

Investigating the Limits of Charge Motion and Combustion Duration in a High-Tumble Spark-Ignited Direct-Injection Engine

Roy Ogink and Aristotelis Babajimopoulos
Volvo Car Corporation

ABSTRACT

This paper describes the experimental study of a tumble-flap mounted in the intake port of a single-cylinder spark-ignited gasoline engine. The research question addressed was whether an optimal tumble level could be found for the combustion system under investigation. Indicated fuel consumption was measured for a number of part-load operating points with the tumble-flap either open or closed. The experimental results were subjected to an energy balance analysis to understand which portion of the fuel energy was converted to work and how much was lost by incomplete combustion, heat losses to walls and to the exhaust gases, as well as to pumping losses.

Closing the tumble-flap resulted in reduced fuel consumption only in a small area of the operating map: only at low-speed, low-load operation, a benefit could be obtained. One way of achieving this was by means of de-throttling due to increased valve overlap, since the combustion system could tolerate a higher level of internal exhaust gas recirculation (EGR) when the tumble-flap was closed. Improved combustion efficiency was another way of achieving lower fuel consumption.

For higher engine speeds and loads, fuel consumption actually deteriorated for closed tumble-flap operation, indicating that an optimal tumble level exists and that it was reached in this study. The reason for the worse fuel consumption was that the intake ports in the baseline combustion system already set up such a strong charge motion that a further increase of turbulence (by closing the tumble-flap) no longer contributed to improved conversion of fuel into products or a higher tolerance to internal EGR. On the contrary, it was found that restricting the intake port resulted in increased pumping losses. A tendency of increased wall heat losses for the highest tumble levels could also be observed. Combustion duration generally became shorter for higher levels of in-cylinder turbulence, but for some operating points with extreme tumble level combustion speed actually decreased.

CITATION: Ogink, R. and Babajimopoulos, A., "Investigating the Limits of Charge Motion and Combustion Duration in a High-Tumble Spark-Ignited Direct-Injection Engine," *SAE Int. J. Engines* 9(4):2016, doi:10.4271/2016-01-2245.

INTRODUCTION

This paper attempts to determine whether the well-documented benefits of tumble and increased turbulent kinetic energy (TKE) deteriorate after an optimal tumble level has been reached for the combustion system under investigation. This combustion system is a state-of-the-art downsized turbocharged direct-injected SI engine with a centrally-mounted fuel injector. As will be described below, intake port design is a trade-off between trapping air and setting up a strong tumble motion. Since the combustion system described here is heavily turbocharged, the intake port shape was chosen to set up a strong tumble motion in the combustion chamber, rather than focusing on filling the chamber with fresh charge. As described in several papers [1, 2, 3, 4], the large-scale tumble motion that is set up during the intake stroke can be broken down into small-scale turbulence towards the end of the compression stroke, enabling fast and stable combustion. Since it is this increased turbulence level prior to ignition that speeds up combustion, instead of the large-scale tumble motion, the question that actually needs answering is whether an optimal TKE level does exist. In this paper it is assumed that a

higher tumble level will result in a higher level of TKE, so the focus is the experimental investigation of different tumble levels. Furthermore, it is not possible to directly measure the in-cylinder turbulence level in a (non-optical) combustion engine. To study the breakdown of tumble into turbulence, extensive CFD analyses have been performed by the authors for operation with opened and closed tumble-flap. The results are discussed to some extent in this paper, but the main focus is on the experimental investigation of different tumble levels.

One advantage of high-tumbling intake ports, besides fast and stable combustion, is a higher tolerance to internal EGR at part-load, leading to reduced pumping losses due to de-throttling. This phenomenon is described in detail later in this paper. It is the authors' experience that the enhanced large-scale motion inside the chamber can promote air/fuel mixing, but careful attention should be paid to the fuel injector spray targeting: a higher tumble level might give rise to an increase in fuel wall wetting. A possible disadvantage of the additional charge air motion connected to high-tumbling intake ports

is mentioned by Wheeler et al. [4], who found that the initial flame development times (CA010) increased for higher tumble levels. They suspected that the high in-chamber gas velocities were distorting the flame propagation. Interestingly, in the current study, CA010 became significantly shorter for increased tumble levels, as described later. This might indicate differences in the in-chamber velocity field or in the spark plug location and protrusion.

Wheeler et al. [4] also discussed the increased convective heat transfer from the chamber walls to the in-cylinder charge during the intake and compression strokes. By means of CFD analyses, they showed that the average in-cylinder gas temperature at spark timing increased by 5 to 30 °C when switching from high-tumbling to extremely high-tumbling intake ports. That temperature increase explained their observation from engine testing that the high-tumbling system provided good knock resistance, but that extremely high tumble ratios increased the knocking tendency.

High-tumbling intake ports provide a higher tolerance to charge dilution by air and/or EGR. In other words, the engine can be operated with higher EGR ratios or leaner mixtures until combustion stability requirements are no longer met. This advantage for high tumble ratios was described by Wheeler et al. as well [4]. As drawbacks of high-tumbling systems, some authors mentioned higher heat losses [3, 5], increased pumping losses [6] and NVH-issues due to too fast, i.e. noisy, combustion [3, 6].

Considering the aforementioned advantages for a high-tumbling combustion system characterized by a fast and stable combustion process, the question comes to mind whether an optimal tumble level can be found, or whether the tumble ratio should be as high as possible. The main focus of this paper is to shed more light on this topic.

CONCEPTUAL ANALYSIS

The Otto cycle describes the four-stroke spark-ignition process by means of an idealized thermodynamic cycle consisting of an isentropic compression of air, addition of heat at constant volume, isentropic expansion of air, and rejection of heat at constant volume [3]. It can thus be noted that combustion is instantaneously fast, no heat transfer occurs, the air is used as the medium behaving as a perfect gas with constant specific heat and that gas exchange is not included in the Otto cycle. To study the differences between the hypothetical Otto cycle and the real processes occurring in combustion engines, a closed-cycle 0-dimensional simulation tool was applied to study the effect of parameters such as combustion duration, combustion phasing and heat transfer on thermal efficiency. The tool simulates only the compression and expansion stroke from bottom to top dead center (BDC to TDC). Combustion is described by means of a Wiebe [3] function and a 2-zone model with 15-species chemical equilibrium in the burned zone. The Woschni model [3] is used to simulate heat transfer to the walls and gas properties are estimated using NASA polynomials.

For the results presented in this section, the geometric compression ratio is 10:1, the engine speed is 2000 rpm, and all wall temperatures are set to 450K. The pressure and temperature at BDC are 1 bar and

350 K respectively, the fuel is iso-octane and the equivalence ratio is 1. A multiplier between 0 and 1 is used to scale the total heat transfer (i.e. the engine is adiabatic when the multiplier is 0).

Figure 1 shows contour plots of the indicated thermal efficiency as a function of combustion phasing (location of 50% burned fuel, CA50) and 10-90% burn duration (CA1090) for adiabatic walls and also for the heat transfer multiplier set to 0.5. It can be seen that for adiabatic walls (left), the optimum combustion phasing is at TDC (0 CAD) regardless of combustion duration. This is of course due to the symmetry of the Wiebe function. The maximum efficiency (approximately 45.5%) occurs for instantaneous combustion. This is lower compared with the efficiency of the ideal Otto cycle for the same compression ratio, which is approximately 60%, because of the lower ratio of specific heats and dissociation.

For non-adiabatic conditions (right), it can be observed that thermal efficiency drops even further. However, it now peaks for a certain optimal combustion duration: too fast combustion results in high in-cylinder gas temperatures, and thus increased heat losses to the combustion chamber walls. On the other hand, when combustion is too slow, a large portion of the fuel energy is released when the piston is traveling downwards during the expansion stroke, resulting in a reduction of piston work. The second conclusion that can be drawn is that the ideal combustion phasing differs from the ideal Otto cycle. In a real engine the optimum combustion phasing is determined by the trade-off between increased heat losses and reduced expansion work. Moreover it can be seen that the optimum CA50 is retarded as combustion duration increases.

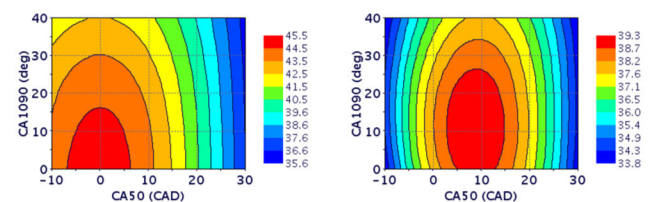


Figure 1. Indicated thermal efficiency as a function of combustion phasing (CA50) and burn duration (CA1090): Adiabatic walls (left) and Non-adiabatic walls (right).

Figure 2 shows contour plots of the indicated thermal efficiency as a function of the heat transfer multiplier and burn duration. This plot has been generated for a fixed CA50 of 10 CAD. It is interesting to note that as heat transfer increases, the burn duration for maximum thermal efficiency, indicated by the dashed line, also increases. This suggests that the trade-off between increased heat losses and reduced expansion work shifts and longer burn duration is required to account for increased heat transfer.

Considering that higher tumble and TKE result in both shorter burn duration and increased heat transfer coefficients, the aforementioned analysis suggests that there must be a limit to how much tumble an engine can tolerate before heat transfer effects dominate and the benefits of shorter burn duration disappear. An optimal (i.e. minimum) combustion duration is thus expected to exist, and a

corresponding maximum tumble level should be associated with that. Tumble ratio and filling are discussed in the next section, where intake port design is discussed.

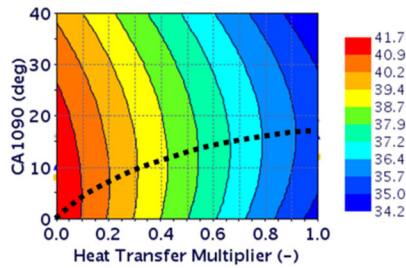


Figure 2. Indicated thermal efficiency as a function of CA1090 and heat transfer multiplier for CA50 = 10 CAD.

INTAKE PORT DESIGN

Turbocharging enables the pressure in the intake manifold to rise substantially above atmospheric pressure, forcing air to be trapped into the combustion chamber. Consequently, intake port design focus can be shifted from filling characteristics to charge motion properties. In short, it is a challenge to design intake ports that can deliver enough air for the engine to produce the required horsepower level, while at the same time setting up a strong tumbling motion that can be converted into small-scale turbulence to support fast and stable combustion. Furthermore, as discussed in the previous section, the tumble level should preferably be approaching its optimal level, i.e. as high as possible, but not too high. A typical intake port design process, including steady-state and transient CFD analyses as well as single-cylinder testing, was described in detail by the authors in [2]. An example of a scatterplot containing mass flow and tumble level results from steady-state CFD for a number of intake port designs at three different valve lifts can be found in Fig. 3. The trade-off between filling and tumble level can clearly be seen.

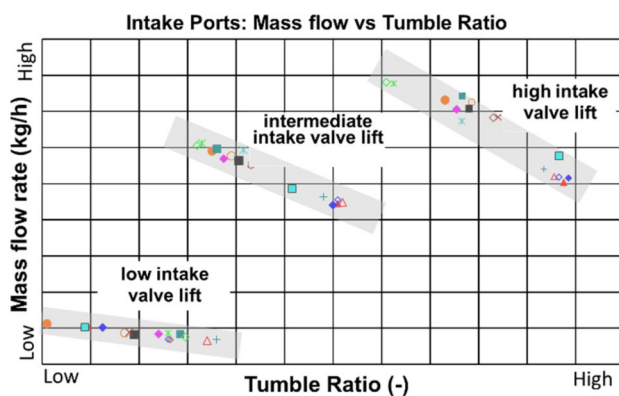


Figure 3. Example of scatter plot obtained from steady-state intake port CFD.

Two examples of intake port designs have been plotted in Fig. 4. The design on the right has a very straight shape, which is typical for a high-tumbling port, where most of the air flow is directed over the upper side of the intake valves to set up a strongly rotating large-scale motion inside the chamber. The port on the left is more parallel to the

valve stem, resulting in a more bend shape and better filling characteristics. It should be noted that both ports generate higher tumble compared with many other engines on the market.



Figure 4. Intake port design for a Fill port (left) and a Tumble port (right).

Ideally, the large-scale motion should be effectively converted into small-scale turbulence towards the end of the compression stroke to support the spark kernel development and flame propagation processes. Unfortunately, the turbulence level usually drops rapidly as the piston approaches TDC (i.e. 720 CAD). This can be observed in Fig. 5, which shows results from transient CFD analyses for the two port designs of Fig. 4. The instantaneous tumble level is plotted on the left, while the TKE level is shown on the right.

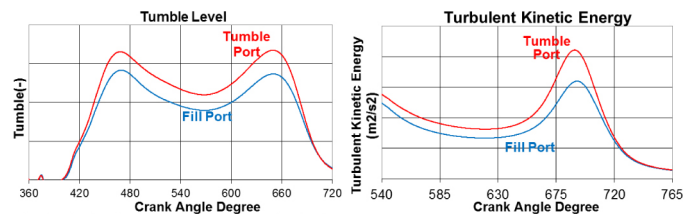


Figure 5. Transient CFD results for Tumble and Fill ports: Tumble level (left) and Turbulent Kinetic Energy (right).

Since intake port design is a compromise between good filling and a high tumble level, attempts have been made to shift the balance of this trade-off by adding a tumble-flap to a filling-oriented port [6, 7, 8, 9, 10]. The next section discusses the advantages and disadvantages of such a system.

Tumble-Flap

The most common application of a tumble-flap is by adding a divider plate into the cylinder head splitting the intake port channels into an upper and a lower part. The flap is located upstream of the ports: opening the flap allows the air to flow through both the upper and the lower parts of the intake ports. Consequently, the intake ports almost behave as those in an ordinary engine would do, except for the fact that the divider plate and flap will slightly affect the mass flow and tumble level. When the tumble-flap is closed, the lower part of the intake ports is blocked and the air is forced into the upper part of the intake runners, forcing the air to rush mainly over the upper half of the intake valves. In this way, a strong air jet is set up in the combustion chamber during the intake stroke. An example of such a divider plate and its effect on the velocity field in the intake ports and combustion chamber can be observed in Fig. 6. The figure shows results from steady-state CFD analyses for a baseline port without divider plate (left) and the same port with divider plate and the lower part of the port blocked (right).

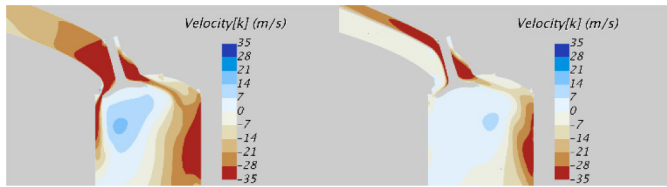


Figure 6. Axial velocity component from steady-state CFD for an intake port without divider plate (left) and with divider plate and closed tumble-flap (right).

Figure 7 shows the effect of closing the tumble-flap on mass flow rate and tumble ratio and how the trade-off between filling and charge motion shifts. It can be concluded from the CFD results that the relatively straight trend line, as was observed in Fig. 3, does no longer hold once results for closed flap are added: the tumble-level does indeed rise significantly, but the mass flow does fall dramatically due to the reduction in effective port flow area. Nevertheless, this is not a problem, since the idea is to design base intake ports that provide enough air flow at high engine speeds and loads. In that part of the operating map, the tumble-flap will be open to not restrict the air flow, but the in-cylinder turbulence level will be relatively high anyway due to the high piston speed and the fact that the engine is operating with wide-open throttle. For low-speed, low-load operation, on the other hand, piston velocity is low and the throttle will be partially closed leading to a low level of TKE prior to combustion. Consequently, closing the tumble-flap in that part of the operating map will contribute to faster and more stable combustion. This increased stability improves the internal EGR tolerance, which in turn means that the valve train settings can be modified towards larger valve overlap. To fill the chamber with enough fresh air, next to the additional EGR gases, the engine has to be de-throttled resulting in lower pumping losses and better fuel consumption. An example of this higher EGR tolerance giving rise to a reduction of pumping losses will be presented in the Experimental Results-section.

Another benefit of adding a tumble-flap is that less strain is put on the charging system: the tumble-flap enables an intake port design that is more oriented towards filling, characterized by better flow coefficients and lower pressure drop. As a result, the same air mass flow can be achieved with a lower boost pressure compared with when a high-tumble port design would have been applied. Obvious disadvantages of a tumble-flap system are added complexity, cost and weight. NVH might also be an issue, since mode-switching between flap activation/de-activation will occur when the engine's operating point is moving from one part of the operating map to another.

Based on the discussion above, a tumble-flap is expected to contribute to fuel consumption reduction in the low-load, low-speed area of the map, while at the same time enabling a high power output and less strain on the boosting system. Since the tumble-flap system adds cost and complexity, the aforementioned advantages should outweigh the disadvantages and the area of the map where a fuel consumption (FC) gain can be obtained should thus be maximized. The following sections deal with results from single-cylinder tumble-flap testing and it will be discussed in which operating points FC could be reduced. Energy balance analyses have been carried out

to get a detailed understanding of how the total fuel energy is converted to work and losses, such as incomplete combustion, heat transfer and pumping losses.

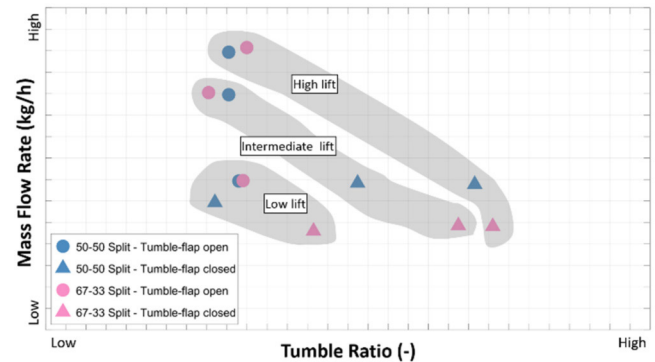


Figure 7. Steady-state CFD results for intake ports with opened and closed tumble-flap.

EXPERIMENTAL SETUP

The experimental investigation of the tumble-flap device was performed on a four-stroke, four-valve single-cylinder engine. The combustion chamber was equipped with a pent roof in the cylinder head. Both the spark plug and fuel injector were mounted in the cylinder head in between the intake and exhaust valves. Engine details are listed in Table 1.

Table 1. Experimental set-up of the four-stroke, single-cylinder engine.

No. of cylinders	1
Stroke	94.6 mm
Bore	82 mm
Compression ratio	10.5:1
Injector	Centrally-mounted, direct-injection, multi-hole
Fuel	Gasoline, RON95, 10% ethanol
Intake Valve Settings	VVT
Exhaust Valve Settings	VVT

The engine was operated by setting the exhaust pressure and exhaust VVT to the values determined for each operating point during the mapping of the multi-cylinder version of the engine. The intake pressure and intake VVT were then optimized to reach the desired load and achieve the lowest BSFC.

During the engine experiments the following data were stored: air and fuel mass flow, pressures, temperatures and emissions. The cylinder pressure transducer was mounted in the cylinder head and measurements were carried out with a resolution of 0.1 CAD. Gaseous engine-out emissions were measured dry with a Horiba MEXA 9100. A Coriolis mass flow meter was applied to measure the fuel flow.

The tumble-flap was mounted in the intake pipe, upstream of the cylinder head. The complete tumble-flap system, including divider plate, is shown in Fig. 8. The divider plate is sticking into the cylinder head intake ports. Three sets of divider plate and corresponding flaps