold. The fuel distributor is composed by an input and four outputs with calibrated holes, meaning that there is one calibrated hole for each engine cylinder. The major difference between the system proposed and the conventional ones is the presence of an electrical injector that allows for a precise adjustment to the gasoline flow according to the engine operating condition. The electrical injector is generally designed with one or more holes in its end. In the system with fuel distributor here presented, the calibrated holes are displayed in each ramification of the intake conduit. The numerical analysis applied concepts of computational fluids dynamics to determine the calibrated holes diameter in order to obtain an efficient gasoline atomization and uniform distribution to the engine cylinders.

2. LITERATURE REVIEW

Concerning cold start and acceleration during warm up in an ethanol fuelled engine, many works proposed the optimization of the feeding fuel system or methodologies and systems to blend more volatile composites. Annand and Gulder (1980) performed experiments with an engine using ethanol as fuel. The results collected from a carburetor equipped with an atomizer performed showed a quicker cold start in comparison to the conventional carburetor.

Kito et al (1998) investigated ethanol dissociation into diethyl ether and water steam to improve cold start. With conversion from 40 to 80% of ethanol into diethyl ether and water, cold start at temperatures much lower than that limit of pure ethanol was obtained.

Silva and Sodré (2000) evaluated the use of methyl tert-butyl ether (MTBE) in ethanol for cold start improvement. Pure ethanol was not able to start up the engine at temperatures below 13°C, agreeing with the observations by Kito et al (1998). A mixture of 75% ethanol and 25% MTBE allowed for cold start at temperatures as low as -6°C.

At present, addition of gasoline has constituted the main resource used to improve cold start in ethanol-fueled engines. Researches have shown that addition of high volumes of gasoline into methanol made possible reduction of the minimum cold start temperature, as was observed in tests performed by Iwai et al. (1981). Haatela and Decker (1989) showed that addition of 10% gasoline in methanol propitiated the engine to reach start up temperatures down to -25ºC.

Recently, Sales and Sodré (2002) performed comparative tests of conventional cold start system (gasoline addition through a tube with calibrated holes) with a system equipped with an electrical injector in the intake manifold of an ethanol-fueled engine. It was shown that the use of an electrical injector allowed for adequate addition of gasoline to shorter, regular cold start, mainly due to the effect of atomization and better control of the gasoline quantity injected. Cold start and warm up acceleration were performed using a leaner fuel/air mixture (less amount of ethanol and gasoline). Later, Sales and Sodré (2003) demonstrated it was also possible to reduce considerably the level of air pollutant emission (hydrocarbons, carbon monoxide and aldehydes) from an ethanol-fueled engine during warm up.

3. METHODOLOGY

3.1. Numerical Model

In this work a mathematical model for the flow inside the injection system based on a turbulent stationary flow of a viscous incompressible fluid was used. The flow regime of interest is modeled by the Reynolds-averaged Navier-Stokes equations (Ilinca et al., 1997):

Continuity equation

$$\nabla.\vec{u} = 0 \tag{1}$$

Momentum equation

$$\rho\vec{u}.\nabla\vec{u} = -\nabla p + \nabla.\left[(\mu + \mu_T)\left(\nabla\vec{u} + \nabla\vec{u}^T\right)\right] \tag{2}$$

where ρ represents the density, p the pressure, u the velocity vector, μ the fluid viscosity and μ_T the turbulent viscosity.

Due to its robustness, economy and acceptable results obtained for a considerable amount of flows, the k-ε model has been the most used model for numerical predictions of industrial flows (Deschamps, 1998). In this model the turbulent viscosity is computed as

$$\mu_T = \bar{\rho} c_\mu \frac{k^2}{\varepsilon} \tag{3}$$

The system is closed by including the transport equations for k and ε,

$$\rho\vec{u}.\nabla k = \nabla.\left[\left(\mu + \frac{\mu_T}{\sigma_k}\right)\nabla k\right] + \mu_T P(\vec{u}) - \rho\varepsilon \tag{4}$$

$$\rho\vec{u}.\nabla\varepsilon = \nabla.\left[\left(\mu + \frac{\mu_T}{\sigma_\varepsilon}\right)\nabla\varepsilon\right] + C_1\frac{\varepsilon}{k}\mu_T P(\vec{u}) - C_2\rho\frac{\varepsilon^2}{k} \tag{5}$$

where the production of turbulence is defined as

$$P(\vec{u}) = \nabla \vec{u}.\left(\nabla \vec{u} + \nabla \vec{u}^{T}\right) \tag{6}$$

Constants $C_\mu = 0.09$, $C_1 = 1.44$, $C_2 = 1.92$, $\sigma_k = 1.00$, and $\sigma_\varepsilon = 1.30$ assume the standard values proposed by Launder and Spalding (1972).

The inlet boundary conditions were defined for a gasoline mass flow rate of 0.00999 kg/s (ρ = 740 kg/m^3 and μ = 560 × 10^{-6} Pa.s), subsonic flow regime and k-ε turbulent flow. Values of k and ε were evaluated according to Eqs. (8) and (9) (Versteeg and Malalasekera, 1995):

$$k = \frac{3}{2}\left(U_{ref} T_i\right)^2 \tag{7}$$

$$\varepsilon = c_\mu^{3/4} \frac{k^{3/2}}{0{,}07L} \tag{8}$$

where U_{ref} is the average velocity along the inlet section, L is a characteristic length, in this case the inlet tube diameter, and T_i is the turbulence intensity, equal to 0.05. No slip boundary condition was adopted for the walls, and the wall functions proposed by Launder and Spalding (1974) and their variations are implemented in the software and used to perform the simulations. For the outlet, the boundary conditions were defined by subsonic flow regime and ambient pressure (101300 Pa). Figure 1 shows the domain where the flow was simulated.

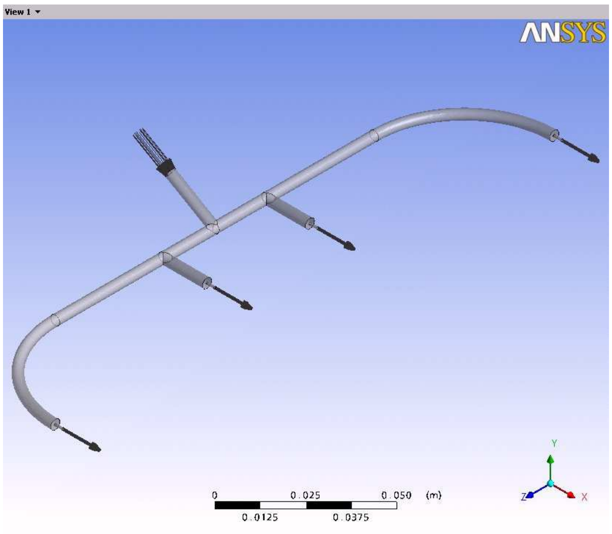

Figure 1 – Isometric view of the injection system device

The device has one inlet tube with a inner diameter of 3.5 mm and four outlet tubes, named outlet 1 to outlet 4, from left to right, considering Fig. 1. At the end of each outlet tube the flow is constrained by a reduction of the tube cross sectional area. Three hole diameters (0.4 mm, 0.6 mm and 0.8 mm) were simulated in order to evaluate the effect of this variable in the system performance.

CFX 11 was used to simulate the flow through the injection system device. Some parameters had to be defined for each simulation. A reference pressure equal to 101300 Pa and a fluid temperature equal to 298 K were used in the domain definition. For each simulation a maximum of 200 time steps was used and a convergence criteria of target residual of 10^{-5} was considered.

In all simulations a mesh with a maximum end of 0.3 mm was used and inflation was used near the wall in order to get a best solution for the flow field and pressure drop. Figure 2 shows the mesh along the surface and a detail of the mesh at the end of one of the outlet tubes.

Ansys Workbench was used as mesh generator. Mesh tests were used to make sure that the numerical flow field was independent of the mesh used during the simulation. The final mesh used 247826 nodes, 758997 elements (tetrahedra and pyramids) with a maximum end length ratio of 15.199.

The numerical results were validated through comparison with the experimental data obtained from the prototype built. Uniform conditions at the outlet are desirable. The computational model was used to predict the most suitable diameters of the outlet tubes, in order to obtain uniform values of the mass flow rate at the outlet.

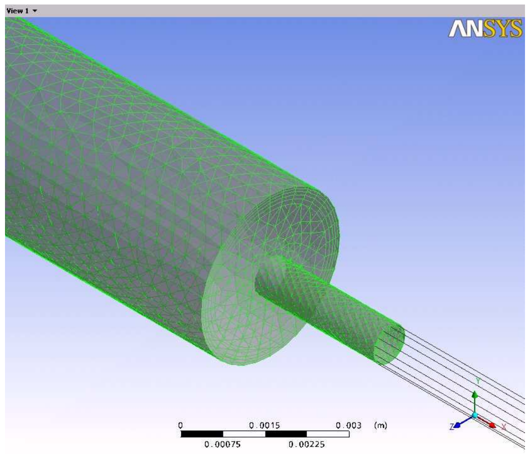

Figure 2 – Mesh along the tube and at the end of the outlet tube

3.2. Test Bench

The cold start system with fuel injector and distributor is basically composed by a gasoline tank, an electrical fuel pump, a fuel injector, a modulator of injector open and closed, a line for gasoline conduction and introduction into the ramifications of the intake manifold through calibrated holes placed in their respective ends. A pressure regulator and a manometer (measurement band up to 4.0 bar, with resolution of 0.1 bar) were installed to provide the researchers with adjustment and monitoring of the system during flow verification tests. The time modulator of injector opening and closing makes possible the adjustment to the quantity of injected gasoline in the intake manifold. Its voltage should be 12 Volts, constituted by a variable oscillator that defines how long the injector is held open and close in a certain unit of time, by a potentiometer that allows the alteration of wavelength and by a frequency divisor. The band frequency in which the modulator works is from 70 Hz to 840 mHz, meaning that the time it is held open and close can be adjusted from 14.3 ms to 1.2 s. There is also the possibility of having a steady state flow at a constant pressure. Figure 3 shows the detailed experimental device and Fig. 4 presents the drawing of a fuel distributor with an electrical injector.

Figure 3. Overview of experimental setup.

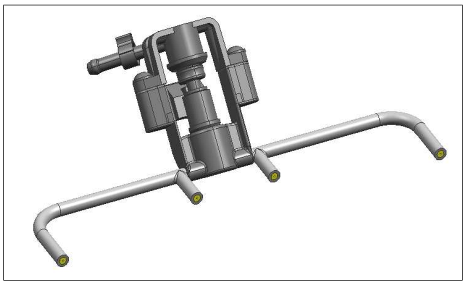

Figure 4. Fuel distributor with electrical injector.

4. RESULTS AND DISCUSSION

4.1. Numerical Results

Figure 5 shows the velocity vector variation along the domain in a plane passing through the centerline of every outlet tube, for orifices with 0.6 mm. Similar results were observed for the other diameters and will not be presented along the paper.

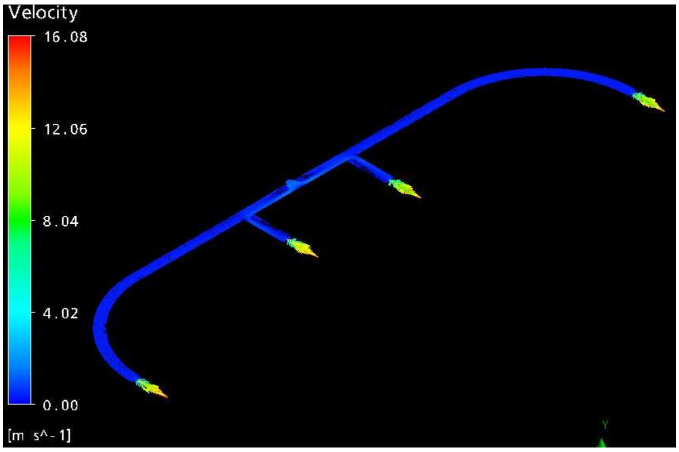

Figure 5 – Velocity variation along the domain.

It is possible to see that the velocity is quite uniform along each pipe and the maximum value happens at the orifice where the area is minimal. The colors indicate a uniform flow distribution at each outlet pipe. Table 2 shows the theoretical and simulated mass flow rate at the outlets.

Table 2. Theoretical and simulated flow rates at the outlets.

Orifice Diameter (mm)	Theoretical (kg/s)	Numerical (kg/s)			
	For each outlet	Outlet 1	Outlet 2	Outlet 3	Outlet 4
0.4	0.002498	0.002491	0.002509	0.002491	0.002498
0.6	0.002498	0.002513	0.002486	0.002483	0.002508
0.8	0.002498	0.002508	0.002493	0.002487	0.002503

The maximum deviation is observed using the orifice with 0.6 mm at outlets 1 and 3, with an absolute deviation of 0.6% with respect to the theoretical mass flow rate. The results show that all configurations tend to present a uniform distribution of gasoline through the orifices. With diameters of 0.6 mm and 0.8 mm, it can be observed that orifices 1 ad 4, at the device ends, have higher mass flow rates. This is due to the flow structure along the curves, which does not present any recirculation area. In the outlet tubes 2 and 3 it is possible to see a vortex at the front end of the outlet tube 2, as shown by Fig. 6.

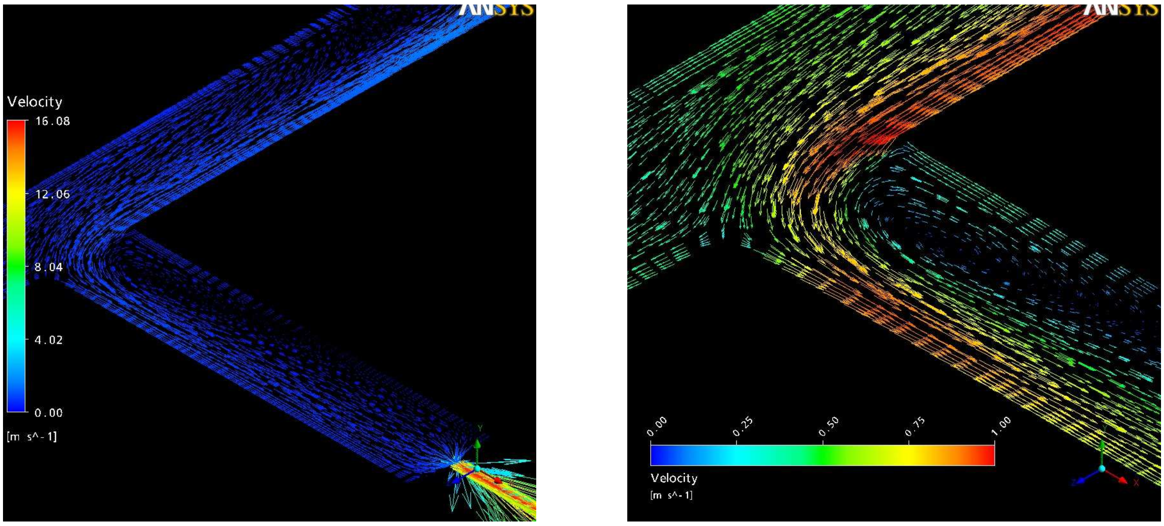

Figure 6 – Vortex at the second tube and a detailed view of the recirculation.

In Fig. 6 the color scale of both images are different. On the left it is shown the general vector field and the velocity variation can be observed only at the tube end. On the right the scale was changed in order to allow a clear identification of the recirculation area. Figure 7 shows the presence of two recirculation areas near the end of the inlet tube.

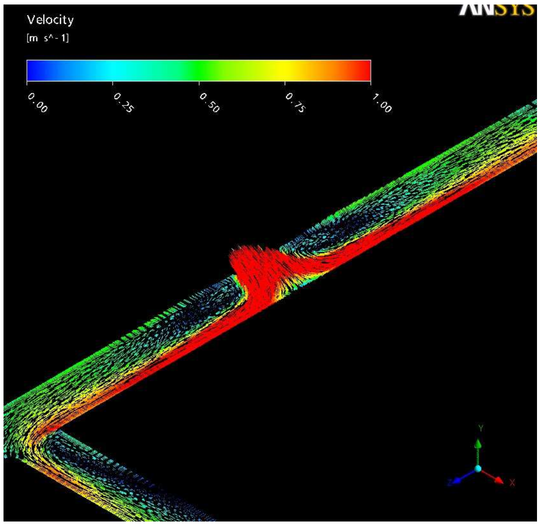

Figure 7 – Recirculation at the inlet tube end extremity

In both cases, the recirculation can be minimized or even eliminated by use of a tube derivation in an angle smaller than 90° or even by a curve. This solution introduces some problems in the manufacturing process. The effect of the recirculation area is lower in the smaller diameter due to the increasing pressure variation along the device. Table 3 shows the inlet pressure as a function of the orifice diameter.

Table 3. Simulated device inlet pressure.

Orifice Diameter (mm)	Pressure (Pa)
0.4	715273
0.6	196603
0.8	130699

As expected, the pressure increases with decreasing orifice diameter. The maximum pressure is higher than the usual value of fuel pumps, of 250 kPa. Thus, the diameter of 0.4 mm cannot be used in a real device. Figure 7 shows the pressure and velocity distribution along the outlet tube 2 centerline.

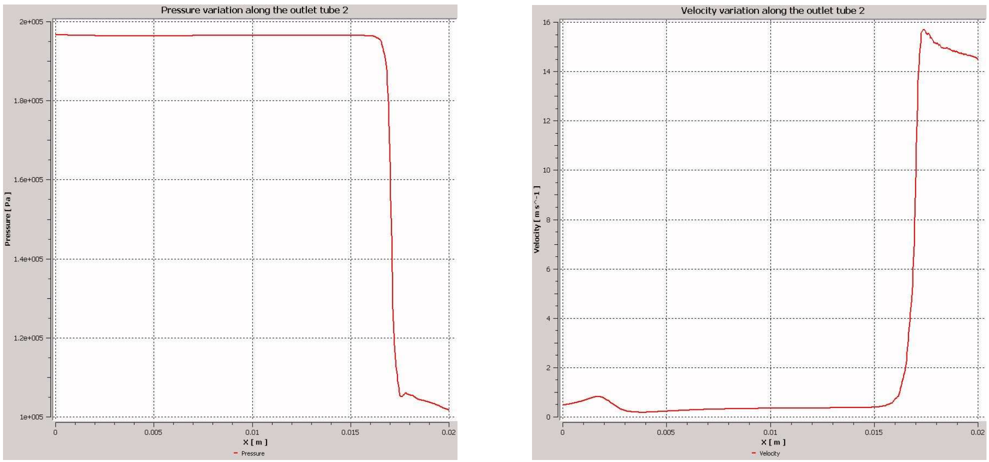

Figure 7 – Pressure (left) and velocity (right) variation along the outlet tube centerline.

The pressure remains constant along the tube and decreases suddenly in the position where the orifice is located. The velocity presents a variation at the front end due to interaction between the main tube flow and the outlet tube flow. Then, the velocity remains constant along the tube, increasing in the region where the flow is constrained by the tube cross-sectional area reduction, as shown by Fig. 8 (left) in outlet tube 1.

Figure 9 shows the eddy viscosity distribution along the domain. The distribution is symmetric with respect to the inlet tube and the highest values are observed in the regions of flow recirculation.

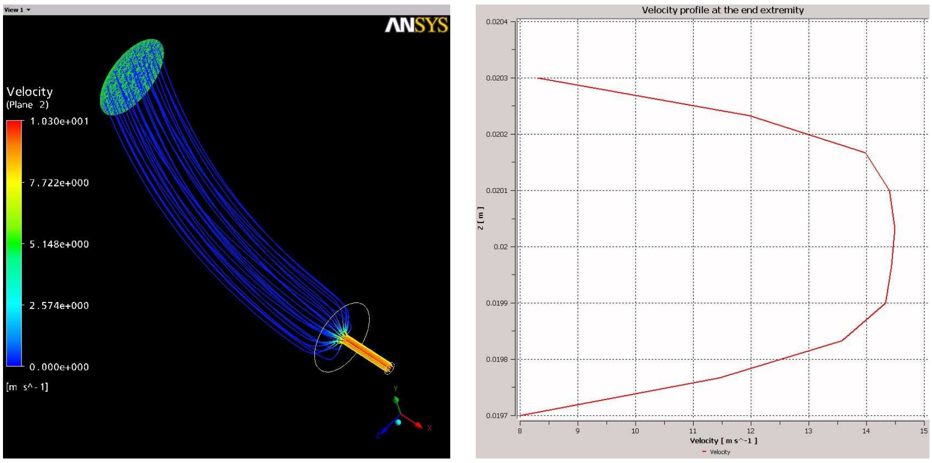

Figure 8 – Streamlines in the outlet tube 1 (left) and velocity profile (right) at the outlet tube back end.

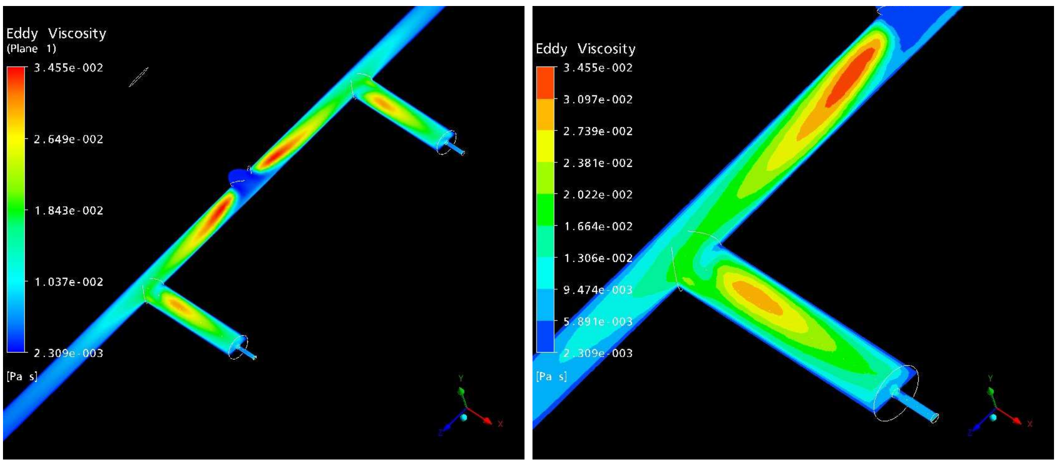

Figure 9 – Eddy viscosity variation along the domain

The turbulent eddy dissipation and turbulent kinetic energy distribution near the outlet tube back end are shown in Fig. 10. Both are symmetrical with respect to the centerline and have their maximum values near a region where the flows present a sudden direction variation due to the area reduction.

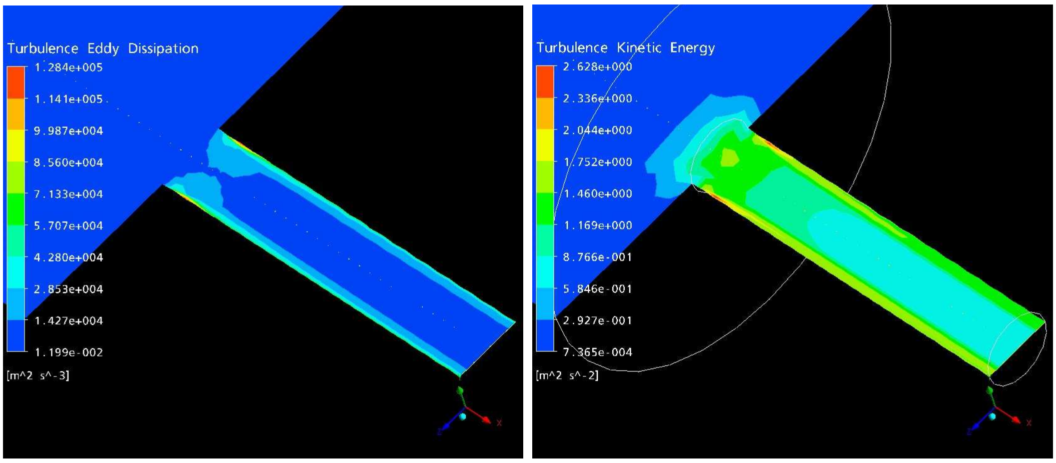

Figure 10 – Turbulent eddy dissipation and turbulence kinetic energy along the outlet tube back end.

The numerical results were validated through comparison with the experimental data obtained using the test bench previously described.

4.2. Experimental Results – Gasoline flow measurement

In order to validate the numerical model, gasoline flow measurements were performed at steady state condition, that is, with the electrical injector open along the measurements. The four calibrated holes were of 0.6 mm diameter and the measurements of the injected volume in each extremity were done every 10 seconds (measured with a resolution of 0.01 s) under a pressure of 2.15 bar. The volume samples were collected in a 50 ml test tube, with resolution of 1.0 ml. Five measurements were done for each end and the results were obtained for the average sample volume.

Table 4 presents the results for the five gasoline flow measurements. In central positions (orifices 2 and 3) it was noticed a flow 3% higher than that in the orifice ends (orifices 1 and 4). The difference can be attributed to small deviations due to the manufacturing process. Table 5 shows the percent difference between numerical and experimental results for the mass flow rate.

Table 4. Results from steady state flow measurement [(1)].

Samples	Hole 1		Hole 2		Hole 3		Hole 4	
	(ml/s)	(kg/s)	(ml/s)	(kg/s)	(ml/s)	(kg/s)	(ml/s)	(kg/s)
1	3.40	0.002516	3.50	0.002590	3.50	0.002590	3.30	0.002442
2	3.20	0.002368	3.50	0.002590	3.20	0.002368	3.20	0.002368
3	3.40	0.002516	3.50	0.002590	3.60	0.002664	3.50	0.002590
4	3.30	0.002442	3.30	0.002442	3.50	0.002590	3.30	0.002442
5	3.40	0.002516	3.40	0.002516	3.30	0.002442	3.40	0.002516
Average	**3.34**	**0.002472**	**3.44**	**0.002546**	**3.42**	**0.002531**	**3.34**	**0.002472**

[(1)]: measured at 25°C

Table 5 – Percent difference between experimental and numerical mass flow rates

Outlet 1	Outlet 2	Outlet 3	Outlet 4
1.66	2.36	1.90	1.46

The maximum deviation observed is 2.36%, which represents a good agreement between numerical and experimental results. The simulated pressure at the inlet was 199603 Pa lower than the measured pressure of 215000. An acceptable percent difference of 7.16%, which can be explained considering that the fuel transport properties were estimated and not measured.

5. CONCLUSION

A concept of a cold start system equipped with an electronic fuel injector was analyzed numerically and experimentally. Numerical results show that the proposed concept led to a uniform fuel distribution to the engine cylinders. The geometry design can be improved in order to avoid recirculation regions in the flow. The pump characteristics used in the car fuel system limits the pressure at the inlet. Measurements show that orifices of 0.6 and 0.8 mm can be used in a real system used in a car. In this work a cold start system using orifices with 0.6 mm was tested and the results compared with numerical results. Numerical and experimental results present a good agreement. The choice of a 0.6 mm instead of 0.8 mm is justified by the necessity to create a spray in the region placed after the start system, which requires the maximum pressure available.

6. ACKNOWLEDGEMENTS

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7. REFERENCES

Annand W.V.D. and Gulder O.L. , 1980, "Exhaust Emissions and Cold Starting of a Four-Cylinder Engine Using Methanol as Fuel", Proc. Instn. Mehc. Engrs., Vol. 194, pág. 139-143.

Deschamps, C.J., 1998. "Modelos Algébricos e Diferenciais", Proceedings of the 1st Spring School of Transition and Turbulence, Vol. 1, Rio de Janeiro, Brazil, pp. 99-155.

Haatela O. and Decker G.,1989, "Present and Future Fuels and Lubricants in Cold Climate Operation", SAE 890032, SAE Technical Paper Series.

Ilinca, F., Pelletier, D. and Garon, A., 1997. An adaptive finite element method for a two-equation turbulence model in wall-bounded flows. International Journal for Numerical Methods in Fluids, vol. 24, pp. 101–120.

Iwai N., 1981, "A Study on Cold Startability and Mixture Formation of High-Percentage Methanol Blends", SAE 880044, SAE Technical Paper Series. G.

Kito B.T., Pacas A.D., Selim S. and Cowley S.W., 1998, "Properties of an Ethanol-Diethyl Ether-Water Fuel Mixture for Cold-Start Assistance of an Ethanol-Fuelled Vehicle", American Chemical Society, Colorado, USA.

Launder, B.E. and Spalding, D.B., 1974, "The numerical computation of turbulent flows", Computer Methods in Appplied Mechanics and Enginnering, v.3, p.269-289

Sales, L.C.M. and Sodré J.R., 2002, "Optimized Cold Start Auxiliary System for Emissions Reduction in Ethanol-Fuelled Engines", In: SAE, Inc. (Org.). SAE – Alternative Fuels. 2002, v. 1725, p. 1-6.

Sales, L.C.M. and Sodré J.R., 2003, "Reducing Exhaust Emissions in Ethanol-Engines During Cold Start. Combustion Science and Technology, USA, v. v, n. 8, p. 1535-1550.

Silva N.R. and Sodré J. R., 2000, "Using Additive to Improve Cold Start in Ethanol-Fuelled Vehicles", SAE 2000-01-1227, SI Combustion, SP-1517, SAE, Inc., pág. 203-207.

Versteeg, H. K., Malalasekera, W., 1995, An Introduction to Computational Fluid Dynamics The Finite Volume Method, Longman Scientific & Technical, Malaysia.

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