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LARGE HSDI CR DIESEL ENGINES MULTIPLE INJECTIONS AND MULTIPLE SWIRLS CONCEPT

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ABSTRACT

All future power developments should consider as primary tasks the achievement of the required emission levels and CO2-values, while still providing optimum torque-to-rpm curves, the lowest SFC (Specific Fuel Consumption) over the widest range possible, good power-to-weight and affordable costs. One method to achieve these objectives is the downsizing. To achieve the levels of engine performance that are required, a significant increase in the rated speed and in the boost pressure is mandatory. In this case, the result is an increase in the flow rate through the intake and exhaust ports and valves. Considering the impact of these changes, the port layout of the system is reanalyzed. Another physical limit to the maximum speed depends on the CR (Common Rail) injector dynamic performances. These performances decrease with size for inertia problems and they depend on the amount of effort involved in their development. Automotive engines in the range of 10 to 100 HP per cylinder are the most common. For this reason, these injectors are the most advanced and cost-effective. Furthermore, their small size and inertia is favorable to the best dynamic performance. The larger number of nozzles improves combustion performance. In fact the better surface to volume ratio or the spray improves heat transfer. For this reason, multiple injection systems can be used in the modern HSDI (High Speed Direct Injection) CR large diesels. This solution was commonplace before WWII, but has been progressively abandoned with the introduction of mechanical high-pressure injection systems and the bowl combustion chamber. In this paper, a dual combustion chamber per cylinder engine is considered. The primary purpose of this study is to examine the best port layout on a modern diesel combustion system and to introduce a new promising concept. The study included flow measurements of intake flow and CFD simulations of the flow field during intake. This design enables the formation of two homogenous swirls centered onto the injectors, with excellent flow coefficient. The design also allows an increase in volumetric efficiency combined with a reduction in flow losses.

Keywords: diesel engines, multiple injections, multiple swirls.

INTRODUCTION

The prerequisites for advanced concepts of High Speed Direct Injection (HSDI) Common Rail (CR) Diesel Engines require that they must improve fuel economy and the heat-trapping ability of carbon dioxide emissions. One traditional method of reducing fuel consumption is through downsizing the displacement of the engine. However, to maintain the same level of performance, the specific power and torque of the engine has to be increased. In addition, the specific load during the test cycle is also increased.

Even with the most recent technologies exhaust gas after treatment technology, a further reduction in the raw emissions of the engine is required. To achieve this target in combination with downsizing, the complete gas exchange system has to be optimized to meet the new requirements. The layout of the intake ports is one important issue, because an increase in specific power will also increase the mass flow rate through the intake and exhaust ports. Reducing the gas exchange work can be accomplished through improving the efficiency of the turbocharger and also through improvements to the intake and exhaust ports that will reduce flow losses and increases the volumetric efficiency of the engine. Furthermore, the charge motion has a significant influence on soot formation, soot oxidation and combustion velocity. It is widely accepted that the swirl number is not a sufficient index, being an average value. In fact, the uniformity of swirl greatly influences the combustion process and soot oxidation as well. The following study analyzes a new concept for intake and exhaust port design that is designed to meet these future challenges.

CONCEPT AND LAYOUT OF INTAKE PORTS

Definition of objectives and limitations

Advanced diesel engine combustion systems require the highest and coldest possible filling in order to support combustion through charge motion, which is realized through swirl motion in advanced HSDI CR diesel engines. The traditional goal when designing the intake and exhaust port geometry is to achieve the lowest flow losses possible and ensure a defined and stable charge motion. However, the relationship between the resulting swirl angular velocity and the relative velocity between injection spray and air is approximately proportional to the mean piston speed when no additional measures are taken. In the automotive field an adequate angular velocity of the

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filling is required for low engine speeds. For this reason the inlet ports layout concept should offer the possibility to increase the swirl at low speeds.

The port design should then achieve the maximum flow in the intake and exhaust ports and an adequate charge motion even at variable loads. On the one hand, the port design is limited due to design constraints, such as the height of the cylinder head, the room required for injectors and the cooling system, while achieving adequate robustness of the cylinder head. The concept introduced in the following is optimized to meet these demands and constraints. The intake ports of Direct Injection (Diesel) diesel engines can be approximately subdivided into three groups: Tangential ports, helical ports, and filling ports. Tangential ports are designed to run the air tangentially at a relatively flat angle into the combustion chamber and thus generate a charge motion around the vertical axis of the cylinder, where usually the injector is positioned. Tangential ports generate high swirl with moderate flow. Helical ports are more common and their geometry lets a part of the main air flow rotate around the valve stem to force the desired swirl on the majority of the gas mass drawn into the combustion chamber. For a two-intake-valves arrangement, it makes sense to combine a helical intake port with a filling port. This port has the primary goal of representing the lowest flow resistance possible. For this purpose, the port is as straight as possible from the top onto the valve.

With parallel valve arrangement, it makes sense to design one port as a helical port and the second one as a filing port. Both intake valves may be fitted with seat swirl chamfers that adjust the integral swirl over the intake event to the desired level. The layout of the exhaust ports minimizes the flow resistance, leading to a reduction in gas exchange work and of residual gas. In order to facilitate a swirl increase at low speeds with this configuration, the "state-of-the-art" is to close the filling port with a swirl flap. These flaps are expensive and proved to be unreliable, especially when their movement is controlled continuously as the engine speed varies. The use of seat swirl chamfers on both intake valves ensures a very high charge motion at low valve lifts. This is the true problem, in fact at low rpm and high load the relatively low combustion velocity limits the fuel charge. Combustion speed depends on temperature Wolfer (1938) [1] (1) and swirl ratio (2).

$$\tau_{ign} = 0.44 P_0^{-1.19} \exp\left[\frac{4650}{T_0}\right] \tag{1}$$

$$Swirl_{ratio} = \frac{c_t}{c_a} \tag{2}$$

In fact the auto ignition point (Point of Start of Combustion PSOC) the energy release due to combustion exothermic reactions begins to exceed the energy losses due to the fuel evaporation. This first stage of combustion is spectrally characterized by a weak flame emission, mainly in the near ultraviolet range, attributed to natural chemiluminescence. Chemiluminescence is linked to the emission of OH, CH and HCO, due to exothermic chemical reactions occurring during thermal decomposition of the hydrocarbon molecules preceding true ignition. This is the concept of "flameless" or low temperature ignition at low thermal loads, which is the problem in diesel-ignited dual fuel (GAS+Diesel) engines. OH distribution at selected crank angle for the FIAT 1900 jtd is shown in Figure-1 [2].

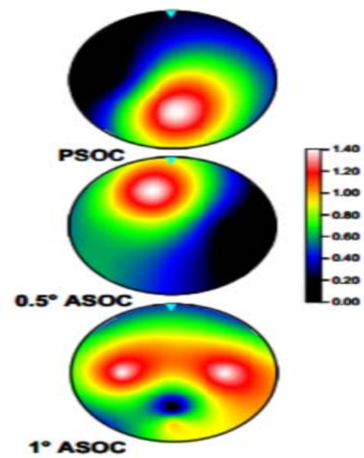

Figure-1. OH emissions PSOC for FIAT 1900 jtd at 1000rpm and 57bar peak combustion pressure [2].

Figure-2 shows that the combustion begins asymmetrically. The jet proceeds straight and radially inside the combustion chamber, slowing down toward the "cool" cylinder/head/piston wall. Given the high swirl ratio the combustion is not symmetrical along the jet axis. The jet burns more quickly in the surface that is more exposed to oxygen reach air. The burning progress depletes the swirl energy reducing the combustion efficiency and speed. However, the combustion speed still increases due to the temperature and pressure increase during the combustion phase. It is then fundamental to have an extremely strong swirl at any rpm (mean piston speed) and load level.

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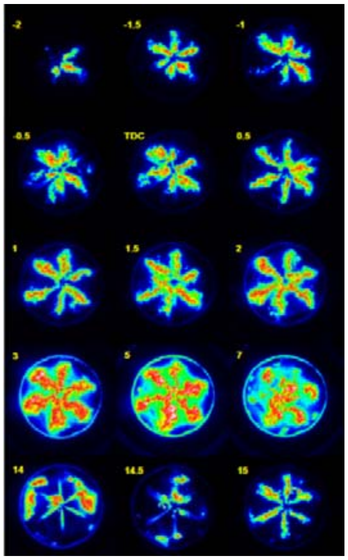

Figure-2. UV flame emission sequence [2].

A reduction of the valve lift has a negative effect on the flow coefficient. However, it is convenient is comparison to the intake port shutoff. In fact, closing the filling port reduces the flow coefficient by more than half. On the contrary, a reduction of the valve lift by half leads to a reduction of the flow coefficient by less than 50%. However, in the automotive field, the swirl increase is required at low speeds, and this is where the gas losses due to the exchange cycle are less sensible to a low flow rate. Therefore, adopting the variable lifting device is not as convenient as in spark ignition engines. To obtain high flow performances of the ports, the bore of the engine is extremely important. The bore defines the room available for valves, injector(s) and glow plug. For this reason, in large engines, it is convenient to use an external heater instead of the glow plugs. However, heaters are less efficient, more expensive and noisier than the glow plugs. In any case, the smaller is the size of the injector(s) the better. The possibility of multiple smaller and more efficient injectors increases the size of the ports and the flow characteristics. Figure-3 shows the results of the flow measurements performed on a stationary flow test rig of a typical 4 valve head with a helical and a straight input port without seat swirl chamfer. The unitary displacement is 500cc. The flow coefficient is the quotient of the isentropic opening area related to the piston area.

There are many methods of studying the in-cylinder air movement, but they can be classified in two groups: experimental measurements and CAD simulation. Experimental research reproduces approximately the "real" operating conditions of the engine and usually it has high times and costs. In the past, experimental validation was always necessary in the development of an engine. Amongst the mostly used experimental methods for in-cylinder flow study are the method of steady flow test rig, which outputs the Swirl and Tumble ratios which characterize the in-cylinder flow, and LDA (Laser-Doppler Anemometry), which can measure the velocity in some limited measuring points in the volume.

In the last few years the Computational Fluid Dynamics (CFD) programs have evolved. They became more robust, easy to use and extremely fast even on standard desktop computer. This paper demonstrates that the results obtained by CFD calculations agree well with experimental data. The cost of studying in-cylinder flow with CFD is only at a fraction of that of experimental research. CFD simulation outputs more information about the flow field such as: velocity fields, pressure, turbulent kinetic energy, etc.

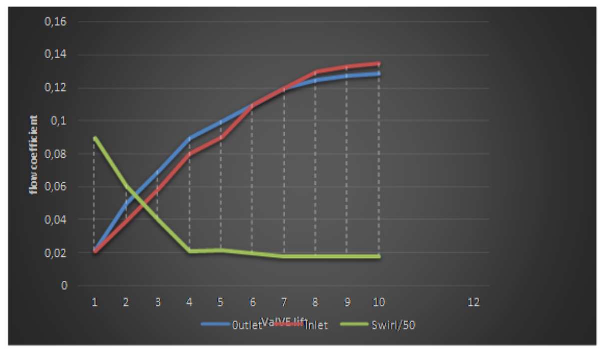

Figure-3. Swirl and flow coefficient vs. valve lift.

The experimental apparatus

The flow-bench experiments are performed to quantify the in-cylinder tumble motion and engine volumetric efficiency by using the same cylinder head and valve. The experimental set up is shown schematically in Figure-5. Special fixtures are used to adjust valves lift with the required precision (1/100 mm). In standard tests on engines, inlet or outlet valves are open simultaneously. Test is performed on a true cylinder head (or on a true scale mock-up usually manufactured with Rapid Prototyping) or flow box with the same ports geometry. A dummy cylinder with a diameter equal to engine bore closes the measured volume. The motor is used in suction mode to study intake parts e.g. intake manifold(s) and intake port(s). In this mode, air is sucked from ambient to simulate the intake stroke of the engine. The motor can be also used in blowing mode which is used to study outlet parts e.g. exhaust manifold and catalyst convertor. In this mode, air is blown to ambient to simulate the exhaust stroke of the engine. A swirl meter measures tumble and/or swirl intensity, based on its orientation. The standard test for inlet port consists of measuring the flow coefficient, swirl and tumble intensities for different valve lift. In this test, after applying the desired differential pressure, the valves lift is set at certain level. In a few test rigs it is also possible to measure velocity vectors and pressures in many points through miniaturized Pitot tubes and pressure gauges.

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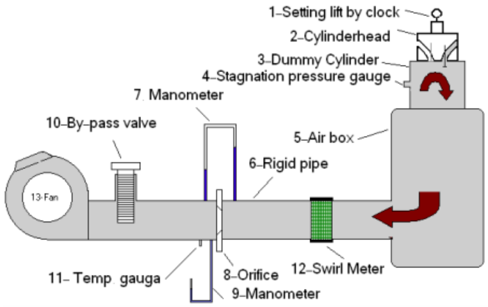

Figure-4. Traditional test rig set-up [3].

Traditional CFD simulation

The boundary conditions used in the traditional CFD simulation are the same as in experimental measurements. It is necessary to estimate the turbulence intensity and the turbulent length scale L. The turbulent length scale may be considered equal to L=0.1 D, where D is a characteristic dimension. In this case the equivalent diameter of the inlet section of the pipes is ≈20mm, so L=0.002.

Our HSDI CR diesel engine has a bore of 108 mm and a stroke. The two intake valve lifts selected are: 4, and 13 mm. Among the studied parameters are velocities, pressure, turbulent kinetic energy, dissipation of turbulent kinetic energy, flow coefficient and swirl ratio. The last two parameters will be compared with experimental measurements may be compared with the traditional test rig (Figure-3).

The turbulence intensity *I* in a duct is the ratio between the mean velocity deviation, *u'* and the flow mean velocity u_{mean}. Input and output boundary conditions are different since they depend on Re. In input we have equations (3) (4) (5).

$$I = \frac{u'}{u_{mean}} \approx 0.16\,\mathrm{Re}^{\frac{1}{8}} \tag{3}$$

$$u_{mean} = \frac{Q}{A_{mean}} \approx 50 \tag{4}$$

$$\mathrm{Re} = \frac{u_{mean}\rho D}{\mu} \approx 150000 \Rightarrow I \approx 0.04 \tag{5}$$

However in modern CFD software these additional boundary conditions are not needed.

The CFD software

New Flow Simulation Tools are easy-to-use CFD programs designed to facilitate the work of product engineers who use 3D CAD. To make CFD usable for mechanical designers and design engineers from other engineering disciplines, this software is largely automated to minimize the specialist expertise required to operate traditional CFD software. Modern CFDs simulate complex turbulent flows with heat and mass transfer. The adaptive turbulence capability against some classic industrial CFD validation cases is the most important feature.

The average coefficients L and I are automatically calculated and optimized through the simulation domain. An immersed boundary Cartesian meshing techniques allow for fairly accurate flow field resolution with low cell mesh densities [4]. At first, the pre-processor detects the computational cells that are cut by the body. The cells are then divided in those that are inside and outside of the body.

Figure-5. The new inlet ports.

Lam and Bremhorst's damping functions fμ, f_1, f_2 decrease turbulent viscosity and turbulence energy and increase the turbulence dissipation rate. These parameters are controlled by the Reynolds number R_y. R_y is calculated on the average velocity of fluctuations and distance from the wall. When $f_\mu = 1$, $f_1 = 1$, $f_2 = 1$ the approach obtains the original k-ε model.

Figure-6. The 4 ports assembly.

A fluid's boundary layer requires a very high-density mesh to give accurate results. This approach impairs the computational efficiency of the simulation and usually requires long computer time on extremely performing computing systems. For this reason, the

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computational mesh used in our CFD code is always an immersive boundary non-body-fitted Cartesian mesh.

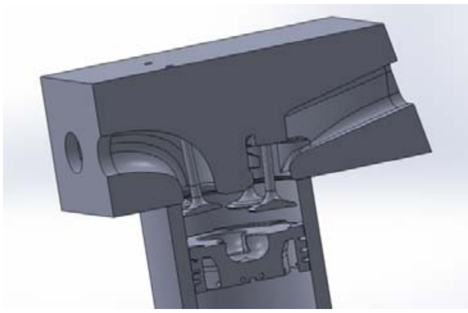

Figure-7. The CAD model for the CFD simulation.

With this approach the CFD explicitly obtains the momentum, heat flux, and turbulent boundary conditions for the RANS. This approach simplifies the mesh and reduces the computational time even in large domains. It is then possible to obtain fairly accurate results with standard desktop computers. A very important aspect is the computer resources necessary to obtain reliable results. In this case a normal desktop computer was used.

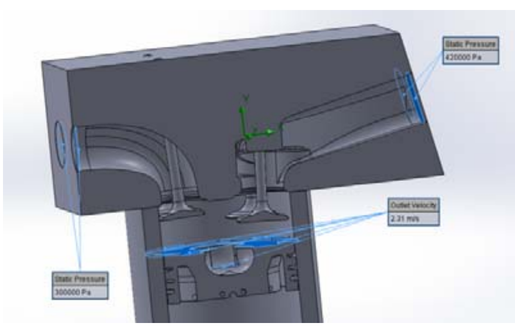

Figure-8. "True" boundary conditions.

The inlet port geometry

The present paper introduces an original geometry of inlet ports for a HSDI CR diesel engine with two intake valves and two injectors per cylinder (Figure. Specifically, the new geometric conformation of inlet ports is capable of generating two extremely strong motions of counter-swirl during the intake stroke. Since the flow analysis should be the more realistic possible, the true boundary conditions that come from the 1D-engine-gas-dynamics simulation software simulation are introduced (Figure-7).

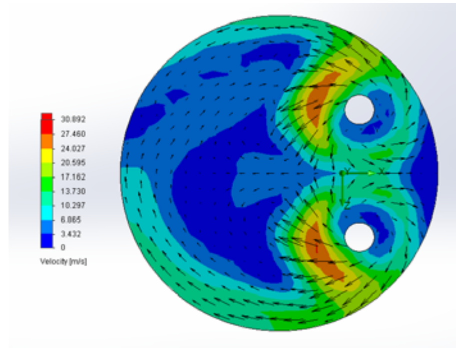

Figure-9. Velocity vectors at 1 mm from head top, valve lift 7mm speed 3000rpm.

The inlet port geometry

The simulation results are illustrated by velocity vector diagrams, as displayed in Figure-10. Figure-10 reveals that there are two strong counter rotating wake jets inside the cylinder. The two vortexes are almost symmetrical.

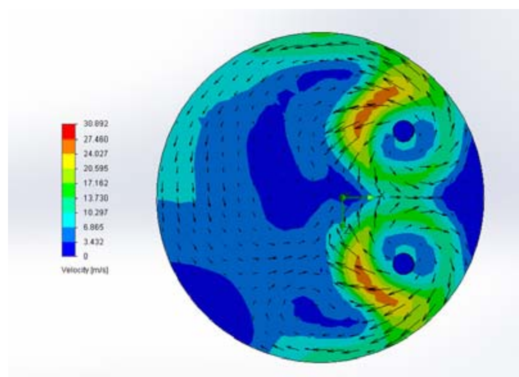

Figure-10. Velocity vectors at 1 mm from head top, valve lift 13 mm speed 3000rpm.

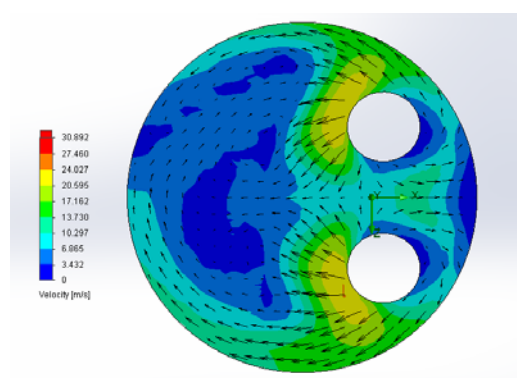

Figure-11. Velocity vectors at 20 mm from head top, valve lift 7mm speed 3000rpm.

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The two vortexes have approximately the same velocity pattern. This effect reduces the friction and the damping of the vortex. This arrangement has a very good flow coefficient even at relatively high rpm (3000 rpm). The good swirl ratio makes it possible to keep the valve lift extremely high (13 mm) (see Figure-11). The velocity pattern is kept along the cylinder axys (Figure-12).

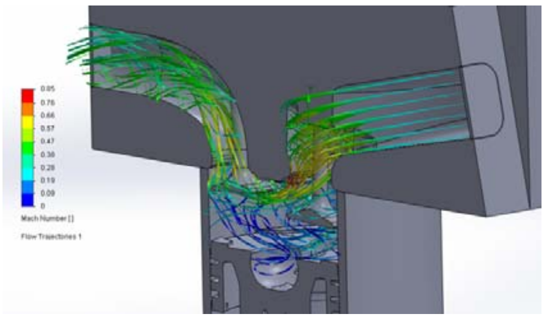

Figure-12. Trajectories at valve lift 7mm speed 3000rpm.

SIMULATION RESULTS AND COMPARISON

A comparison of the flow coefficients predicted with the CFD model to the flow coefficients measured on a flow bench is shown in Figure-13.

The boundary conditions were modified to simulate test rig instead of the engine (Table-1).

Table-1. Boundary condition for test rig simulation.

Boundary condition	**Pressure** [Pa]	**Temperature** [K]
Inlet	102125	296
Outlet	98205	296

The predicted and measured flow coefficients trends are quite similar which provides more confidence that the CFD model is able to replicate the main flow trends within the intake system.

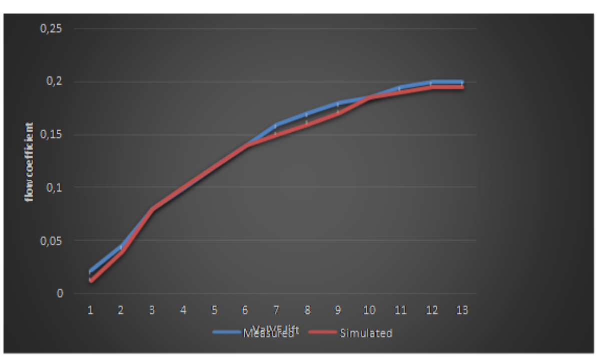

Figure-13. Flow coefficient comparison.

A benefit of using CFD to analyze the intake flow is that it is possible to examine the detailed behavior of the intake flow at any location within the port, head, or cylinder.

It is also possible to modify the geometry in very short time. In the future it will be possible also to automatically optimize the port geometries for optimum performances.

CONCLUSIONS

The primary purpose of this study is to examine the best port layout on a modern diesel combustion system and to introduce a new promising concept. The study included flow measurements of intake flow and CFD simulations of the flow field during intake. This design enables the formation of two homogenous swirls centered onto the injectors, with excellent flow coefficient. The design also allows an increase in volumetric efficiency combined with a reduction in flow losses.

Symbols

Symbol	**Description**	**Unit**
τ_{ign}	Ignition delay	ms
P_0	Combustion chamber pressure	MPa
T_0	Combustion chamber temperature	K
c_l	Tangential velocity	m/s
c_a	Axial velocity	m/s
u'	Mean velocity deviation	m/s
u_{mean}	Mean velocity	m/s
Q	Volumetric flow	m^3/s
A_{mean}	Mean port area	m^2
ρ	Air density	m^3/s
μ	Fluid viscosity	Pa s

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