

Analysis of Different Gasoline Combustion Concepts with Focus on Gas Exchange

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ABSTRACT

Novel combustion technologies, which de-throttle the gasoline spark ignition (SI) engine, show high potential in reducing the fuel consumption. Technologies like variable valve actuation and/or gasoline direct injection, allow new strategies to run the SI engine unthrottled with early inlet valve closing (SI-VVA), charge stratification (SI-STRAT) and controlled auto ignition (CAI), also known as gasoline homogeneous charge compression ignition (HCCI). These diverse combustion concepts show thermodynamic gains that stem from several, often different, sources.

A multitude of definitions of thermodynamic gas exchange potentials arise when looking at the various publications concerning de-throttled combustion concepts. This paper shows a summary and comparison of these definitions and points out which one can be applied in general to evaluate various combustion concepts under the same basis of evaluation.

Additionally, a comparison between de-throttled gasoline combustion concepts at two representative engine operating points and a load sweep for the SI-VVA combustion concept is shown to illustrate the different interpretations of thermodynamic analysis of the gas exchange process. Advantages of the different combustion concepts are analyzed using thermodynamic split of losses method, which starts its potential analysis with the ideal constant volume cycle. A combustion concept comparison regarding these thermodynamic potentials, especially concerning gas exchange losses, is also performed.

The purpose of this paper is to propose a clear definition of thermodynamic benefits of de-throttled gasoline combustion concepts.

INTRODUCTION

The demand for lowering the fuel consumption of gasoline engines to limit the CO₂ emission, i.e. the fuel consumption, strongly advances the development of novel combustion concepts.

In an earlier work [1] we compared the different combustion concepts:

- spark ignition, homogenous mixture using port fuel or direct injection (SI-PFI, SI-DI)
- spark ignition, homogenous mixture with early inlet valve closing (SI-VVA)
- spark ignition, homogenous mixture with early inlet valve closing and high internal EGR (SI-VVA+)
- spark ignition, stratified mixture with direct fuel injection (SI-STRAT)
- controlled auto ignition (CAI) also known as gasoline homogeneous charge compression ignition (HCCI)

There we showed in detail the benefit of each combustion process considering the thermodynamic split of losses.

All concepts can benefit from variable valvetrain to improve fuel consumption and/or emissions.

Using conventional homogeneous stoichiometric mixture, a variable valve train gives the possibility of managing load by means of trapping the needed amount of fresh air at atmospheric pressure rather than throttling the intake manifold.

Stratified combustion, with an overall lean mixture, has a fuel rich to lean and an inert gas zone only comprising air and residuals, similar to a DI diesel process. Using quality control to manage the load, there is always atmospheric pressure in the intake manifold. In this case, variability of the valve train system could be helpful to drive higher internal EGR to lower NO_x emissions.

The DI combustion process benefits also from adapted valve timings, especially for wide-open throttle.

The CAI or gasoline HCCI combustion process involves the auto ignition and combustion of a premixed charge, which consists of air, fuel and a considerably high amount of hot residual gas. The auto ignition starts at the end of the compression stroke when the cylinder charge reaches a determined temperature and pressure. The necessary thermodynamic mixture properties can be achieved, e.g., by residual trapping or reinduction using a VVA system.

Figure 1, which stems out of [1], shows that the gas exchange losses, which are more or less influenced by the VVA system, state the third highest loss source for the thermodynamic process and the loss source with the highest differences between the combustion concepts.

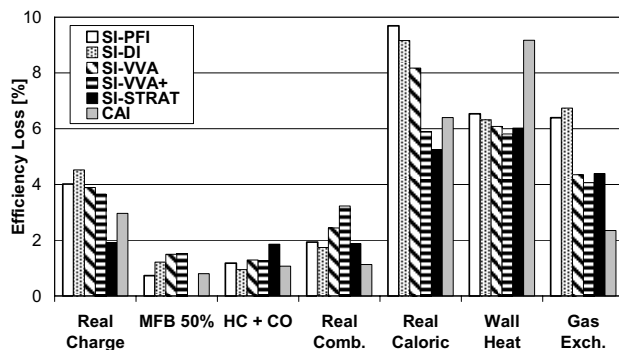


Figure 1 – Individual thermodynamic losses compared to ideal constant volume cycle at 2000 rpm, 3 bar NMEP

In this paper, we want to extend the analyses we did in [1], putting the emphasis on the gas exchange losses.

The analysis shown in this article, bases on experimental data of a single-cylinder research engine using centrally-mounted direct fuel injection [2] and fully variable electro-hydraulic valve-train [3] to enable SI-VVA+ and CAI operation. The Pressure Trace Analysis and the Thermodynamic Split of Losses analysis tools [4] are applied to evaluate the thermodynamic losses of each combustion concept, with focus here on the pumping losses.

DIFFERENT DEFINITIONS FOR THE GAS EXCHANGE OR PUMPING LOSSES

When referring to gas exchange or pumping losses, one has to realize that there are different definitions of what is called pumping work or pumping losses.

The most common method is to define pumping work as the integral of cylinder pressure with respect to cylinder volume calculated from BDC of the exhaust stroke to BDC of the intake stroke.

Looking at the cylinder pressure diagram of a conventional throttled gasoline engine in Figure 2, the sum of Area B and Area C represents the pumping work, known as “Pumping Mean Effective Pressure” (PMEP). Area A plus Area C is defined as “Indicated Mean Effective Pressure” (IMEP) and represents the indicated positive work per displacement volume. “Net Mean Effective Pressure” (NMEP) is the difference between IMEP and PMEP and therefore equal to Area A minus Area B.

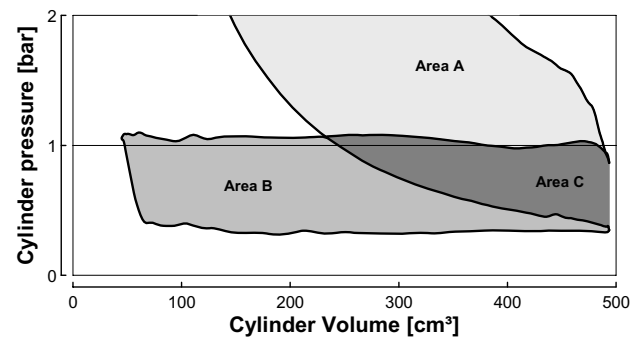


Figure 2 – SI throttled operation engine work cycle diagram with Areas A, B and C

With the upcoming of variable valve-trains another definition came up, which considers only Area A as indicated positive work and Area B as pumping work. In this definition, Area C is called a fictive quantity, which is cancelled out anyways when calculating NMEP [5].

This definition, however, includes some disadvantages described, e.g., in [6] and [9].

With the feasibility of early intake valve closing it is possible to reduce Area B dramatically while Area C stays constant or even increases. Without considering Area C in the pumping losses, the losses in Area C are assigned to the high-pressure cycle. This could lead to wrong conclusions regarding possible improvement of a combustion concept.

Löbbert [8] did a simulation with constant combustion properties at different loads and showed that with neglecting Area C in the pumping work the high-pressure efficiency depends on the load (NMEP) although the combustion parameters like burn duration and MFB50% stay the same. The reason is the wrong balancing of the pumping losses.

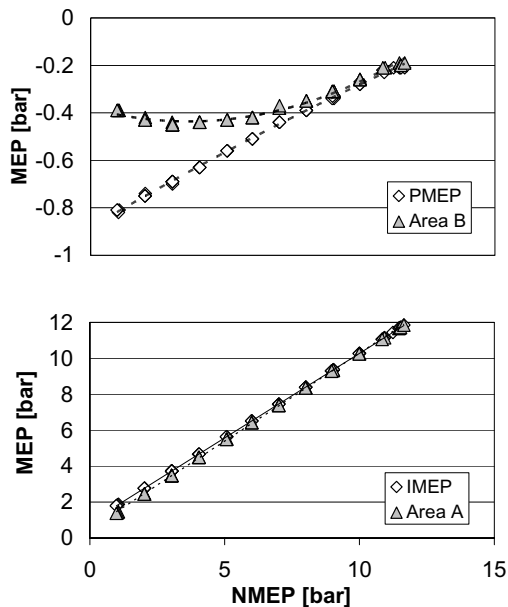


Figure 3 – SI conventional throttled at 2000 rpm: PMEP compared with Area B and IMEP compared with Area A

We can confirm these simulated results with measured results of our research engine, introduced in the next chapter. Figure 3 shows the differences in the pumping work as well as in the indicated high-pressure work using the different definitions of gas exchange losses for the conventional throttled combustion process. Considering only Area B for the gas exchange and Area A for the high-pressure cycle leads to lower pumping losses at low loads but decreases the positive work by the same amount. Accounting only Area A in the high-pressure cycle reduces the high-pressure efficiency dramatically by decreasing load. This looks rather implausible because the main parameters which effects the high pressure process, – burn duration and MFB50% – stay nearly constant (Figure 4).

Only at very high and very low loads the high-pressure efficiency is really affected by the combustion process. Low loads show longer burn duration, while operating at WOT the ignition angle needs to be retarded due to knock, thus resulting in a late MFB50% and decreasing high pressure efficiency.

Especially the energy balance of the thermodynamic analysis gives a clear hint that Area A does not represent the complete high-pressure work (Figure 4). One can recognize that the energy balance using Area A goes down to 87% at low load, which is way too small. Using IMEP the energy balance stays in the tolerable area between 95% and 102 % for every operating point.

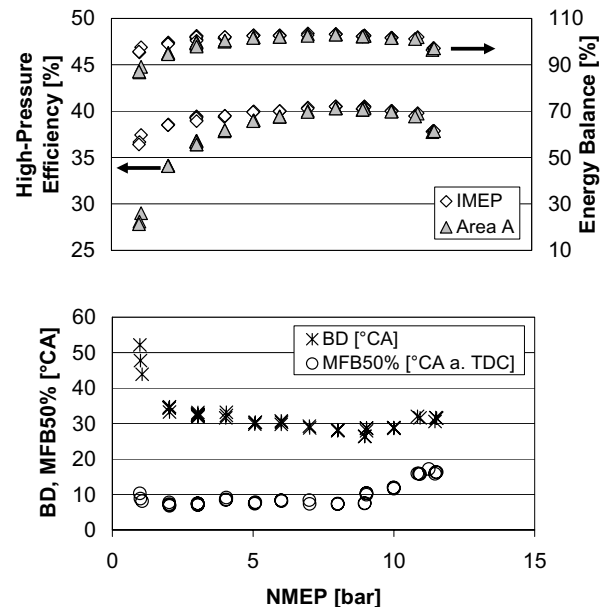


Figure 4 – SI conventional throttled at 2000 rpm: high-pressure efficiency and energy balance calculated with IMEP and Area A; burn duration and MFB50%

Figure 5 shows the same quantities as Figure 3 but for a combustion process with early inlet valve closing. It is obvious that Area B nearly disappears using this valve strategy at low loads, but the high-pressure efficiency is also incorrect (Figure 6). Already the progression of Area B in Figure 5 gives a strong hint that the energy balance in this case must be wrong. It is unlikely that the pumping losses increase with increasing load.

Operating the engine dethrottled with early inlet valve closing leads to longer burn duration, especially in case of low loads (compare Figure 4 with Figure 6). This can be attributed to several effects. After inlet valve closing there is only little air motion in the chamber which impairs the conditions for spraying, i.e., good mixture formation. Furthermore, the charge is stretched in the following expansion period from inlet valve closing to BDC where turbulence gets dissipated, thus reducing the turbulent flame speed. With SI-VVA the high pressure cycle shifts towards a lower load and must be operated with an earlier spark advance to reach an optimum combustion phasing MFB50%. All effects together make it difficult to take advantage of the potential associated with early intake valve closing at very low loads [10]. Practical solutions for this engine set-up are under evaluation.