NO_x EMISSIONS WITH APPLICATION OF THE ATKINSON AND THE MILLER CYCLES AT PARTIAL LOAD WITH MIXTURE HEATING

LUBOMÍR MIKLÁNEK

Josef Božek Vehicle Centre for Sustainable Mobility, Czech Technical University in Prague, Czech Republic

Tel.: +420 224 351 855; E-mail: lubomir.miklanek@fs.cvut.cz

SHRNUTÍ

Obsah článku je zaměřen na výzkum emisí NO_x v aplikaci obou známých nekonvenčních cyklů, tj. modifikovaného Atkinsonova pracovního cyklu s pozdním zavíráním sacího ventilu (LIVC) a Millerova pracovního cyklu s extrémně předčasným zavíráním sacího ventilu (EIVC) v porovnání s Ottovým cyklem v částečném zatížení (se škrcením v sání) s ohřevem směsi v sání motoru.

Je známo, že aplikací Atkinsonova a Millerova cyklu dochází ke snížení teploty náplně ve válci na začátku komprese. V případě konstantního kompresního poměru je snížena také teplota náplně na začátku spalování. Spalování je tak pomalejší (v porovnání s Ottovým cyklem) což může negativně ovlivnit indikovanou účinnost cyklu. Tomuto lze zabránit např. ohřevem pracovní látky v sacím traktu motoru. Tím však může docházet k zvýšení emisí NO_x následkem zvýšení teploty ve válci během spalování.

Pro výpočet emisí NO_x během spalování se použil rozšířený Zeldovičův mechanismus jakožto součást použitého komerčního 1-D kódu. Pomocí simulací byl zjištěn určitý pokles emisí NO_x v aplikaci nekonvenčních pracovních cyklů v porovnání s Ottovým cyklem a to i s ohřevem směsi v sacím traktu.

KLÍČOVÁ SLOVA: ZVYŠOVÁNÍ ÚČINNOSTI ZÁŽEHOVÉHO MOTORU, NO_x, ATKINSONŮV CYKLUS, MILLERŮV CYKLUS, OHŘEV SMĚSI, NÍZKÉ ZATÍŽENÍ MOTORU

ABSTRACT

The objective of this work is an investigation of NO_x emissions with the application of both 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 comparison with the Otto cycle at partial load (throttled) with heating of working fluid (mixture) in the intake.

It is known that application of the Atkinson and the Miller cycles causes a decrease in the in-cylinder charge temperature before the compression stroke. However, in-cylinder charge temperature at the beginning of combustion is also decreased in the case of a constant value of geometric compression ratio. Combustion is then slower (compared to a standard Otto cycle) and indicated efficiency could be negatively influenced. To avoid this, the mixture can be heated in the intake manifold of the engine.

However, this can lead to an increase in NO_x emissions due to the higher charge temperature during combustion.

A commercial 1-D code was used in order to calculate NO_x emissions during combustion using an extended Zeldovich mechanism.

Some NO_x content reduction in exhaust gas was calculated due to application of the unconventional cycles compared to the Otto cycle even with mixture heating.

KEYWORDS: FUEL ECONOMY IMPROVEMENT, SI ENGINE, NO_x, ATKINSON CYCLE, MILLER CYCLE, MIXTURE HEATING, PARTIAL ENGINE LOAD

1. INTRODUCTION

Based on results from measurements with a passenger car equipped with a spark ignition (SI) engine in city traffic (speed up to 50 km/h), the engine works at partial load (bmep less than 25% of its maximum value) almost 50% of the operating time. In the event of a traffic jam, the engine partial load running increases to more than 68% of the operating time, see [1].

Unfortunately, as is generally known, the efficiency of an SI engine at partial load is very low, within the range 10–20%, [2], [3]. Thus, improvement of the SI engine efficiency at partial load would relate to about 50% (or even more) of its operating time. Simultaneously, a reduction in CO₂ emissions would also be achieved. These are reasons for continuing the research into SI engine fuel economy improvement at partial load at the author's laboratory.

The common method for achieving less than full power operation of 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, [4]. Therefore, one of the goals of the mentioned research was to investigate the possibility of achieving a charge density reduction using: **mixture heating**, **uncooled recirculated exhaust gas (EGR) delivery** (into the intake downstream of the throttle), and a **combination of these techniques** in order to reduce throttling losses (with the Otto cycle). This research was carried out in an experimental way using a conventional SI engine fuelled with natural gas at partial load (bmep = 2 bar, about 23% of max. bmep) at a range of engine speeds (1400 – 3400 RPM). The tested engine was equipped with a conventional ignition system and spark plugs. Ignition timing (AL1Z) was always set to the MBT (Maximum Brake Torque) point. Setting of the lambda-control loop was unchanged ($\lambda = 1$). Unfortunately, based on experimental data, the SI engine fuel economy improvement at partial load using techniques of mixture heating and EGR delivery is almost insignificant. Other techniques are needed for the SI engine fuel economy improvement at partial load, see details in work [1].

An alternative load control strategy at partial load is to run the engine without throttling and to regulate the charge mass by Late Intake Valve Closing (LIVC) – known as the Atkinson cycle, or by Early Intake Valve Closing (EIVC) – termed an extreme Miller cycle. The effective compression stroke is shorter than the expansion stroke in these applications. Thus there is a potential for cycle efficiency improvement, [5], [6], [7].

It is known that a reduction in compression stroke brings a reduction in effective compression ratio, which could eliminate the positive

effect of the LIVC or EIVC techniques. A variable compression ratio technique could be applied to eliminate the reduction in compression ratio. However, this is expensive and complex.

Therefore, the mixture heating technique in the intake manifold is investigated at the author's laboratory in order to eliminate the negative influence of LIVC or EIVC on the compression ratio.

2. INFLUENCE OF LIVC AND EIVC TECHNIQUES ON CHARGE TEMPERATURE

It is known that in-cylinder mixture temperature at the beginning of compression is decreased by application of the mentioned cycles (Atkinson and Miller) compared to the conventional (Otto) cycle. Reduction of the in-cylinder charge temperature at the beginning of compression due to LIVC (Atkinson cycle) is shown in Figure 1, and due to EIVC (Miller cycle) is shown in Figure 2.

In the case of maintaining a constant compression ratio value, the temperature of the in-cylinder charge at the beginning of combustion is also reduced, see Figure 3. Combustion is then slower (compared to a standard Otto cycle). This could negatively influence the indicated efficiency of the unconventional cycles. In order to avoid this, there is the possibility of increasing the in-cylinder charge temperature due to mixture heating in the engine intake manifold.

However, NO_x emissions may increase as a consequence of the higher charge temperature during combustion. Of course, there is the possibility of eliminating the increase in NO_x using an appropriate after treatment. Nevertheless, the aim of this research is investigation of the potential of unconventional cycles for reduction of increased NO_x .

FIGURE 1: Reduction of the in-cylinder charge temperature at the beginning of compression due to the LIVC technique (Atkinson cycle).

OBRÁZEK 1: Snížení teploty náplně ve válci na začátku komprese použitím techniky LIVC (Atkinsonův cyklus).

FIGURE 2: Reduction of the in-cylinder charge temperature at the beginning of compression due to the EIVC technique (Miller cycle).

OBRÁZEK 2: Snížení teploty náplně ve válci na začátku komprese použitím techniky EIVC (Millerův cyklus).

2. MODEL SET-UP AND CALIBRATION

Because at the author's laboratory there was no test SI engine equipped with the Atkinson or Miller cycle available, the investigation for the SI engine fuel economy improvement at partial load using LIVC and EIVC techniques was carried out using simulations. Computer simulations were performed using GT-Power, a one dimensional CFD code developed by Gamma Technologies for engine performance calculations, coupled with a predictive combustion model – spark-ignition turbulent flame model [8]. An extended Zeldovich mechanism, (as a part of CFD code), see details in work [9], was applied for NO_x emissions calculation during combustion with and without mixture heating.

The main engine model specifications are listed in Table 1.

Calibration of the engine model, the turbulent flame model and also the NO_x model was carried out using measured data from the SI engine, shown in Figure 4 (using the Otto cycle, $\text{EGR}=0\%$), with the same main specification as described in Table 1. Calibration was carried out at two different steady state operating points, as shown in Table 2.

Ignition timing (AL1Z) data were also used from experiments. AL1Z was set at the MBT (maximum brake torque) point for each engine operating speed.

The selected engine speed of 1700 RPM represents a range of lower engine speeds. On the other hand, the selected engine speed of 2900 RPM represents a range of higher engine speeds.

One of the goals of the calibration was to achieve the smallest differences possible between calculated and measured patterns of the in-cylinder pressure. At the same time, differences between measured and calculated fluxes of air and fuel (through the engine) should be less than 5%. More details can be found in work [1].

Moreover, the goal of the predictive turbulent flame model calibration (using template "EngCylCombSITurb") was to achieve the smallest differences possible between calculated and measured patterns of the normalized heat release (marked as Q_{nm}).

Finally, the goal of the NO_x model calibration (using template "EngCyl NO_x ") was to achieve NO_x emissions very similar to the measured values.

It was observed that it is necessary to calibrate the model at each engine speed operating point (without mixture heating) to achieve the mentioned calibration goals.

2.1. DESCRIPTION OF THE TEST BENCH FOR CALIBRATION DATA (OTTO CYCLE)

To obtain calibration data, all experiments were carried out on the engine test bench at the author's laboratory. Engine test bench layout is shown in Figure 4.

The examined SI engine used a four-stroke cycle, with four-cylinders, naturally-aspirated ($\varnothing 75.5 \times 72$ mm, see Table 1), natural gas-fueled (NG), equipped with a closed-loop lambda-control system

FIGURE 3: Reduction of the in-cylinder charge temperature during combustion under LIVC and EIVC techniques in the case of constant compression ratio.

OBRÁZEK 3: Snížení teploty náplně ve válci během spalování s použitím technik LIVC a EIVC v případě konstantního kompresního poměru.

FIGURE 4: Layout of the engine test bench.

OBRÁZEK 4: Schéma uspořádání zkušebního stanoviště.

Bore	75 mm
Stroke	72 mm
Geometric CR	9.8
Number of cylinders	4 (in line)
Chosen bmep / bmep _{max}	200 / 877 kPa (approx. 23% of bmep _{max})
Fuel	Natural gas (through central mixer)
Ignition	Using a spark plug
Power control	Using a throttle in intake
Mixture heating	Heating sucked air (desired mixture temperature: 120° downstream of throttle)
Combustion model	SI Turbulent Flame Combustion Model
Heat transfer model	Flow
NO_x model	Extended Zeldovich mechanism

TABLE 1: Main engine model specifications.

TABULKA 1: Hlavní specifikace modelu motoru.

TABLE 2: Two selected operating points of the test engine for model calibration (the Otto cycle).

TABULKA 2: Dva vybrané provozní body testovacího motoru pro kalibraci modelu (Ottův cyklus).

Speed [RPM]	Ignition timing (at MBT point)	Mixture temperature (downstream of throttle)
1700	40° bTDC	29 °C (No heating (NH))
2900	50° bTDC	29 °C (No heating (NH))

for stoichiometric conception, three-way catalyst placed in the exhaust manifold, conventional ignition system and spark plugs. For more details see work [1].

The engine test bench is equipped with a DAQ system for acquisition of slowly changing time-based quantities and crank-angle based in-cylinder pressure acquisition. In-cylinder pressure was indicated in the first cylinder from the pulley. Special software has been developed (using the TestPoint development system [10]) for recording both integral data and in-cylinder pressure patterns (usually over 100 working cycles) during the engine test bench operation [11], as well as for the evaluation of such data off-line [12].

A purpose-designed heater was placed into the intake manifold to increase the aspirated air temperature upstream of the mixer. To keep the average value of the mixture temperature downstream of the throttle within a range of 1 °C, a feedback was used for heating power control of the applied heater, as shown in Figure 1. For more details see work [1].

2.2. NO_x MODEL CALIBRATION AND VERIFICATION (OTTO CYCLE)

Nitric oxide (NO) and nitric dioxide (NO₂) are grouped together as NO_x. However, NO is the predominant oxide of nitrogen produced inside the combustion chamber. The source of NO is the oxidation of the atmospheric nitrogen. Moreover, if the fuel contains significant amounts of nitrogen, this may be an additional source of NO, see [9]. Because the applied fuel (natural gas) contains a negligible amount of nitrogen, it is assumed that only atmospheric nitrogen oxidation will be a source of NO.

All three principal reactions of the extended Zeldovich mechanism listed below are reversible. Zeldovich was the first to suggest the importance of reactions (1) and (2). The last reaction (3) of atomic nitrogen with the hydroxyl radical, OH, was added by Lavoie, see work [9].

The NO calculation is based on the extended Zeldovich mechanism. Rate constants k_1 , k_2 and k_3 , which are used in CFD code to calculate the reaction rates of the three above-mentioned reactions, are given as:

$$k_1 = F_1 \cdot 7.60 \cdot 10^{10} \cdot \text{EXP} \left(\frac{-38000 \cdot A_1}{T_b} \right) \quad (4)$$

$$k_2 = F_2 \cdot 6.40 \cdot 10^6 \cdot T_b \cdot \text{EXP} \left(\frac{-3150 \cdot A_2}{T_b} \right) \quad (5)$$

$$k_3 = F_3 \cdot 4.10 \cdot 10^{10} \quad (6)$$

The above rate constants are forward rate constants. Based on work [9], these constants have been measured in numerous experimental studies. Reverse rate constants for the three reactions above are published in work [9].

The NO_x model can be calibrated using the multipliers F_1 , F_2 , F_3 , A_1 and A_2 . As observed during the calibration, the best option appeared to be the following: multiplier A_1 was tuned both in terms of the smallest differences between calculated and measured NO_x emissions with mixture heating, and also with respect to acceptable time-duration of tuning. Values of other parameters were kept at the default values of 1. There are of course more possibilities for tuning; however, the other tuning options may be more time-consuming. Comparison of the measured and calculated values of NO_x using the calibrated model is shown in Figure 5 (marked Calibration). There is a good agreement between the measured and calculated values achieved at both engine speeds. Differences between calculated and measured values are within the range of 1%.

Moreover, the calibrated NO_x model was verified using measured NO_x data under conditions of mixture heating (120°C downstream of throttle). Results from verification of the NO_x model at both engine speeds are shown in Figure 5. As can be seen, calculated NO_x values are higher at both engine speeds compared to the measured data. Differences between calculated and measured values are about 16% (1700 RPM) and 24% (2900 RPM). Reasons for these differences are shown in Figure 6, where patterns of measured (using the in-cylinder pressure acquisition) and calculated in-cylinder temperature during combustion are depicted. Calculated in-cylinder temperature is slightly higher during combustion compared to the measured patterns. The time-duration of high charge temperature (with NO_x formation) is thus longer compared to the measured data. These differences confirm the high sensitivity of the NO_x model to the in-cylinder temperature during combustion.

It should be mentioned that it was unfortunately not possible to obtain better agreement between measured and calculated in-cylinder temperature patterns using the predictive turbulent

flame combustion model in GT-Power, ver. 6.2.0. There are several reasons for this, some of which could be: properties of the turbulent flame combustion model, simplified shape of the calculated combustion chamber in the model and so on. The calibrated NO_x model was applied for NO_x emissions calculations with the Atkinson and also the Miller cycle with (HE) and without (NH) mixture heating.

2.3. MIXTURE HEATING FEEDBACK CONTROL APPLIED TO THE ENGINE MODEL

The selected method for mixture heating in the engine model and also for control of mixture temperature is presented in Figure 7. There are two air sources placed at the beginning of the intake manifold upstream of an air filter. One air source contains cold air, the other hot air. Cold and hot air mass flows are controlled by throttles based on the feedback – mixture temperature downstream of the throttle in the intake of the engine model. More details are given in work [1].

3. THE ATKINSON CYCLE

It was originally invented by James Atkinson in 1882 with aim of improving the Otto cycle efficiency by having an increased expansion stroke of the piston compared to the compression stroke. The length of stroke was changed using a unique crankshaft mechanism, for more details see work [1].

However, recently a “modern” Atkinson cycle has been applied in SI engines (with the Otto cycle), which is characterized by a shorter effective compression stroke than expansion stroke due to late intake valve closing (LIVC), [5].

In the case of a constant compression ratio, the effective compression ratio is thus reduced. As a consequence, the in-cylinder charge temperature is also reduced, as already mentioned.

In order to investigate the influence of the Atkinson cycle on NO_x emissions with and without mixture heating, simulations with the above-mentioned engine model were carried out. Loading of the engine remained unchanged, ($\text{bmep}=200 \text{ kPa}$), as listed in Table 1. Simulations were carried out at two engine speed operating points: 1700 and 2900 RPM, as in the simulations above.

Intake valve closing (IVC) retardation was chosen with the addition of 15°CA in each variant. Thus, the selected LIVC variants were: 75, 90, 105 and 120°CA after BDC. To achieve appropriate intake valve lift, an original valve lift curve was extended at the highest point, as shown in Figure 8.

Because heat release (Q_{nm}) patterns under Atkinson cycle conditions were not available, calculated Q_{nm} patterns (with the Otto cycle) were chosen as reference patterns. The chosen strategy for achieving reference Q_{nm} patterns under unconventional cycle conditions $\rightarrow$ only ignition timing (AL12) could be changed, more details are given in [1].

FIGURE 5: Measured and calculated values of NO_x using the calibrated model without (NH) and with (HE) mixture heating.

OBRÁZEK 5: Naměřené a vypočítané hodnoty NO_x pomocí zkalibrovaného modelu bez ohřevu (NH) a s ohřevem (HE) směsi.

FIGURE 6: Experimentally observed and calculated patterns of in-cylinder temperature without (NH) and with (HE) mixture heating before and during combustion.

OBRÁZEK 6: Experimentálně zjištěné a vypočítané průběhy středních teplot ve válci bez ohřevu (NH.) a s ohřevem (HE) směsi před a v průběhu spalování.

FIGURE 7: Detail of mixture heating feedback control applied in the engine model.

OBRÁZEK 7: Detail zpětnovazebního řízení teploty směsi aplikovaný v modelu motoru.

Extreme early IVC for each calculated variant was chosen: 60, 75 and 90 °CA before BDC as shown in Figure 11. To achieve appropriate intake valve lift, some parts of the original valve lift curve (of the test engine) were applied in the engine model. Both start and end parts of the original valve lift curve were applied. The peak of the lift curve was properly rounded, as shown in Figure 11. The same strategy was chosen for achieving reference Q_{nm} patterns under unconventional cycle conditions, as in the case of the Atkinson cycle, see above.

4.1. CALCULATED PATTERNS OF THE IN-CYLINDER TEMPERATURE UNDER EIVC CONDITIONS

5. CALCULATED CHANGES IN NO_x EMISSIONS

Calculated NO (designated NO_x) emissions due to application of the Atkinson and the Miller cycle in comparison with the Otto cycle are presented in Figure 14. Relative changes of NO_x compared to the Otto cycle are presented in Figure 15.

OBRÁZEK 11: Původní zdvihová křivka sacího ventilu testovacího motoru a modifikované zdvihové křivky pro počítané Millerovy cykly.

OBRÁZEK 12: Porovnání vypočítaných průběhů středních teplot ve válci u Ottova a Millerova cyklu (Miller 60) při 1700 min⁻¹, před a v průběhu spalování, s a bez (NH) ohřevu směsi.

OBRÁZEK 13: Porovnání vypočítaných průběhů středních teplot ve válci u Ottova a Millerova cyklu (Miller 60) při 2900 min⁻¹, před a v průběhu spalování, s a bez (NH) ohřevu směsi.

The assumption regarding a reduction in NO_x emissions (molar fractions) due to application of both unconventional cycles has been confirmed, as shown in Figure 14 and Figure 15. This is due to both a reduction in the maximum values of T_{cyl} and the shortened time-duration of high T_{cyl} (with NO_x formation) compared to the Otto cycle.

Moreover, the efficiency changes can be seen in Figure 16. As shown, cycle efficiency increases with the unconventional cycle, and in particular the Miller cycle compared to the Otto cycle. The reason for this is a reduction in pumping work due to the LIVC and EIVC techniques, see [1].

Moreover, the brake specific NO_x production is also reduced due to application of unconventional cycles compared to the Otto cycle, see Figure 16. This is caused by a reduction in the required amount of working fluid for the unconventional cycles.

As can be seen in Figure 16, there is the possibility of obtaining values of brake specific NO_x production due to application of the Miller cycle (even with mixture heating) very close to the values of the Otto cycle without mixture heating, which is positive.

6. CONCLUSION

The NO_x emissions were calculated in applications of the Otto cycle and both of the unconventional cycles (Atkinson and Miller) at partial load using the extended Zeldovich mechanism, as a part of applied 1-D CFD code. As observed, molar fractions of NO_x emissions are decreased with the application of both unconventional cycles even with intake mixture heating at 120 °C, compared to the Otto cycle (decrease in NO_x is about 13% and 30% for the Atkinson and the Miller cycles, respectively). This is a consequence of the in-cylinder charge temperature decrease at the end of compression (and during combustion) due to application of the unconventional cycles. The maximum value of the in-cylinder temperature is thus lower, even with the mixture heating, compared to the Otto cycle. At the same time, the time-duration of high charge temperature (with NO_x formation) is shorter compared to the Otto cycle.

The Miller cycle (EIVC) appears to be a more suitable cycle for NO_x reduction than the Atkinson cycle (LIVC) in this research. Reasons for this could be the lower amount of in-cylinder charge (higher efficiency) in comparison with the Atkinson cycle. Maximum value of the in-cylinder pressure is thereby also lower.

Moreover, the brake specific NO_x production is also reduced due to application of the unconventional cycles compared to the Otto cycle. This is caused by a reduction in the necessary amount of working fluid for the unconventional cycles.

There is the possibility of obtaining values of brake specific NO_x production due to application of the Miller cycle (with mixture heating) that are very close to the values achieved with the Otto cycle (without mixture heating), which is very positive.

FIGURE 14: Calculated NO_x emissions (molar fractions) due to application of the Atkinson and the Miller cycle in comparison with the Otto cycle with (HE) and without (NH) mixture heating.

OBRAZÉK 14: Vypočítané emise NO_x (molární zlomky) v aplikaci Atkinsonova a Millerova cyklu v porovnání s Ottovým cyklem s ohřevem (HE) a bez ohřevu (NH) směsi.

FIGURE 15: Relative changes of calculated NO_x emissions due to application of the Atkinson and the Miller cycle in comparison with the Otto cycle with (HE) and without (NH) mixture heating.

OBRAZÉK 15: Relativní změny vypočítaných emisí NO_x v aplikaci Atkinsonova a Millerova cyklu v porovnání s Ottovým cyklem s ohřevem (HE) a bez ohřevu (NH) směsi.

FIGURE 16: Calculated changes of brake specific NO_x production due to application of unconventional cycles in comparison with the Otto cycle with (HE) and without (NH) mixture heating.

OBRAZÉK 16: Vypočítané hodnoty měrné produkce emisí NO_x v aplikaci Atkinsonova a Millerova cyklu v porovnání s Ottovým cyklem s ohřevem (HE) a bez ohřevu (NH) směsi.

It must be noted that all the experiments were carried out under both the following conditions: optimum value of ignition timing and unchanged setting of lambda-control loop ($\lambda = 1$).

Temperature of the mixture was increased up to 120 °C downstream of the throttle of the engine during the simulations and experiments. The value of 120 °C was chosen based on known maximum allowable value for mixture temperature in the intake manifold of modern naturally aspirated SI engines.

ACKNOWLEDGEMENT

This research has been realized using the support of EU Regional Development Fund in OP R&D for Innovations (OP VaVpl) and Ministry for Education, Czech Republic, project #CZ.1.05/2.1.00/03.0125 Acquisition of Technology for Vehicle Center of Sustainable Mobility. This support is gratefully acknowledged.

This research has been realized using the support of Technological Agency, Czech Republic, programme Centres of Competence, project # TE01020020 Josef Božek Competence Centre for Automotive Industry. This support is also gratefully acknowledged.

LIST OF NOTATIONS AND ABBREVIATIONS

λ	Air excess coefficient [–]
A_1	N_2 oxidation activation energy multiplier
A_2	N oxidation activation energy multiplier
AL1Z	Ignition timing before the TDC [°bTDC]
BDC	Bottom Dead Center, Czech → DU
bmp	Brake mean effective pressure [Pa], Czech → p_e
bsfc	Brake Specific Fuel Consumption [g/kWh]
bsNO _x	Brake Specific NO _x production [g/kWh]
CA	Crankshaft Angle [°], Czech → OKH
CO ₂	Carbon dioxide
CR	Compression Ratio
DAQ	Data Acquisition system
ECU	Electronic Control Unit
EGR	Exhaust gas recirculation
EIVC	Early Intake Valve Closing
F_1	N_2 oxidation rate multiplier
F_2	N oxidation rate multiplier
F_3	OH reduction rate multiplier
HE	Heating the mixture
ICE	Internal combustion engine
IVC	Intake Valve Closing
LIVC	Late Intake Valve Closing
MBT	Maximal Brake Torque
Mt	Engine torque [Nm]
NG	Natural gas
NH	No heating the mixture
NO	Nitric oxide

NO ₂	Nitric dioxide
NO _x	Mono-nitrogen oxides NO and NO ₂
OH	Hydroxyl radical
optim.	optimal
pCyl	In-cylinder pressure [bar]
Qnm	Normalized heat release [–]
RPM	Revolutions Per Minute [min ⁻¹]
SI	Spark-ignition
T_b	Calculated burned sub-zone temperature [K]
Tcyl	In-cylinder temperature [K]
TDC	Top Dead Center, Czech → HU

REFERENCES

- [1] Miklánek L. (2011). [Tools for Optimization of SI engine at Part Load](#) (in Czech). Dissertation Theses, CTU in Prague.
- [2] Kutlar A. O., Arslan H., Calik T., A. (2005). [Methods to Improve Efficiency of Four Stroke Spark Ignition Engines at part Load](#), Elsevier Ltd.
- [3] Macek J. (2000). [Combustion Engines I](#) (in Czech). Czech Technical University in Prague, 2000, ISBN 80-01-02085-1.
- [4] Brehob D. D., Amlee D. R. (1991). [Effects of Inlet Air Heating and EGR on Thermal Efficiency of a SI Engine at Part Load](#), SAE Paper 901713, SAE Int. Warrendale.
- [5] Blakey C. S., Saunders J. R., et al. (1991). [A Design and Experimental study of an Otto Atkinson Cycle Engine Using Late Intake Valve Closing](#), SAE Paper 910451, SAE Int. Warrendale.
- [6] Goto T., Hatamura K., et al. (1994). [Development of V6 Miller Cycle Gasoline Engine](#), SAE Paper 940198, SAE Int. Warrendale.
- [7] Anderson K. M., Assanis N. D., Filipi S. Z. (1998). [First and Second Law Analyses of a Naturally-Aspirated, Miller Cycle, SI Engine with Intake Valve Closure](#), SAE Paper 980889, SAE Int. Warrendale.
- [8] Gamma Technologies Inc. (2006). GT-POWER, User's Manual And Tutorial. GT-Suite TM version 6.2, Westmont, IL, USA.
- [9] Heywood J.B. (1998). [I.C.E. Fundamentals](#), McGraw Hill. ISBN 0-07-028637-X.
- [10] Capital Equipment Corp. (2003). TestPoint Version 5.0. Professional Development System.
- [11] Takáts M. (2003). ITI-ONL – software for acquisition of in-cylinder pressure pattern, (Josef Božek Research Center Code Library, CTU Prague).
- [12] Takáts M. (2009). INTEC – software for evaluation of in-cylinder pressure record, (Josef Božek Research Center Code Library, CTU Prague).