# STUDIES ON VARIABLE SWIRL INTAKE SYSTEM FOR DI DIESEL ENGINE USING COMPUTATIONAL FLUID DYNAMICS

by

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*It is known that a helical port is more effective than a tangential port to attain the required swirl ratio with minimum sacrifice in the volumetric efficiency. The swirl port is designed for lesser swirl ratio to reduce emissions at higher speeds. But this condition increases the air fuel mixing time and particulate smoke emissions at lower speeds. Optimum swirl ratio is necessary according to the engine operating condition for optimum combustion and emission reduction. Hence the engine needs variable swirl to enhance the combustion in the cylinder according to its operating conditions, for example at partial load or low speed condition it requires stronger swirl, while the air quantity is more important than the swirl under very high speed or full load and maximum torque conditions. The swirl and charging quantity can easily trade off and can be controlled by the opening of the valve. Hence in this study the steady flow rig experiment is used to evaluate the swirl of a helical intake port design for different operating conditions. The variable swirl plate set up of the W06DTIE2 engine is used to experimentally study the swirl variation for different openings of the valve. The sliding of the swirl plate results in the variation of the area of inlet port entry. Therefore in this study a swirl optimized combustion system varying according to the operating conditions by a variable swirl plate mechanism is studied experimentally and compared with the computational fluid dynamics predictions. In this study the fluent computational fluid dynamics code has been used to evaluate the flow in the port-cylinder system of a DI diesel engine in a steady flow rig. The computational grid is generated directly from 3-D CAD data and in cylinder flow simulations, with inflow boundary conditions from experimental measurements, are made using the fluent computational fluid dynamics code. The results are in very good agreement with experimental results.*

Key words: *swirl, turbulence, swirl ratio, tangential velocity, turbulence intensity*

## Introduction

The swirl port is designed for lesser swirl ratio to reduce $NO_x$ emissions at higher speeds. But this condition increases the air fuel mixing time and particulate and smoke emissions in lower speeds. The combustion system of an internal combustion engine is usually designed for a certain working condition, if the engine load or speed changes, then the designed parameters will not match. For example, the combustion system of a diesel engine is designed basically on full load condition, when it is run under partial load condition, the engine's fuel economy will be poorer. A variable swirl system can change the situation and meet the combustion needs of swirl in broader operating conditions [1]. If a variable swirl system is applied, when the engine works under full load condition, the inlet swirl will be decreased so that the charging efficiency is increased, and under the partial load condition, the inlet swirl will be enhanced at a little expense of charge. Hence a variable swirl system can improve engine performance.

## Need for variable swirl system

Engine needs variable swirl to enhance the combustion in the cylinder according to its working conditions, at partial load or low speed condition requires stronger swirl, and the air quantity is more important than swirl under very high speed, full load and maximum torque conditions. The swirl and charging quantity can be easily trade-off and controlled by the opening angle of the valve. If the engine is designed for high incoming swirl at the inlet of the port, it results in even further increase in swirl at higher speeds which has few adverse effects like:

(1) at higher speeds, increase in swirl leads to increase in $NO_x$ emissions because higher swirl produces more homogeneous mixture which leads to premixed combustion producing high temperatures, and

(2) increase in swirl will result in an increase in heat transfer and there by loss in the thermal efficiency of the engine.

But at the lower speeds, increased swirl is needed for better combustion. Hence the need at the inlet port is high swirl at lower speed range and low swirl at higher speed range for this reason variable swirl mechanism is needed [2]. Variable swirl intake port is a deflector mechanism at the inlet. This mechanism causes higher swirl at lower speeds and lower swirl at higher speeds. This would lead to optimum mixing of diesel and air at those speeds and hence better combustion so that the engine produces less smoke at lower speeds.

## Principle of swirl plate control at various speeds

Adverse increase in swirl at higher speeds must be reduced, and at the same time there is a need for higher swirl at lower speeds to reduce smoke by better mixing of air and fuel, which leads to more complete combustion of the mixture. In order to achieve both these requirements, one of the cost effective method, is to design the intake port for low incoming swirl and increase the amount of swirl at lower speeds by partially closing the intake port *i. e.* by blocking the intake air coming into the port in such a way that the air has to travel a longer path of the larger radius ($r$), mainly about the cylinder central axis. As a result, the angular velocity ($\omega$) also increases. However, there is a slight decrease in mass flow rate due to throttling action of intake port closure. The decrease in mass flow rate is less than the increase in angular velocity. Hence the angular momentum increases as the port opening is decreased this leads to an increase in swirl energy of the incoming air. Higher helical component of swirl results in higher kinetic energy of the fuel air mixture even at lower speeds, which leads to better combustion, less smoke, higher torque and a decrease in specific fuel consumption (SFC). The adverse increase in swirl with increase temperature at higher speeds can be reduced by controlling the swirl plate at 100% open condition [3].

## Variable swirl intake mechanism in two valve ports

The same principle is used in the variable swirl intake port to control the swirl according to speed range. If the area of intake helical port is varied, it can increase the swirl at lower speeds without any design changes in the engine intake system. Air flow during the suction follows a particular path determined by the blockage at the entry. If the obstruction at the entry ensures that the flow is along the larger radii of the engine cylinder. The swirl will be enhanced and this aids combustion at lower speed ranges. For high speed ranges the obstruction can be changed such that a lower swirl is obtained at the intake.

### *Swirl plate arrangement*

The variable swirl plate set up for the W06DTIE2 engine consists of a stainless steel plate as shown in fig. 1. The plate has pockets of same size and shape as the entry of the cylinder head and made to slide in slots in the intake manifold, butting against the cylinder head. The plate was moved by a shaft attached to its end and passing through the side of the manifold, sealed by O ring. The sliding of the plate results in variation of the area of inlet port.


**Figure 1. Experimental swirl plate set up for 2-valve port variable swirl mechanism (color image see on our web site)**

For the variable opening swirl variation case, the variable swirl plate set up readings is compared with computational fluid dynamisc (CFD) predictions in this study.

## Essential steps in CFD analysis

The complete computational methodology applied in this study includes four-step hierarchy. The four tasks are: (1) geometry and grid generation, (2) boundary conditions, (3) solver and discretization scheme, and (4) turbulence model and wall treatment [6].

### *Geometry and grid generation*

The computational model involved in this study is shown in fig 2. This is the port model of AL 6.65 L Euro II engine. This simulation model in the present study includes intake ports; intake valves and cylinder. The model replicates the situation in actual internal combustion engine tests.

The gird generation was done by importing the geometry into the GAMBIT and the T Grid pre-processors from Fluent. The GAMBIT was used to generate a surface mesh, and then the T Grid was used to generate a

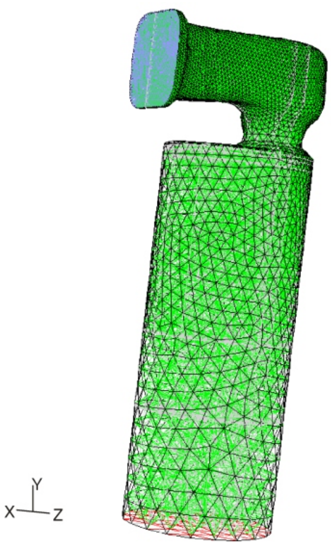



**Figure 2. Model with tetrahedral grid**

volume mesh with interval size 1, which consisted of tetrahedral elements throughout the rest of the flow domain. For complex geometries like helical ports quad/hex meshes have no numerical advantages. So tetrahedral meshing is used for this study.

### *Turbulence model and wall treatment*

#### *Turbulence model*

The turbulence model used for this study is Standard $k$-$\varepsilon$ model. The model is very suitable for initial iterations, initial screening of alternative designs and parametric studies. This model is valid for fully turbulent flows only. Since the flow inside the engine cylinder is fully turbulent this model is one of the preferred model for in cylinder flow analysis. The simplest "complete models" of turbulence are two-equation models in which the solution of two separate transport equations allows the turbulent velocity and length scales to independently determined. The standard $k$-$\varepsilon$ model in fluent package falls within this class of turbulence model.

#### *Near wall treatment*

A strong variation of the flow and turbulence properties occurs near the solid boundary. This necessitates modification of the turbulence model and use of very fine mesh. To avoid using fine grids near the solid boundary an special "wall function" is employed. These wall-functions are derived from a one-dimensional steady coquette flow analysis of the wall region and define the flow conditions across the wall-layer as functions of local Reynolds number $y^+$. Most $k$-$\varepsilon$ turbulence model will not predict the correct near wall behavior. So separate near wall function should be used. Near wall functions are a set of laws that serve as boundary conditions for momentum, energy and species as well as for turbulence quantities. The standard and non-equilibrium wall functions are preferred for high Reynolds number flows. Standard wall functions are used for this study because of its better convergence properties than other.

### **Boundary conditions**

Mean flow velocity at the inlet at zero lift, $U_{ref}$ = mass flow rate /(density  area)
Reynolds number, Re = $\rho U_{ref} D/\mu$
Turbulent intensity, $U'/U_{avg} = 0.16\ \mathrm{Re}^{-1/8}$
Turbulent length scale, $l = 0.07$ L
Turbulence kinetic energy $k = 3/2\ (U_{ref} T_i)^2$
Dissipation rate, $\varepsilon = C_\mu^{\ 3/4}\ k^{3/2}/l$
Turbulent viscosity ratio, TVR = $\mu_t/\mu$

#### *At the exit face*

Gauge pressure –2451 (appox. 250 mm of water column)
Temperature 302 K

#### *At the inlet face*

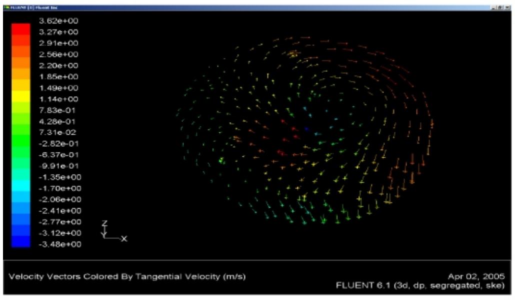
Gauge pressure 0
Temperature 302 K

Operating pressure 91326 Pa (approx. 685 mm of Hg) .

## Results and discussion

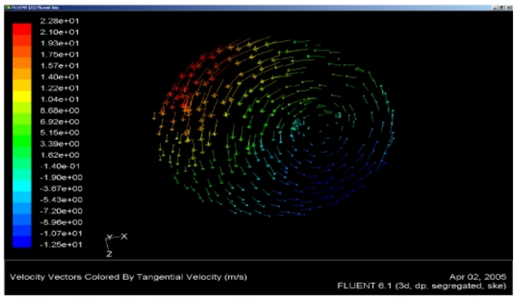
### *Variable swirl velocity plots for 5.74 valve lift (rightopen):*

100% opening



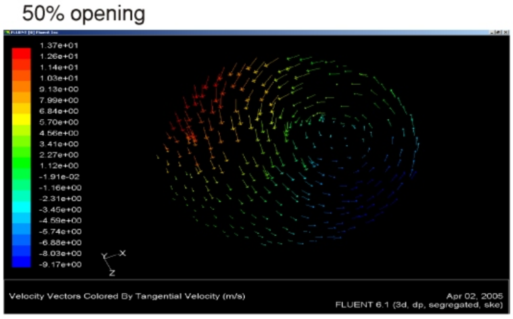

At paddle-wheel plane, facet average =1.0236 m/s
paddle speed = 1144 rpm

70% opening




At paddle-wheel plane, facet average = 2.933 m/s
paddle speed = 3276 rpm

50% opening



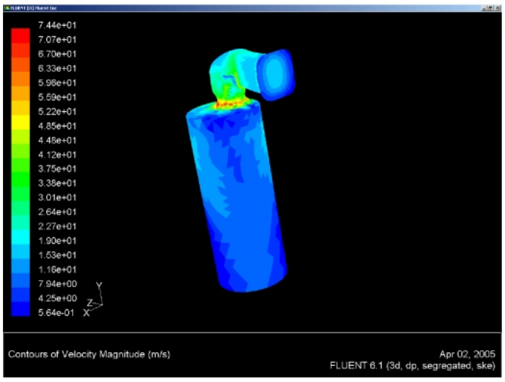

At paddle-wheel plane, facet average =1.4845 m/s
paddle speed = 1660 rpm

### *Velocity magnitude plots*

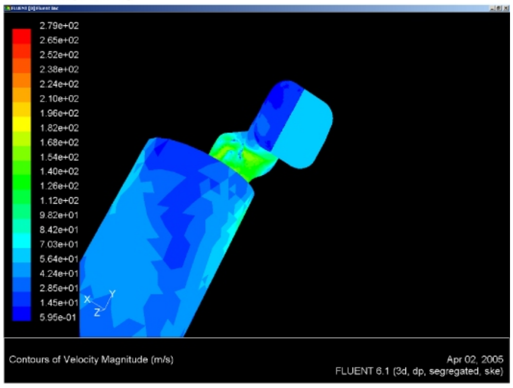
In the variable swirl analysis, the inlet face area of the port is varied according to the percentage openings. The velocity at the inlet is taken as 68 m/s and an outflow rate of 1 is used for the outlet boundary condition. From the tangential plots plotted above it is clear that for 70% opening the intensity of swirl is maximum. The maximum swirl energy is at the 70% port open condition. Therefore when the engine runs at lower speeds from 1500 to 1080 rpm, the possibility of smoke reduction is more. Because at lower speeds especially in conventional jerk type direct injection systems the kinetic energy of the spray is less, hence the increase in angular velocity of air due to obstruction plate improves the swirl, which ensures better combustion.

From the velocity plots it is clear that maximum velocity magnitude is at the 70% port opening. This increase in velocity of inlet air in helical ports increases the swirl generation capacity of the port. The increase in the swirl generation capacity of the port ensures better combustion at lower speeds. Also, it provides way to reduce both $NO_x$ and particulates, the two main

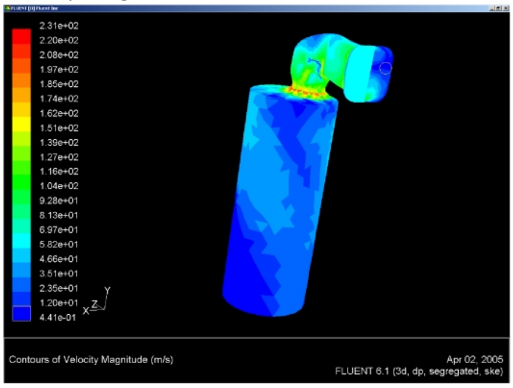
100% opening

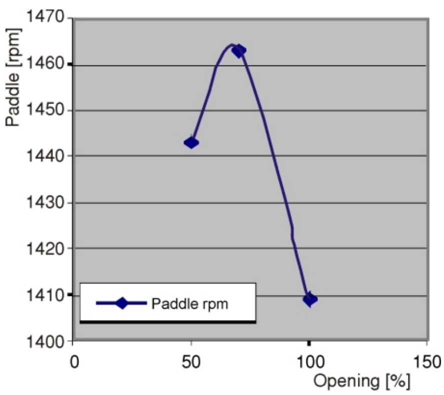



70% opening

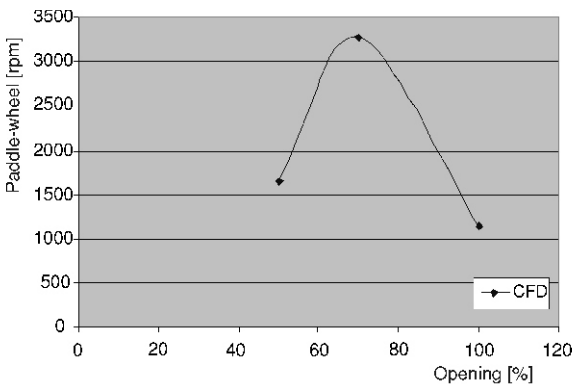



50% opening




**Variation of paddle-wheel rpm with port opening for W06DTIE2 engine – Experimental results**



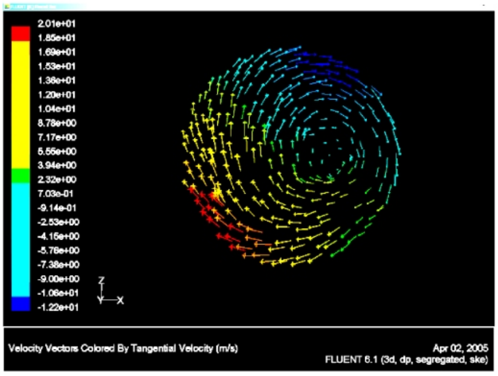

**Variation of paddle-wheel rpm with port opening for ALE2 engine – CFD results (5.74 mm valve lift)**

pollutants in diesel engines [5]. The increase in the swirl generating capacity of the port due to variable swirl plate ensures better combustion even with less amount of oxygen. The CFD curves shown are very well matched with the experimental results conducted on the helical port.

### *Variation of swirl by varying the side of closing*

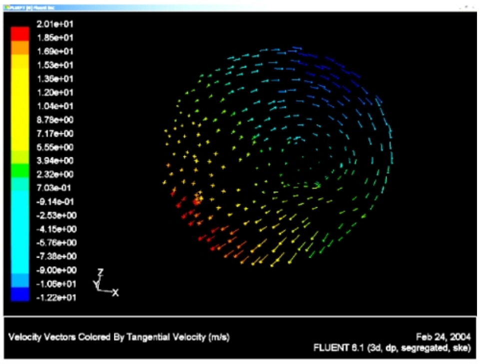
From the variable swirl study, at 5.74 mm valve lift it is found that there is nearly 50% increase in the angular velocity of the air when the right side valve is opened instead of left side, without the affecting the mass flow rate. Hence the swirl generating capacity of the port can be improved further [6].

Left 50% open




At paddle-wheel plane, facet average = 1.4845 m/s, paddle speed = 1660 rpm

Right 50% open




At paddle-wheel plane, facet average = 2.85 m/s, paddle speed = 3187 rpm

## Conclusions

From the above study it is clear that, the variable swirl for a specific valve lift, there is an increase in angular velocity for 70% opening of the intake port and decrease in angular velocity for 50 % opening of the port is clearly predicted by this model. Therefore this model has an application in improving the swirl generating capacity of the port and evolution/distribution of the in-cylinder swirl during intake process according to emission norms.

It can provide ways to design high swirl generation capacity intake port which reduces both $NO_x$ and particulates the two main pollutants in diesel engines, which is very necessary to attain Euro IV emission norm.

The increase in the swirl generating capacity of the port due to variable swirl plate ensures better combustion even with less amount of oxygen. For optimum design of the variable swirl plate, this methodology can be used.

The results indicate that the CFD model can be used as a tool to understand the effect of various parts of air intake system for optimization. This effect will reduce the number of experiments to be carried out for arriving at final optimized system.

In this study, the numerical simulation of the helical variable swirl intake port for 2-valve DI Diesel engines under steady state condition is discussed. Flow calculations are performed for the port-valve-cylinder system using the FLUENT- 6 code. The CFD curves shown are very well matched with the experimental results conducted in the helical port.

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