PREDICTION OF THE NET INDICATED EFFICIENCY AND PUMPING LOSSES OF A THROTTLED SI ENGINE AT PART LOAD

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SHRNUTÍ

Tato studie popisuje vliv změny indikované účinnosti vysokotlaké části pracovního cyklu (η_{iHP}) na indikovanou účinnost celého cyklu (η_i) a pumpovních ztrát (pro udržení konstantní hodnoty středního efektivního tlaku) pro případ zážehového motoru škrceného v sacím traktu v nízkém zatížení. Za účelem zjištění těchto závislostí byl sestaven jednoduchý matematický model. Model umožňuje počítat vztahy mezi indikovanou účinností vysokotlaké části pracovního cyklu, indikovanou účinností celého cyklu a také střední hodnotou tlaku v průběhu sacího zdvihu pístu (p_s). Navržený model byl kalibrován a následně verifikován pomocí experimentálně pořízených dat.

Experimentální práce byly provedeny na testovacím zážehovém, čtyřválcovém motoru s pohonem na zemní plyn, pracujícím se stechiometrickou koncepcí a vybaveným také třícestným katalyzátorem. Cílem experimentů bylo navíc najít řešení pro další zvýšení účinnosti vysokotlaké části v porovnání s optimálním konvenčním nastavením motoru. I když samozřejmě existuje jednoduchý způsob pro změnu (snížení) hodnoty indikované účinnosti vysokotlaké části pracovního cyklu pouhým rozladěním optimální hodnoty předstihu zážehu, tato varianta nebyla cílem výzkumu autorů. Z tohoto důvodu byla pro tento účel ve finále zvolena varianta s aplikací speciální zapalovací svíčky do každého válce motoru.

Jak ze simulací tak z experimentů bylo zjištěno, že se zvýšením indikované účinnosti vysokotlaké části pracovního cyklu dochází ke zvýšení jak indikované účinnosti celého cyklu tak také pumpovních ztrát a obráceně, což se předpokládalo.

KLÍČOVÁ SLOVA: ZVÝŠOVÁNÍ ÚČINNOSTI ZÁŽEHOVÉHO MOTORU, ZAPALOVACÍ SVÍČKA, NÍZKÉ ZATÍŽENÍ MOTORU

ABSTRACT

This study describes the influence of a change in gross indicated efficiency (η_{iHP}) on both net indicated efficiency (η_i) and pumping losses (to obtain a constant value of b.m.e.p.) for a throttled spark ignition (SI) engine at part load. A simple mathematical model was developed to understand these interactions. Model is able to describe the relationships between the gross indicated efficiency, net indicated efficiency and also average value of inlet manifold pressure during intake (p_s). Proposed mathematical model was calibrated and also verified using experimental data.

Experiments were carried out on a four-cylinder SI engine, natural gas fueled, equipped with a three-way catalyst with stoichiometric concept. Moreover, a further goal of the experiments was to find a solution for increasing gross indicated efficiency in comparison with a conventional optimal setting of the engine. Nevertheless, there is a simple option of changing (decrease) the gross indicated efficiency just due to detuning of the optimal settings of ignition timing of the engine. However, this simple solution was not the authors' goal. For this reason a variant with special spark plug application was chosen as the best choice for this purpose.

Experiments indicated that both the pumping losses and the net indicated efficiency increase with improvement in the gross indicated efficiency and vice versa, as expected.

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KEYWORDS: FUEL ECONOMY IMPROVEMENT, SI ENGINE, SPARK PLUG, PART ENGINE LOAD

1. INTRODUCTION

The common method for achieving reduced power operation of an SI engine is reducing charge density via a throttle. However, a significant fuel economy penalty is associated with the pumping losses [1], [2].

Moreover, poor combustion quality, i.e. lower combustion rate and cycle-to-cycle variations, additionally contribute to a decrease in the efficiency of an SI engine at part load, [3].

In order to reduce throttling losses and improve SI engine efficiency at part load, an investigation was carried out at the authors' laboratory with the aim of investigating the potential for charge density reduction, see [4]. Charge density reduction was achieved by mixture heating, by uncooled recirculation of exhaust gas (EGR) (into an intake system downstream of the throttle), and by a combination of these two techniques. This research was carried out using both experiments and 1-D simulations, see [5].

The increased mixture temperature may accelerate flame velocity. Unfortunately, based on experimental data, the improvement in SI engine fuel economy at part load due to mixture heating and EGR delivery was found to be almost negligible. Other techniques need to be employed for SI engine fuel economy improvement at part load, see details in works [5] or [6].

An alternative load control strategy at part load is to run the engine without throttling controlling the charge mass utilizing Late Intake Valve Closing (LIVC) – the modern Atkinson cycle, or by Early Intake Valve Closing (EIVC) – the extreme Miller cycle. Moreover, the effective compression stroke is shorter than the expansion stroke in these applications, which creates further potential for cycle efficiency improvement, [7], [8], [9]. However, pumping losses are decreased together with the flame velocity due to low temperature at the end of the compression stroke in the case of constant compression ratio. Nevertheless, 1-D simulation has shown a certain SI engine efficiency improvement at part load using the application of those unconventional cycles – more details in [5] or [6].

The above-mentioned methods influence the pumping losses, but they also give rise to changes in indicated efficiency of the high-pressure part of the working cycle (gross). Therefore, both factors working together change the indicated efficiency of the whole working cycle (net). Any increase in gross cycle efficiency is balanced by increased throttling at a part load of a standard four-stroke engine with fixed

b.m.e.p. The fresh mixture density is proportional to absolute inlet manifold pressure, but pumping work depends on the difference between exhaust and inlet manifold pressures. This is why the change of net cycle efficiency cannot easily be estimated, even if the gross cycle efficiency change is known. To understand this issue, a simple mathematical model was developed, calibrated and verified using experimental data, as follows.

2. MATHEMATICAL MODEL

The aim of this chapter is to describe the chosen approach to create a mathematical model which can accomplish the following requirements:

- the model should be as simple as possible (with short computational time and low requirements for PC performance),
- it should also be able to describe the relationships between the gross indicated efficiency (η_{iHP}), net indicated efficiency (η_i) and also average value of inlet manifold pressure during intake (p_s).

The negative pumping work in the case of throttling, assuming a rectangular shape of charge-exchange p-V diagram, can be expressed by [1]:

$$W_n = K (p_s - p_v) V_{z1} \quad (1)$$

Then the indicated efficiency of the low-pressure part of the cycle (losses) can be expressed:

$$\eta_{iLP} = \frac{W_n}{Q_d} = \frac{K (p_s - p_v) V_{z1}}{Q_d} = \frac{K (p_s - p_v) V_{z1}}{H_u m_p} \quad (2)$$

Assuming that the net indicated efficiency is the sum of the gross indicated efficiency and the indicated efficiency of the low-pressure part as follows:

$$\eta_i = \eta_{iHP} + \eta_{iLP} \quad (3)$$

Then it is possible to calculate the net indicated efficiency of the cycle using equations (3) and (2) and the well known relationship for the net indicated work (which includes combustion efficiency), as follows:

$$\begin{aligned}
W_i &= p_i V_{z1} = \eta_i H_u m_p = \left(\eta_{iHP} + \frac{K(p_s - p_v) V_{z1}}{H_u m_p} \right) H_u m_p \\
\eta_i &= \frac{p_i V_{z1}}{H_u m_p} = \eta_{iHP} + \frac{K(p_s - p_v) \eta_i}{p_i} \Rightarrow \eta_i \left(1 - \frac{K(p_s - p_v)}{p_i} \right) = \eta_{iHP} \\
\eta_i &= \frac{\eta_{iHP}}{1 - \frac{K(p_s - p_v)}{p_i}}
\end{aligned} \tag{4}$$

It is possible to write an equation for pressure p_s , using equation (4):

$$1 + \frac{K p_v}{p_i} - \frac{\eta_{iHP}}{\eta_i} = \frac{K p_s}{p_i} \quad p_s = \frac{p_i}{K} \left(1 + \frac{K p_v}{p_i} - \frac{\eta_{iHP}}{\eta_i} \right) \tag{5}$$

The relationship for pressure p_s can be obtained as described above. However, it is necessary to find an equation for the net indicated efficiency. To do this, it is possible to use well known relationships for calculation of fuel mass per cycle (m_p), as follows (assuming that the in-cylinder mixture behaves as an ideal gas):

$$m_p = \frac{m_{trapped} \frac{\sigma_{O2trapped}}{0.23}}{\lambda L_t} = \frac{\lambda_{charging} V_{z1} \frac{p_s}{r T_s} \frac{\sigma_{O2trapped}}{0.23}}{\lambda L_t} = \frac{p_i V_{z1}}{H_u \eta_i} \tag{6}$$

Then:

$$\eta_i = \frac{r p_i T_s \lambda L_t}{\lambda_{charging} H_u \frac{\sigma_{O2trapped}}{0.23} p_s} \cdot 1 \tag{7}$$

The resulting equation for calculation of p_s can be obtained by combination of equations (6) and (5). The equation for p_s yields:

$$\begin{aligned}
p_s &= \frac{\frac{p_i}{K} + p_v}{1 + \frac{\eta_{iHP}}{K} \frac{H_u}{0.23 r L_t} \frac{\lambda_{charging} \sigma_{O2trapped}}{T_s \lambda}} \\
\eta_i &= \left(\eta_{iHP} + \frac{0.23 r L_t K \lambda T_s}{H_u \lambda_{charging} \sigma_{O2trapped}} \right) \frac{1}{1 + K \frac{p_v}{p_i}}
\end{aligned} \tag{8}$$

Thus, searched for relationships between changes in η_i and p_s with respect to changes in η_{iHP} can be calculated using equations (6), (7) and (8). The changes of net indicated efficiency are reduced due to throttling in comparison with gross efficiency changes.

To make this contribution more complete, the relationship for $\lambda_{charging}$ is presented below according to [10]:

$$\lambda_{charging} = \frac{m_{trapped}}{m'_{id}} = \frac{m_p + m_L + m_b}{V_{z1} \cdot \frac{p_s}{rT_s}} \quad (9)$$

Estimating the density of unscavenged burnt residual gas in the compression volume from mean exhaust pressure and temperature (with correcting factor K') and assuming external mixture formation, mass fraction of oxygen in trapped mass can be found from:

$$\frac{\sigma_{O2trapped}}{0,23} = \frac{\lambda L_t}{1 + \lambda L_t} \left(1 - \left(1 - \frac{\sigma_{bO2}}{0,23} \right) \frac{m_b}{m_{trapped}} \right) = \frac{\lambda L_t}{1 + \lambda L_t} \left(1 - \left(1 - \frac{\sigma_{bO2}}{0,23} \right) \frac{K' r T_s p_v}{(\varepsilon - 1) r_v T_v \lambda_{charging} p_s} \right) \quad (10)$$

Then, inlet pressure relationship is modified to:

$$p_s = \frac{(1 + \lambda L_t) \left(\frac{p_i}{K} + p_v \right) + \frac{\eta_{iHP}}{K} \left(1 - \frac{\sigma_{bO2}}{0,23} \right) \frac{K' H_u p_v}{(\varepsilon - 1) r_v T_v}}{\left(1 + \lambda L_t + \frac{\eta_{iHP}}{K} \frac{H_u}{r} \frac{\lambda_{charging}}{T_s} \right)} \quad (11)$$

3. EXPERIMENTAL CALIBRATION OF THE MODEL

The above described mathematical model had to be calibrated first. The correction coefficient for pumping loop work " K " was tuned by calibration.

Experimental data were applied for calibration of the mathematical model. Experiments were carried out on a 4-stroke, 4-cylinder SI engine at part load, naturally aspirated, natural gas fuelled, equipped with a closed-loop lambda-control system with stoichiometric concept in the authors' laboratory. A detailed description of the SI engine can be seen e.g. in [4].

The engine is also equipped with a three-way catalyst placed in the exhaust manifold, a conventional ignition system and spark plugs. The engine test bench is equipped with a DAQ system for acquisition of slowly changing time-based quantities and also crank-angle based in-cylinder pressure acquisition. In-cylinder pressure is indicated in the first cylinder from the pulley. Special software was developed for recording both integral data and in-cylinder pressure patterns (usually over 100 working cycles) during the engine test bench operation and for the evaluation of such data off-line [11].

A diagram of the tested SI engine is shown in Figure 1.

Experiments were carried out on the engine at part load at three engine speeds (1400, 1700 and 2400 RPM). The selected b.m.e.p. value for the experiments was 200 kPa. Optimal value of spark advance was set for each engine operating speed (using the method of maximum value of brake torque of the engine (MBT), [4]).

The intake mixture was not heated and also no EGR was delivered into the intake during the calibration. The heater was not in operation during the measurements.

The aim of calibration was to determine the correction coefficient for pumping loop work " K ", see equation (1). The value was estimated as an arithmetical average value of all three performed experiments. The " $\lambda_{charging}$ " value was also calculated during the calibration. Both resulting calibration values of coefficients K and $\lambda_{charging}$ are presented in Table 1.

	All RPM
K	1.1833
$\lambda_{charging}$	90%

TABLE 1: The resulting values of coefficients K and $\lambda_{charging}$ at part load (b.m.e.p.=200 kPa) at all measured RPM.

TABULKA 1: Výsledné hodnoty koeficientů z kalibrace K a $\lambda_{charging}$ v nízkém zatížení ($p_e = 200$ kPa) při všech měřených otáčkách motoru.

The oxygen content in a trapped mass can be found from known air excess λ and from the assumption of zero scavenging flow through both opened valves in this simple case. The residual gas mass can be roughly estimated in the case of an exhaust manifold pressure higher than the inlet manifold one from the (unscavenged) volume and in-cylinder pressure at EC, and temperature after isentropic expansion from EO temperature.

A better way is to simulate the exhaust process with wall heat transfer as is possible in several simulation codes (e.g., three-pressure analysis in GT Power).

4. EXPERIMENTAL VERIFICATION OF THE MODEL

To solve this issue, it was necessary to find some means of changing the gross indicated efficiency (at the high-pressure part of the cycle), without any changes in mixture temperature ($T_s = \text{const.}$) and its composition during intake. Thus it was necessary to find a way for in-cylinder changes during charge combustion. The other goal of the authors was to find a solution for further increase in η_{iHP} at optimal settings of the SI engine. (Of course, there is a simple option of changing η_{iHP} just due to detuning of the optimal settings of the engine. However, it was not the authors' goal.)

Two research works served as inspiration in the solving of this issue, [12] and [13]. Two spark plugs in each cylinder of the SI engine were applied in these two works. It is possible to reduce the combustion duration using a two-spark plug arrangement (Figure 2) in comparison with the conventional one-spark plug arrangement.

To realize the mentioned two-spark plug arrangement of each cylinder it would be necessary to significantly modify the tested engine.

However, because it was not possible to perform such a large modification of the tested engine, an alternative method was chosen. This involved application of a special multi-spark plug (DR14 ZC) with three sparks at the same time, [14], as shown in the detail in Figure 3, [15].

In this way, ignition of the charge (start of combustion) in three various places can be realized. Of course, it is not possible to fully achieve the conditions in a double spark chamber by application of this one multi-spark plug because two spark plugs are placed on the opposite sides of the double spark chamber, see Figure 2. The distance between the two seats of combustion is therefore large.

FIGURE 2: Double spark combustion chamber, [13].

OBRÁZEK 2: Uspořádání spalovací komory se dvěma zapalovacími svíčkami, [13].

FIGURE 1: Layout of the engine test bench.

OBRÁZEK 1: Schéma uspořádání zkušebního stanoviště.

FIGURE 3: Special multi-spark plug DR14 ZC with three sparks at the same time, [14], [15].

OBRAZEK 3: Speciální zapalovací svíčka se třemi jiskrami současně, [14], [15].

Verification experiments were performed at the same part engine load as used for calibration (b.m.e.p. = 200 kPa) at the same three engine speeds (1400, 1700 and 2400 RPM). Experimentally acquired data, required for verification, were inserted into the model. Calculated and measured results are presented below.

5. CALCULATED AND MEASURED RESULTS

This chapter presents the measured and calculated results. Measured and calculated results for the conventional spark plug (with only one spark) are designated **Basic FR7**. Calculated and measured results for the multi-spark plug (with three sparks at the same time) are designated **Change ZC**.

5.1. INFLUENCE ON PUMPING LOSSES AND NET INDICATED EFFICIENCY

The influence of a change in the gross indicated efficiency η_{iHP} on the average intake manifold pressure p_s (which represents the pumping losses) is presented in Figure 4 for all three engine speeds. As can be seen in Figure 4, the calculated trends (increase or decrease) for values of p_s are similar to the measured trends, which is positive.

Note that large deviations between calculated and measured values of p_s are probably caused by the applied high-pressure in-cylinder pressure transducer. In-cylinder pressure data should be corrected using the three pressure analysis in the future.

Values of the gross indicated efficiency η_{iHP} , presented in Figure 4 (also in Figure 5), are calculated using measured in-cylinder pressure data.

The multi-spark plug increases low-speed gross efficiency apart from at the highest used speed, see Figure 4 and

Figure 5, which is reflected by the reaction of inlet pressure. The simplified simulation also displays this feature, but in a more moderate form.

Calculated and measured values of the net indicated efficiency η_i are presented in Figure 5 at all three engine speeds. As can be seen in Figure 5, calculated trends in values of η_i (in relation to the changes of η_{iHP}) are also similar to the measured trends, which is a positive finding.

Based on the measured results, it appears that the positive effect of the special multi-spark plug on the gross indicated efficiency at part load takes place only within a lower RPM range (up to about 1700 RPM in this case) compared to the conventional spark plug. One reason for this is probably the higher intensity of turbulence in the combustion chamber at higher RPM, which may contribute to improvement of the gross indicated efficiency even with a conventional spark plug.

6. CONCLUSION

The calculated and measured results show that the proposed mathematical model is applicable for prediction of the values of both the net indicated efficiency η_i and pressure p_s (pumping losses) in relation to a change in the gross indicated efficiency η_{iHP} while throttling is used to fix the required b.m.e.p. The research was carried out at part load on a throttled SI engine and with fixed values of b.m.e.p. and RPM.

According to the results, it appears that an increase in the gross indicated efficiency results in an increase in both the net indicated efficiency and also the pumping losses, and vice versa.

An increase in the pumping losses causes the positive change in the net indicated efficiency to be smaller compared to the increase in the gross indicated efficiency. The negative change of the pumping losses is nevertheless smaller than the change in the gross indicated efficiency. For the cases tested it only reaches approx. 15% of the change in the gross indicated efficiency.

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FIGURE 4: Influence of a change in the gross indicated efficiency η_{iHP} on the pressure p_s (at $T_s = \text{konst.}$); sim. & simulation of trends using (11), exp. & experiments.

OBRAZÉK 4: Vliv změny indikované účinnosti vysokotlaké části cyklu η_{iHP} na tlak p_s (při $T_s = \text{konst.}$); sim. & simulace trendu podle (11), exp. & experimenty.

FIGURE 5: Influence of a change in the gross indicated efficiency η_{iHP} on the net indicated efficiency η_i (at $T_s = \text{konst.}$); sim. & simulation of trends according to (8), exp. & experiments.

OBRAZÉK 5: Vliv změny indikované účinnosti vysokotlaké části cyklu η_{iHP} na indikovanou účinnost celého cyklu η_i (při $T_s = \text{konst.}$); sim. & simulace trendu podle (8), exp. & experimenty.

LIST OF NOTATIONS AND ABBREVIATIONS

λ	Air excess coefficient [-];
η_i	Net indicated efficiency [-];
η_{iHP}	Gross indicated efficiency [-];
η_{iLP}	Indicated efficiency of the low-pressure part of cycle [-];
ε	Compression ratio [-];
$\lambda_{\text{charging}}$	Charging ratio [-];
σ_{bO_2}	Mass fraction of oxygen in residual gas [-];
$\sigma_{O_{2\text{trapped}}}$	oxygen mass fraction in the trapped mass (including residual gas and fuel) [-];
AL1Z	Ignition timing before the TDC [$^\circ$ bTDC];
bme _p	Brake mean effective pressure [Pa], in Czech & p_e ;
bsfc	Break Specific Fuel Consumption [g/kWh];
CA	Crankshaft Angle [$^\circ$], Czech & OKH;
CH ₄	Molar fraction of unburned hydrocarbons (Methane) [ppm];
CO ₂	Molar fraction of carbon dioxide [%vol.];
CR	Compression Ratio;
DAQ	Data AcQuisition system;
dQ _{nm}	Rate-of-Heat-Release [1/ $^\circ$ CA];
ECU	Electronic Control Unit;
EGR	Exhaust gas recirculation;
FR7	Marking of conventional spark plug;
H _u	Lower calorific value of fuel [J/kg];
ICE	Internal combustion engine;
K	Correction coefficient for pumping loop work [-];
Lt	Stoichiometric A/F ratio [kg/kg];
m _{id}	Reference ideal charging mass related to the swept volume V_{z1} [kg];
m _b	Burnt residual gas mass per single cycle [kg/cycle];
MBT	Maximal Brake Torque;

m _L	Air mass per single cycle [kg/cycle];
m _p	Fuel mass per single cycle [kg/cycle];
Mt	Engine torque [Nm];
m _{trapped}	trapped mass in a cylinder per cycle [kg/cycle];
NG	Natural gas;
NO	Molar fraction of nitric oxide [ppm];
optim.	optimal;
pCyl	In-cylinder pressure [bar];
p _i	Indicated mean effective pressure (IMEP);
p _s	Average inlet manifold pressure [Pa];
p _v	Average exhaust manifold pressure [Pa];
Q _d	Chemical energy delivered to the cycle [J/cycle];
r	specific gas constant [J/kg/K];
RPM	Revolutions Per Minute [min ⁻¹];
SI	Spark-ignition;
T _{cyl}	In-cylinder temperature [K];
TDC	Top Dead Center, Czech & HU;
T _s	Inlet manifold temperature [$^\circ$ C];
V _{z1}	Displacement of one cylinder [m ³];
W _i	Net indicated work of the cycle [J];
W _n	Negative pumping work of the cycle [J];
ZC	Marking of multi-spark plug;

REFERENCES

- [1] MACEK, J. (2012). [Combustion Engines](#) (in Czech), Czech Technical University in Prague, pp. 262, ISBN 978-80-01-05015-6.
- [2] 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.
- [3] Kutlar, A., O., Arslan, H., Calik, T., A. (2005). [Methods to Improve Efficiency of Four Stroke Spark Ignition Engines at part Load](#), Elsevier Ltd.
- [4] MIKLÁNEK, Ľ. (2012). The Influence of Mixture Heating and EGR Delivery on SI Engine Efficiency at Part Load – Final Results, In: [MECCA](#), Vol. 10, No. 1, pp. 28-37. ISSN 1214-0821.
- [5] MIKLÁNEK, Ľ. (2011). [Tools for Optimization of SI engine at Part Load](#) (in Czech), Dissertation Theses, Czech Technical University in Prague.
- [6] MIKLÁNEK, Ľ. (2012). NO_x Emissions with Application of the Atkinson and the Miller Cycles at Partial Load with Mixture Heating, In: [MECCA](#), Vol. 10, No. 2, pp. 44-52. ISSN 1214-0821.
- [7] 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.
- [8] GOTO T., HATAMURA, K., et al. (1994). [Development of V6 Miller Cycle Gasoline Engine](#), SAE Paper 940198, SAE Int. Warrendale.
- [9] 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.
- [10] HEYWOOD, B., J. [Internal Combustion Engine Fundamentals](#). McGraw-Hill, Inc. USA, 1988, pp. 930, ISBN 0-07-028637-X.
- [11] Takáts M. (2009). [INTEC – software for evaluation of in-cylinder pressure record](#), Josef Božek Research Center Code Library, CTU in Prague.
- [12] Tsao C. K., Han Z. (1993). [An Exploratory Study on Combustion Modeling and Chamber Design of Natural Gas Engines](#), SAE Paper 930312, SAE Int. Warrendale.
- [13] Fiorenza R., Formisano G., Martorelli M., Sbarbati F. (2005). [Combustion/NVH Analysis for Development of a 2-Valve Double Spark Plug Engine](#), SAE Paper 2005-01-0236, SAE Int. Warrendale.
- [14] <http://www.brisk.cz/>.
- [15] <http://www.adverts.ie/>.