

The Effect of Tumble Flow on Efficiency for a Direct Injected Turbocharged Downsized Gasoline Engine

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ABSTRACT

Direct gasoline injection combined with turbo charging and down sizing is a cost effective concept to meet future requirements for emission reduction as well as increased efficiency for passenger cars. It is well known that turbulence induced by in-cylinder air motion can influence efficiency. In this study, the intake-generated flow field was varied for a direct injected turbo charged concept, with the intent to evaluate if further increase in tumble potentially could lead to higher efficiency compared to the baseline. A single cylinder head with flow separating walls in the intake ports and different restriction plates was used to allow different levels of tumble to be experimentally evaluated in a single cylinder engine. The different levels of tumble were quantified by flow rig experiments.

Two series of experiments were performed, one aiming to evaluate tumble in the region of low to medium load and engine speed, mainly focusing on efficiency, and one for the high load region to evaluate any negative consequences of increased tumble. The results indicate that tumble positively can influence the efficiency and emissions, however, the shape of the incoming flow dictate the level of impact significantly. Even for a relatively small change in tumble great differences in heat loss were seen. The efficiency increase seen originated mainly from lower heat loss through the exhaust gases. Additional gain came from lower in cylinder heat loss for the more favorable shape of the flow, where CFD indicates that the incoming air initially follows the cylinder head to a greater extent. Negative consequences are also associated with increased tumble. For instance, excessive pressure rise rates which can result in noise issues at higher loads. However, from a combustion perspective, the turbulence induced by the tumble positively effects the main parameters with reduced combustion duration, increased

stability and increased exhaust gas recycling tolerance as well as increased combustion efficiency, features that are beneficial especially for a direct injected down sized turbo charged concept.

INTRODUCTION

Future requirements to reduce exhaust emissions and fuel consumption are prompting the development of more advanced combustion concepts. Diesel engines have displayed high efficiency, but with the drawback of generating high amounts of NO_x and/or particles, necessitating the use of complex and expensive after-treatment systems. In contrast, traditional gasoline engines have shown the capacity to meet stringent emission standards, but with lower efficiency than diesel engines. Naturally, the aim is to combine high efficiency with the possibility of meeting ultra low emission levels. For the gasoline engine, direct injection (DI) have been identified to be positive for efficiency [1] and especially when combined with down-sizing through turbo charging [2-3]. The efficiency increase of DI and down-sizing combined with the potential to meet stringent future emission levels makes the concept promising for future vehicle [3,4,5].

With DI and turbo-charging the possibility to use scavenging at low engine speed and high loads allows higher loads to be achieved at low engine speeds compared to a port fuel injected (PFI) engine. This is due to a reduction of in-cylinder residuals by scavenging which limits engine knock [6], and an increased mass-flow through the turbine which increases the available boost. This available high load at low engine speed is essential for vehicle efficiency since it in practice allows a higher gear ratio.

However, several challenges associated with DI and down-sizing exists. High power outputs for small engine displacement lead to an increased ratio between the lowest and highest amount of fuel that need to be supplied to the engine. This is further enhanced if significant enrichment is required for component protection combined with retarded combustion phasing to limit peak in cylinder pressure. On the other hand, at idle or during start-up, very small amounts of fuel is needed. Consequently, a compromise regarding the choice of injector is required when balancing properties at low and high power. The down-sizing concept also leads to high peak in-cylinder pressures due to the higher loads, which in turn also influences the pressure rise rates. Consequently NVH can be negatively influenced since high pressure and pressure rise rates can lead to noise.

One way to cope with the mixing issues that can be associated with the concept of DI can be to use large scale air motion and generate turbulence. To distribute fuel droplets over the whole in-cylinder volume and on a microscopic scale enhances fuel air mixing [1, 7]. Turbulence can also positively help entrain EGR [1] and reduce combustion duration [7].

A tumbling air motion appears positive for the down-sized DI concept, since both a large scale air motion and turbulence can enhance mixing. Tumble can also lead to strong levels of turbulence near TDC [8-9] which can reduce the delay from spark to combustion [10] and reduce combustion duration [11-12]. The reduced combustion duration is desired for efficiency at part load where traditionally the duration is too long, it can also lead to reduced knock at high load [13]. However, one negative consequence of tumble for down-sized DI engines might be too great pressure rise rates at high loads induced by the shortened combustion duration combined with reduced knock sensitivity.

In the study presented here the aim was to evaluate different levels of tumble, focusing on down-sized turbo-charged DI gasoline engines. The effect of tumble was evaluated through single cylinder experiments and calculations. The effect on efficiency were quantified by using operational points typical for drive cycle operation and the effect on pressure rise rates for boosted operation. The intention with the study was to increase the tumble to such an extent that the efficiency started to decrease.

METHODOLOGY

In order to achieve different levels of tumble for the engine experiments a modified single cylinder head was used. This cylinder head used flow separating walls in the intake ports (see [figure 1](#)) which in combination with restriction plates (see [figure 2](#)) allowed the flow in the intake ports to be altered, in such a way that the tumbling air motion of the in cylinder flow could be influenced.

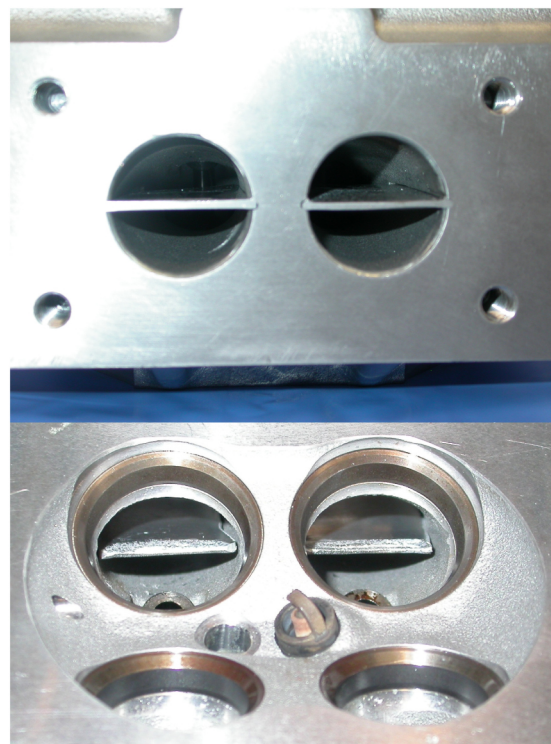


Figure 1. The intake ports with flow separating walls for the single cylinder head, seen from the side of the mating surface between cylinder head and the intake plenum (top) and seen from the combustion chamber side without intake valves (bottom).

By using different restriction plates between the intake port and intake plenum, different proportions of the flow could be achieved over and under the flow separating plates resulting in altered direction of the air entering the combustion chamber could be altered, see [figure 2](#). Naturally this procedure with flow separating walls and restriction plates will not perfectly replicate the behavior of conventional intake ports with more or less tumble. There are drawbacks associated with using this simplified procedure to evaluate tumble. It was assumed that the major drawback with this procedure compared to studying separate ports optimized for each level of tumble, were associated with the effects of area restriction caused by the fact that not the complete cross sectional port area were used when restriction plates were used. To limit the effect of this, the indicated work in the experiments were analyzed over the entire four strokes as well as separately over the compression and expansion strokes in order to separate the negative effect of the simplified procedure on pumping losses. However there would additionally be a difference in gas velocity between the simplified procedure and if separate ports optimized for each levels of tumble would have been used and it proved to be challenging to quantify any eventual effect of the difference in the momentum of the incoming air. In [14-15] changes to the intake ports have been used to limit the flow in

the lower parts of the intake pipes and thus force a greater proportion of the flow to the upper part of the port and valve when entering the combustion chamber and thereby achieve more tumble motion. Measurements were also made with restriction of the upper part of the port. I.e. to direct all the air under the flow restriction walls, with the assumption that this would lead to similar gas velocity (since the cross sectional area were similar), and thus impulse, as when all air were directed over the flow restriction walls but with the distinction that far less tumble would be achieved. So that the effect of the increased gas velocity alone of the simplified procedure could be isolated and thereby quantified. This assumption of that directing the air under the flow separating walls would lead to a low tumble value (which was required to isolate the effect of gas velocity) were proved to be incorrect and high tumble levels were achieved. This were interesting but led to that the effect of the gas velocity alone could not completely be quantified, since closing either the upper or lower part of the port led to higher tumble.

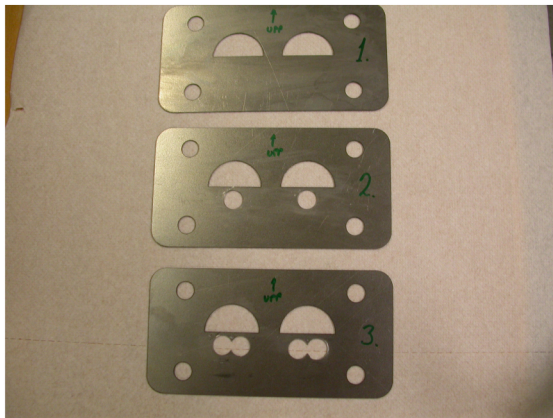


Figure 2. The different restriction plates used on the mating surface between the cylinder head and intake plenum.

A steady state flow rig were used to verify that different levels of tumble motion were achieved by using the different restriction plates. Five different levels of tumble motions were identified in the flow rig. These five settings were used in the single cylinder engine to evaluate the effect of tumble, where a mini-map, which represents typical drive-cycle operation points, were used to evaluate the potential of tumble for fuel efficiency increase. To understand how the tumble influence the engine behavior at higher loads, boosted operation were tested at different engine speeds. The measurement data from the single cylinder engine acted as the base of this investigation and CFD, flow-rig or optical measurements were used as complements to shed light on the seen trends from the engine experiments.

EXPERIMENTAL APPARATUS

SINGLE CYLINDER ENGINE

The engine used for the experiments was a single cylinder Volvo engine, see [figure 3](#), which especially is intended for combustion system optimization, parameters and settings for the engine can be seen in [table 1](#). This engine was equipped with variable cam timing (VCT) on both intake and exhaust which made it possible to phase the opening and closing times of the valves 60 CAD on the intake side and 30 CAD on the exhaust side. The engine management consisted of a Volvo Rapid Prototype System (VRPS) controlled with a PC. The system gave full control of all the main control parameters in real time. The engine was equipped with a conditioning system allowing the intake air pressure and temperature to be controlled. To limit any effect of differences in ambient conditions the intake air temperature was controlled to 24 °C and intake air pressure to 105 kPa just prior to the throttle, for the cases that did not require boosting. For the same reasons the exhaust back pressure was controlled to 105 kPa, if the mass flow for the specific case did not require a higher back pressure.

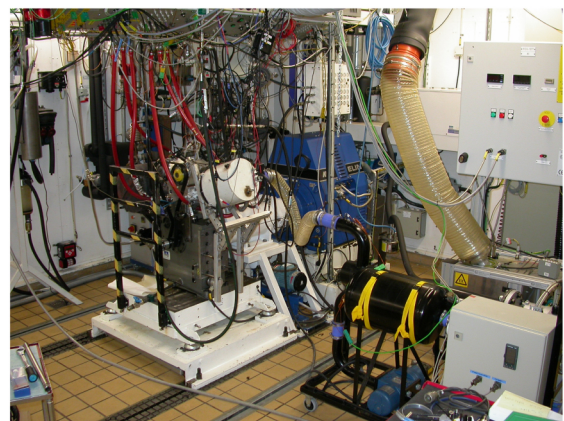


Figure 3. The Volvo single cylinder research engine.

In combination with the VRPS a FEV data acquisition system was used for real time measurements of both instantaneous and cycle based combustion parameters, essential to find settings with beneficial compromises in balancing the different demands while increasing efficiency. For emission measurements a Horiba MEXA 7100DEGR was used to measure the HC, NO, NO₂, O₂, CO and CO₂ emissions, allowing a rather complete understanding of the constituent components in the exhaust gases to be achieved.

Table 1. Single cylinder engine parameters and settings.

Displaced volume	492 cm ³
Stroke	82 mm
Bore	93.2 mm
Compression ratio	10.8:1
Exhaust Valve Open	-236.5 to -206.5CAD
Exhaust Valve Close	-3.5 to 26.5 CAD
Inlet Valve Open	-36.5 to 23.5 CAD
Inlet Valve Close	190.5 to 250.5CAD
Fuel	95 RON with 10 % ethanol
Fuel pressure	100 bar
Injector	Centrally mounted Denso 6 hole injector 17.5 cm ³
Water/Oil temperature	90 °C
Cylinder pressure transducer	Kistler 6061B (water cooled)
Exhaust and intake pressure transducer	Kistler 4045A5

Mini-map operation

To estimate fuel consumption and emissions for the most frequently used operational points for the Volvo customer or for drive cycle operation a mini-map were used, see [figure 4](#). For all points the ambition was to achieve highest efficiency while still meeting demands and requirements. During the optimization, measures of maximum tolerable combustion instability, manifested by the standard deviation of IMEP, were used specific for each mini-map point. So the target for any of the different levels of tumble was to achieve at least less than the required value of combustion instability for each of the mini-map points. This parameter acted mainly as guideline for setting the optimum valve timing over the map, especially for the lower loads where the maximum valve overlap was especially restricted due to the increasing internal EGR levels associated with increasing overlap, which in turn leads to increased instability. For the setting of injection timing a generalized maximum tolerable level of soot (manifested by FSN-value) were used, which no chosen settings were allowed to exceed. So while succeeding the soot and stability limits the injection timing, valve timing and combustion phasing were optimized for each mini-map point. However for the highest points in the mini-map, the optimum combustion phasing were compromised due to knock. The knock were quantified by the amplitude of the pressure oscillation above a certain frequency.

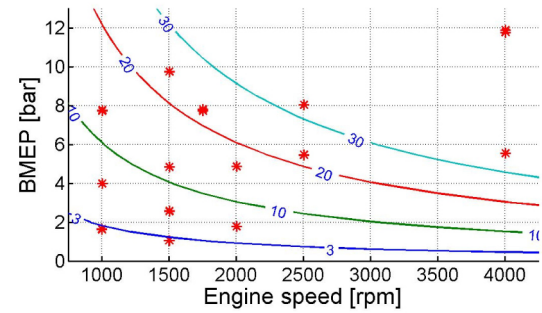


Figure 4. Speed (rpm) and load (BMEP) points for the mini-map, stars. The corresponding engine power [kW] of a 4 cylinder application, lines.

Boosted operation

The mini-map operational points merely represent the medium to low load and engine speed, which were most important for evaluating the potential in fuel consumption reduction. To get a more complete understanding on how the tumble influenced the engine behavior boosted operation were also tested. A fixed intake air pressure of 152 kPa were used over a span of different engine speeds. As for the highest loads in the mini-map, knock compromised the optimum combustion phasing and the combustion phasing were adjusted to meet the chosen level of pressure oscillation amplitude.

FLOW-RIG

The steady state flow-rig used for the experiments can be seen in [figure 5](#). The rig was set up for tumble measurements for this study. The single cylinder head was mounted on a T-shaped pipe where the tumbling torque for steady state flows for the different valve lifts where measured on a honeycomb structure at both ends of the “T”.

A tumble number was calculated for each of the tested configurations, as a Ricardo tumble number, R_t [16]:

$$R_t = L_D \frac{\int_{\alpha_1}^{\alpha_2} C_f \cdot R \, d\alpha}{\left(\int_{\alpha_1}^{\alpha_2} C_f \, d\alpha \right)^2} \quad (1)$$

L_D is an engine shape factor:

$$L_D = \frac{B \cdot S}{n \cdot D^2} \quad (2)$$

Where B is the bore, S is the stroke, n is the number of valves and D is the diameter of the valve seat. C_f is a flow coefficient:

$$C_f = \frac{4 \cdot \dot{V}}{D^2 \cdot \pi \cdot V_0 \cdot 3.6 \cdot 10^{-3}} \quad (3)$$

$\dot{V}$ corresponds to the volume flow and V_0 is a reference velocity. R represent the tumble for each specific valve lift:

$$R = \frac{8 \cdot (T_{left} + T_{right})}{\dot{m} \cdot B \cdot V_0} \cdot 3.6 \cdot 10^6 \quad (4)$$

T_{left} and T_{right} are the measured tumble torque for each end of the T-shaped tube. The Ricardo tumble number is calculated from IVO (α_1) to IVC (α_2) and attempts to represent the in cylinder tumble for engine operation based on steady state flows.

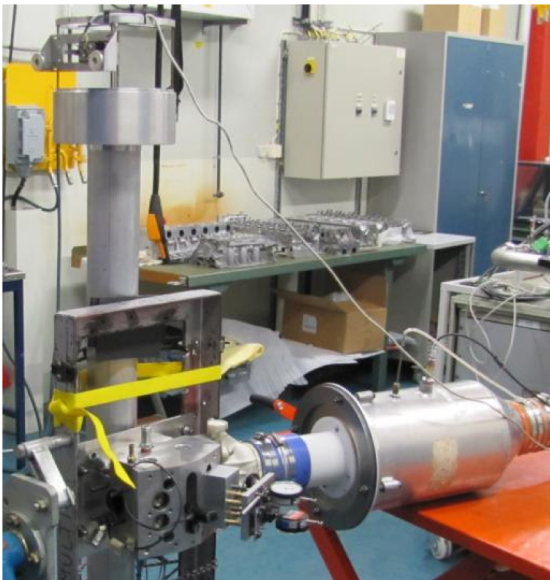


Figure 5. Steady state flow rig with tumble measurement setup.

CFD

CFD calculations were performed to investigate the effect of different tumble levels on the in cylinder flow fields. The commercial STAR-CD (version 3.26) software was used to perform 3-D in-cylinder runs. The mesh represented a single-cylinder engine consisting of a combustion chamber, intake ports and exhaust ports. The moving mesh generation necessitated by the movement of piston and valves, was taken care of by the commercial es-ice software. CFD runs started during the exhaust stroke, and continued until combustion TDC. Combustion modeling was not included in the runs, but

since the complete intake and compression strokes were evaluated, together with the fuel injection event, the mixing of fuel and air as well as the in-cylinder velocity field could be analyzed. This approach makes CFD an excellent tool to study the direction and magnitude of the air jet set up by the intake ports in the combustion chamber, as well as the break-down of large-scale tumble motion into turbulent kinetic energy near combustion TDC.

Figure 6 shows two sections through the CFD mesh applied in the calculations. Note that the valve looks a bit odd in the figures. This is due to the fact that the figures show a mesh template on which es-ice bases the final meshes used during the calculation. Boundary conditions applied on the intake/outlet boundaries of the ports were taken from 1-D GT-Power gas exchange calculations and consisted of crank angle-resolved pressure and temperature data. The turbulence model chosen for the CFD analyses was the RNG version of the k- ϵ model.

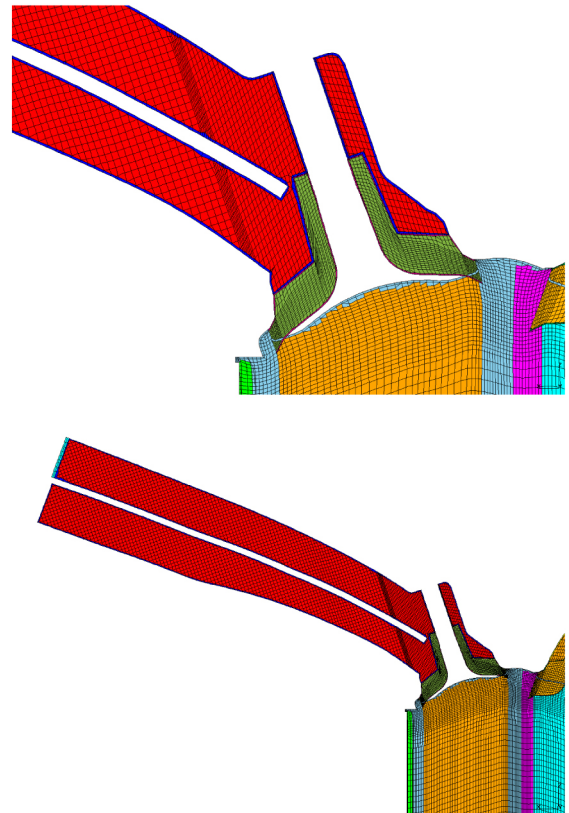


Figure 6. Sections through the CFD mesh showing the tumble plate in one of the intake ports.

OPTICAL SINGLE CYLINDER ENGINE

An optical single cylinder Volvo engine, see figure 7, was used as complementation to the conventional single cylinder engine, parameters and settings for the optical engine can be seen in table 2. Both engines had similar setups, the stroke