

Study of Unconventional Cycles (Atkinson and Miller) with Mixture Heating as a Means for the Fuel Economy Improvement of a Throttled SI Engine at Part Load

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ABSTRACT

This contribution is focused on an investigation of two well-known techniques, i.e. the modified Atkinson working cycle with a late intake valve closing (LIVC) and the Miller working cycle with an extreme early intake valve closing (EIVC) in order to increase the fuel economy of a throttled SI engine at a part load (high throttled mode). However, the application of the Atkinson and Miller cycle causes a decrease in the in-cylinder charge temperature before the compression stroke. In the case of a constant value of the geometric compression ratio, the in-cylinder charge temperature at the beginning of the combustion is also decreased and the combustion is then slower (compared to a standard Otto cycle). This could negatively influence the indicated efficiency of the unconventional cycle. In order to avoid this, increase in the in-cylinder charge temperature was provided due to mixture heating in the intake manifold of the engine.

A commercial 1-D code was utilized in order to calculate the in-cylinder pressure and the fuel economy improvement for both unconventional cycles with and without mixture heating. Numerical simulations were carried out in two various engine speeds. A predictive turbulent flame model for SI engine, as a part of applied 1-D code, was used for combustion calculation. Data from experiments carried out on the SI engine with the Otto cycle were used for calibration of the engine model as well as of the predictive model of combustion.

Based on results from simulations, a significant increase in fuel economy improvement has been observed compared to the Otto cycle, especially due to the application of the Miller cycle. Further fuel economy improvement has been achieved due to a combination of both unconventional cycles and the mixture heating. During simulations certain limits have been observed at both Atkinson and Miller cycles.

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INTRODUCTION

Based on results from measurements of a car equipped with a spark ignition (SI) engine in a city traffic (speed up to 50 km/h), the engine works at a part load (bmep less than 25% of its maximal value) almost 50% of the operating time. In the case of traffic jam, engine works at part load even more than 68% of the operating time. However, it is generally known that the efficiency of an SI engine at part load is very low, within the range 10 - 20%, [1]. It is clear that improvement of the SI engine efficiency at part load would concerned about 50% (and even more) of its operating time. Simultaneously CO₂ emissions reduction would achieved by improvement of the SI engine efficiency at part load. These are reasons to continue the research on the investigation of

possibilities for the SI engine efficiency improvement at part load at author's laboratory.

The common method for achieving less than full power operation in a SI engine is reducing charge density via a throttle. However, a significant fuel economy penalty is associated with the pumping losses across the throttle valve, [2]. Therefore, one of the purposes of the mentioned research was to investigate the possibility to achieve the charge density reduction using: mixture heating, uncooled recirculated exhaust gas (EGR) delivery (into intake downstream of the throttle) and combination of these techniques in order to reduce throttling losses. This research was carried out in an experimental way using a conventional SI engine fuelled with natural gas at part load (bmep = 2 bar, about 23% of max. bmep) in a range of engine speeds 1400 - 3400 RPM. The tested engine was equipped with a

conventional ignition system and spark plugs. Ignition timing (ALIZ) was set always to MBT (maximal brake torque) point.

Based on results from experiments presented in work [3], the engine efficiency increasing through mixture heating was insignificant. Reasons for observed insignificant effect of mixture heating on the SI engine efficiency at part load probably are: increase in the indicated efficiency (due to combustion rate increasing) is eliminated by heat losses through the walls of combustion chamber. Moreover, the increase in mixture temperature during the combustion causes both an increase in degree of dissociation of combustion gases, see work [2], and decrease in polytropic index of thermal efficiency of the Otto cycle, see work [4].

A very small engine efficiency increasing (up to 1%) due to the uncooled EGR delivery was observed only in the range of low engine speeds (up to 2300 min^{-1} in this case) and just in the range of low EGR rates (at most 12% in this case). This is caused by the small decrease in the pumping work without any significant deterioration in the combustion rate. Combustion rate retardation (due to EGR delivery) can be compensated by a small change of the ignition timing. However, this is possible only in the range of a small engine speed. The combustion rate of the mixture with NO EGR delivery is smaller in the range of low engine speeds than in the range of higher engine speeds. Then the retardation in the combustion rate due to the same EGR rate is more significant in the range of higher engine speeds compared to the range of low engine speeds, [3].

Based on the measured data, an optimistic assumption about the compensation of decreased combustion rate (in consequence of EGR delivery) due to mixture heating was not feasible under higher EGR rate (about 20% EGR and more). It means that successful compensation of decreased combustion rate due to mixture heating is possible only in the range of low EGR rate, in which the decrease in combustion rate is not significant, [3].

So, based on experimental data, it is clear that the SI engine efficiency increase at part load using techniques of mixture heating and EGR delivery is very small. Other techniques are needed to use for improvement of the SI engine efficiency at part load.

One of the techniques may be Variable Valve Timing (VVT). A proper choice of valve timing can provide significant reductions in pumping losses of the SI engines at part load operation. Variable valve timing mainly influences charge exchange processes, then the engine volumetric efficiency. In some circumstances, variations of valve timing could modify the charge composition and therefore the flame development and propagation, [5].

An alternative load control strategy is to run the engine without throttling and to regulate the charge mass by Late Intake Valve Closing (LIVC). The intake valve is then kept open well into the compression stroke allowing part of the charge to be returned to the intake manifold. The engine

could then be set to be operating on an Atkinson type cycle since the expansion stroke is longer than the compression stroke. Engines in which the load is controlled by the late intake valve timing suffer a loss of effective compression ratio due to the reduction of the compression stroke. Hence the improvement in part load efficiency does not match the reduction in the pumping loss. However, it has been observed that if load control by LIVC is combined with Variable Compression Ratio (VCR), the total improvement in efficiency exceeds the sum of the two features, [6].

So called Otto-Atkinson cycle engine operates on the Otto cycle at full load, for good power density, and on the Atkinson cycle at part loads, for reduced fuel consumption. However, using the Atkinson cycle (LIVC) through reduced effective compression ratio, the mixture temperature in the end of compression is lower and this leads to poor combustion. In order to eliminate this unwanted effect, geometric compression ratio is increased. Increasing the geometric compression ratio will restore the lost effective compression ratio, so intake valve closing (IVC) can be retarded further to the point of running unthrottled at part load of engine, [7]. However, too high compression ratio is inconvenient in case of high or max. load of the engine because of knock limit. So it is clear that Otto-Atkinson cycle engine is necessary to equipped with mentioned VCR system, which is expensive and complex.

Another way in load control strategy is to run the engine without throttling and to regulate the charge mass by Early Intake Valve Closing (EIVC) - so called an extreme Miller cycle. Intake valve is closed before the bottom dead center (BDC), so effective expansion stroke is longer than effective compression stroke, [8]. The expansion and subsequent compression back to the volume at which the valve was closed is relatively cost free in terms of energy as both processes are nearly isentropic at such low temperatures. Thus the EIVC cycle is equivalent to an Atkinson cycle where the expansion stroke of the piston is greater than the required compression work. The cycle efficiency is thereby improved using EIVC. For maximum power output, an increasing the boost of the inlet charge is necessary to compensate for the reduced inlet duration, [9].

It is necessary to noted that Miller cycle is typically air-charged combustion cycle with intercooler. Goal of this cycle, introduced by Ralph Miller in 1947 ASME (LIVC), was decrease of NOx and particulate pollutants of diesel engines at maximal power output. However, necessary boost pressure is very high in case of EIVC compared to LIVC especially in high engine speeds. That is why the Miller cycle with LIVC concept is more useful for engines operating in wide speed range, [9], [10]. Alternatively, the Miller concept can be used in an naturally-aspirated configuration, as a means of achieving load control with higher efficiency, [9].

However, in-cylinder mixture temperature is decreased in the beginning of compression by applying mentioned cycles (Atkinson and Miller) compared to conventional (Otto) cycle. In case of constant value of compression ratio, the

temperature of in-cylinder charge at the beginning of combustion is also decreased and combustion is then slower (compared to a standard Otto cycle). This could negatively influence the indicated efficiency of unconventional cycles. In order to avoid this, increase of in-cylinder charge temperature has been provided due to mixture heating in the intake manifold of engine.

Because no test SI engine, equipped with Atkinson or Miller cycle was at disposal, the investigation of possibilities for the SI engine efficiency improvement at part load using EIVC and LIVC technique was carried out using simulations. A commercial 1-D code was utilized in order to calculate the in-cylinder pressure and the fuel economy improvement for both of above unconventional cycles with and without mixture heating.

MODEL SET-UP AND CALIBRATION

Computer simulations were carried out using GT-Suite, a one dimensional CFD code developed by Gamma Technologies for engine performance calculations, coupled with a predictive combustion model - spark-ignition turbulent flame model, see [11] a [12]. The main engine specifications are listed in Table 1.

Table 1. Main features of the test engine.

Bore	75 mm
Stroke	72 mm
Geometric CR	9.8
Number of cylinders	4 (in line)
chosen bmep / bmep_{max}	200 / 877 kPa (cca. 23% of bmep _{max})
Fuel	Natural gas (through central mixer)
Ignition	Conventional (one spark-plug for each cylinder)
Power control	Using throttle
Mixture heating	Heating air upstream of mixer using an electric heater

Calibration of both engine model and turbulent flame model was carried out using measured data of the SI engine described above (with the Otto cycle) in two various steady state operating points, as can be seen in Table 2. Ignition timing (ALIZ) data were used from experiments. ALIZ was set to value at MBT (maximal value of brake torque of test engine) in each operating point as mentioned in Introduction.

Chosen engine speed of 1700 RPM represents a range of lower engine speeds. Contrary, selected engine speed of 2900 RPM represents a range of higher engine speeds. Temperature of mixture, shown in Table 2, was measured in intake downstream of throttle during the experiments.

Table 2. Two measured operating points of the test engine for model calibration (the Otto cycle).

Speed [RPM]	Ignition timing (at MBT point)	Mixture temperature
1700	40° bTDC	29 °C (Without heating (W.H.))
2900	50° bTDC	29 °C (Without heating (W.H.))

One of the goals of the calibration was to achieve calculated patterns of in-cylinder pressure that are very close to the measured patterns (especially in low-pressure part of working cycle - for pumping work decrease investigation). Simultaneously, differences between measured and calculated fluxes of air and fuel (through the engine) should be less than 5%.

Moreover, goal of the calibration of predictive turbulent flame model (using template "EngCylCombSITurb") was to achieve patterns of normalized heat release (marked as Q_{nm}) that are very close to the measured patterns. In simulations, the spark timing was set on the base of measured data (ALIZ at MBT point), see Table 2, as mentioned. Calibration of turbulent flame model was carried out by settings of three parameters (upon recommendation in [11]):

- Turbulent Flame Speed Multiplier,
- Flame Kernel Growth Multiplier,
- Dilution Exponent Multiplier.

Final calculated in-cylinder pressure patterns using calibration are shown in Figures 1a and 1b (1700 RPM) and in Figures 2a and 2b (2900 RPM) in comparison of measured patterns. As can be seen, there was not achieved full agreement of calculated and measured patterns of in-cylinder pressure. Reasons for this could be e.g. some deviations in flow coefficients of both valves and intake ports in cylinder head and deviations in pressure fluctuations in intake and exhaust manifolds.

Calculated patterns of normalized heat release (Q_{nm}) using calibration in comparison of measured patterns are shown in Figure 3 (1700 RPM) and in Figure 4 (2900 RPM). However, there was not achieved full agreement of calculated and measured patterns of normalized heat release during calibration of the model. It is visible that achieved agreement of patterns was only in section of 30% to 70% of entire normalized heat release. Reasons for this could be e.g. wedge shape of combustion chamber in comparison of supposed shape of combustion chamber in model (spherical shape which center is shifted in direction of real wedge shape (no SLT data were at disposal), see [11].), another type of fuel (based on statement in work [12] the turbulent flame model was calibrated with fuel similar to gasoline) and also the part load of engine. Turbulent flame model in template of GT-

Power v. 6.2.0 was calibrated apparently for high/max. load of the SI engine.

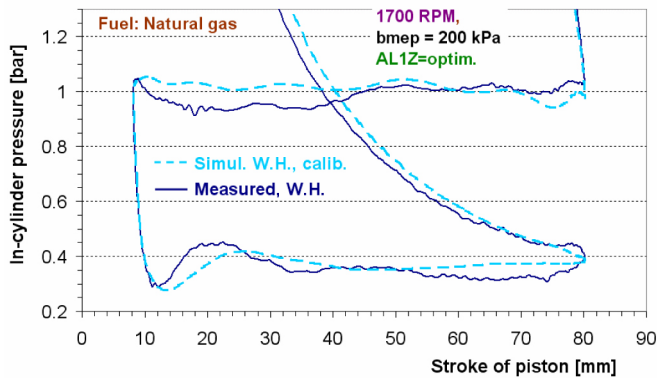


Figure 1a. Calculated in-cylinder pressure patterns using calibration (dashed line) in comparison of measured patterns (solid line) in the low-pressure part of working cycle at 1700 RPM.

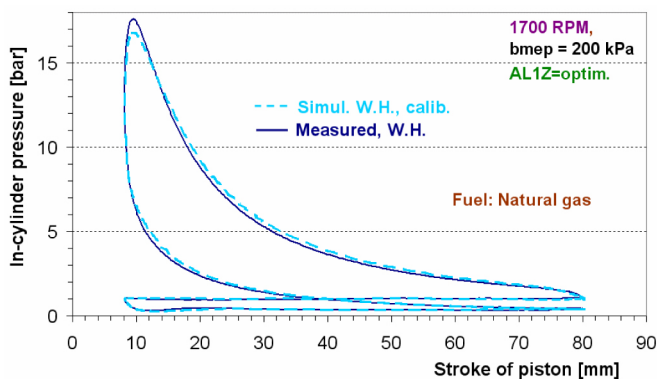


Figure 1b. Calculated in-cylinder pressure patterns using calibration (dashed line) in comparison of measured patterns (solid line) in the whole working cycle at 1700 RPM.

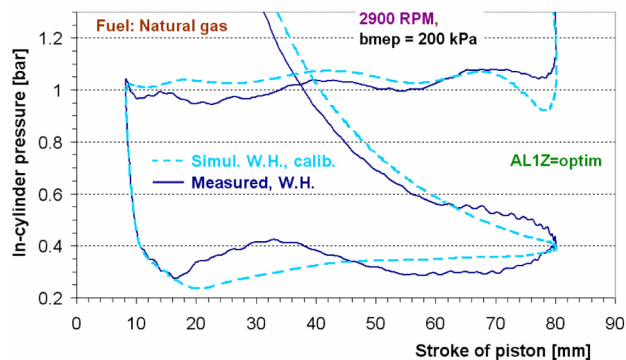


Figure 2a. Calculated in-cylinder pressure patterns using calibration (dashed line) in comparison of measured patterns (solid line) in the low-pressure part of working cycle at 2900 RPM.

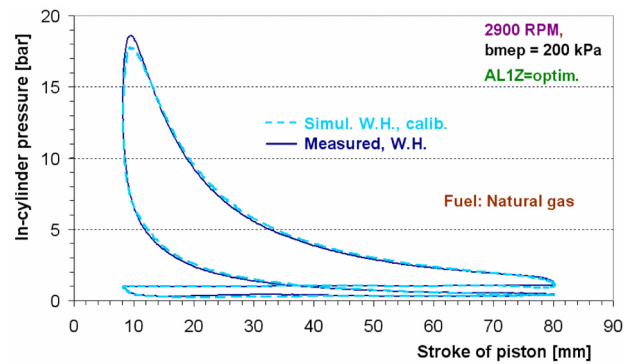


Figure 2b. Calculated in-cylinder pressure patterns using calibration (dashed line) in comparison of measured patterns (solid line) in the whole working cycle at 2900 RPM.

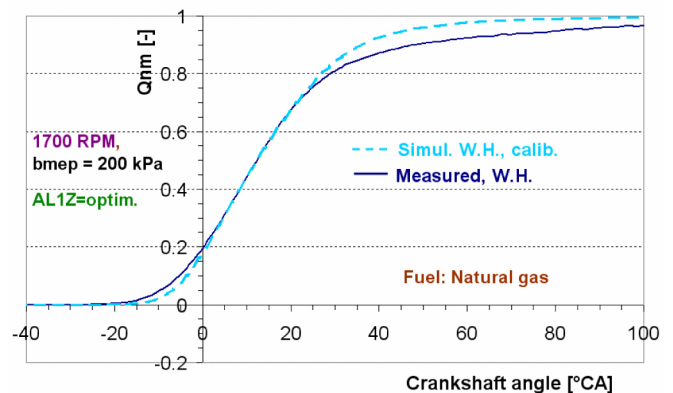


Figure 3. Calculated normalized heat release patterns using calibration (dashed line) in comparison of measured patterns (solid line) at 1700 RPM.

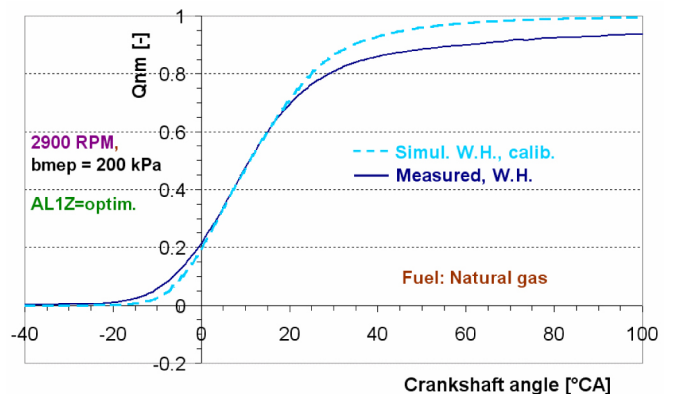


Figure 4. Calculated normalized heat release patterns using calibration (dashed line) in comparison of measured patterns (solid line) at 2900 RPM.