

NUMERICAL ANALYSIS OF THE IMPACT OF EGR ON THE KNOCK LIMIT IN SI TEST ENGINE

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Summary. The paper presents results of modelling of the thermal cycle of the SI test engine using AVL FIRE software. Results are presented of impact of exhaust gas recirculation on the knock limit, thermal efficiency and mean indicated pressure of the test engine. Modelling was done for the test engine powered by gasoline. This engine runs at a constant speed. The engine is designed for lean mixtures combustion. The paper presents results of optimizing ignition advance angle of the engine with EGR with the detection of knock process. The knock process was modelled with the use of AnB model.

Key words: internal combustion engine, thermal cycle, modelling, combustion, knock, exhaust gas recirculation.

INTRODUCTION

Engines are designed to maximize power and economy while minimizing exhaust emissions. In some engines, a fraction of the engine exhaust gases is recycled to the intake to dilute fresh mixture for control of NO_x emission [6]. In order to reduce the in-cylinder temperature, a charge dilution must be done. One of the effective methods used to dilute the fresh charge is to recycle some part of the exhaust gases back into the cylinder. This method is called exhaust gas recirculation (EGR). Using EGR will lead to a decrease in the in-cylinder temperature and a decrease in knocking tendency. This method makes possible to improve advance timing to achieve higher thermal efficiency compared to engines operating on mixtures without fraction of recycled exhaust gas. In addition, EGR will reduce the in-cylinder NO_x production [11]. Shizuo and all [13] investigated the effects of EGR on direct injection gasoline engine. They confirmed that an appropriate volume of EGR improves fuel economy and HC emission. This phenomenon is presumably due to the intake temperature increase by EGR, which improved the flame propagation in the relatively lean area of the air-fuel mixture, which is not uniformly distributed. Excessive EGR induces deterioration of fuel economy and HC emission prior to occurrence of misfire. The limit to the volume of EGR is presumably due to the worsened flame propagation associated with reduced oxygen concentration, not to the worsening ignition performance [13]. Amr and Saiful [11] have investigated a supercharged natural gas SI engine with exhaust gas recirculation. They have confirmed that the

increase of the percentage of EGR dilution in the inlet mixture decreases the oxygen concentration, and consequently, it decreases the combustion rate significantly. For instance, the increase of EGR dilution from 0% to 10% at atmospheric inlet conditions increased the total combustion duration from about 74 deg to 95 deg. The maximum tolerable EGR dilution limit increased from about 10% to 12%. The use of cooled EGR with a dilution rate ranged from 20% to 30% depending on engine speed suppressed engine knocking and allowed using high inlet pressure condition at relatively high compression ratio values. Cooled EGR has the potential of reducing the maximum burned gas temperature and consequently NO emission in high compression ratio conditions [11]. The improvement in fuel consumption with increasing EGR is due to three factors: firstly, reduced pumping work; secondly, heat loss to the cylinder walls; and thirdly, a reduction in the degree of dissociation in the gases burned at high temperature [14]. Li and all [15] presented the experimental study results carried out on an electronically controlled fuel injection „stoichiometric gasoline engine” by using cold EGR and increasing compression ratio to improve fuel economy and reduce emissions. They stated that stoichiometric mixture is not an economical mixture for gasoline engines and its use may lead to poor fuel economy and, consequently, an increase in CO₂ emission. It is an important topic to improve the fuel economy of gasoline engines and make them operate at stoichiometric air–fuel ratio without deteriorating emissions from the engines [15]. An investigation was carried out to improve both fuel economy and emissions on gasoline engine at stoichiometric ratio by using cold EGR to suppress the knocking and increasing in compression ratio of the engine [15]. In addition, they stated that when the compression ratio was increased from 8 to 11.8 and EGR ratio as well as air swirl ratio of the engine were optimized, the fuel economy was improved by 5.3% and the NO_x and (NO_x + HC) emission was decreased by 54.8% [15].

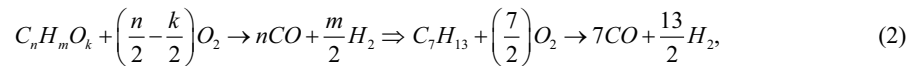
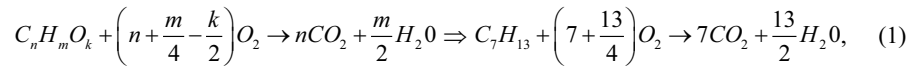
Knock is an abnormal combustion process occurring in SI engines. It is characterized by the occurrence of pressure oscillations. This phenomenon can destroy piston, exhaust valves or rings. Due to this disadvantageous effect it is important to avoid knock phenomenon but on the other hand, due to efficiency reasons, it is desirable to work as close to knock combustion as possible. The most accepted theory that explains engine knock is the auto-ignition theory [6]. The auto-ignition theory states that when the fuel–air mixture in the end gas region ahead of the flame front is compressed to sufficiently high pressure and temperature, the fuel oxidation process can occur in parts or in the entire end gas region. This releases the chemical energy in the end gas region at extremely high rates resulting in high local pressures. The non-uniform pressure distribution inside the combustion chamber causes pressure waves or shock waves to propagate across the chamber causing noise which is known as knock [6].

DESCRIPTION OF THE ECFM COMBUSTION MODEL

Turbulent combustion process is based on the conservation equations of chemical reaction flows. A variety of combustion models is available and successful calculations have been conducted. The CFM (Coherent Flame Model) has been successfully used for modelling the process of combustion in spark ignition engines. The ECFM-3Z (Extended Coherent Flame Model 3-Zones) model belongs to a group of advanced models of the combustion process in a compression ignition engine. For several years it has been successfully used, constantly modified and improved by many researchers [19]. Together with turbulence process sub-models (e.g. the k-zeta-f), exhaust gas component formation, knock combustion and other sub-models, they constitute a useful tool for modelling and analysis of the thermal cycle of the compression ignition internal combustion engine.

Modelling of combustion process was carried out using an advanced model of combustion. ECFM (Extended Coherent Flame Model) model was used based on the phenomenon of turbulent

mixing zone of air, fuel and exhaust. The ECFM has been developed in order to describe combustion in spark ignition engines. This model allows the modelling of the combustion process of air-fuel mixtures with EGR effect and NO formation. The model relies on description of unburned and burnt zones of the gas. A turbulent premixed combustion zones are characterised by chemical time scale, integral length scale and turbulence intensity. In reciprocating internal combustion engines the chemical time scales are much smaller in comparison to the turbulent scales. The concept of combustion model is based on a laminar flamelet idea, whose velocity and thickness are mean values, integrated along the flame front. The thickness of the flame front layer depends on the pressure, the temperature and content of unburned fuel in the fresh zone. In addition, it is assumed that reaction takes place within relatively thin layers that separate the fresh unburned gas from the fully burnt gas. This model uses a 2-step chemistry mechanism for the fuel conversion [1]:



where: n , m , k – number of the atoms of carbon, hydrogen and oxygen in the fuel, respectively. Hydrocarbon C_7H_{13} was taken as a fuel.

The reaction (2) of formation of CO and H_2 is taken into account for stoichiometric and fuel-rich mixtures, while for lean mixtures this reaction is omitted. The unburned gas phase consists of 5 main unburned species: fuel, O_2 , N, CO_2 and H_2O . At the burnt gas phase it is assumed that no fuel remains. The burnt gas is composed of 11 species, such as O, O_2 , N, N_2 , H, H_2 , CO, CO_2 , H_2O , OH and NO.

To model the formation of the nitrogen oxides Zeldovich extended mechanism was used. To detect the knock process, which may occur in the modeled engine, the knock model AnB was used. The model is based on the knowledge of the auto-ignition delay. This knock model is based on two equations. It was especially constructed for the extended coherent combustion flame model. The first equation determines the fuel consumption y_{fu} during auto-ignition process and the second equation defines the ignition delay θ [1]:

$$\frac{\partial y_{fu}}{\partial t} = y_{fu} c_1 e^{-\frac{T_a}{T_{gb}}}, \quad (3)$$

where: c_1 – constant, T_a – the activation temperature, T_{gb} – the local temperature of the burnt gas.

The second equation defines the ignition delay θ :

$$\theta = A \left(\frac{RON}{100} \right)^{c_2} p^n e^{\frac{B}{T_{fr}}}, \quad (4)$$

where: RON – fuel octane number, limited up to 140, used in the ignition delay expression, c_2 – constant, p – pressure in bar, T_{fr} – the local temperature of the fresh gas in K, n – pressure exponent in the ignition delay expression, A – pre-exponential factor in the ignition delay expression due to auto-ignition (Arrhenius approach) and B – activation temperature in the ignition delay expression.

MODELLING RESULTS

Modelling of the thermal cycle of the test spark ignition engine in the AVL FIRE program was carried out. The object of investigation was a spark ignition S320ER internal combustion engine fed with gasoline. The engine was operated at a constant speed of 1000 rpm. The work investigates the influence of EGR on engine operating parameters, on NO content in the exhaust gases and reduces engine knocking phenomena. The study was conducted for the excess air ratio equal to 1.2. The choice of this point of operation was dictated by the fact that in these conditions, the engine was working properly, the knock phenomenon occurred at the larger angles of ignition advance.

The percentage of exhaust gases recycled back to the engine intake (%EGR) was calculated as a percentage of the total inlet mass flow rate as follows:

$$\%EGR = \frac{\dot{m}_{EGR}}{\dot{m}_a + \dot{m}_f + \dot{m}_{EGR}}, \quad (5)$$

where: \dot{m}_{EGR} is mass rate of EGR, \dot{m}_a is mass rate of air, \dot{m}_f is mass rate of fuel, respectively in kg/s.

One of the parameters determining the performance of the combustion process in the engine is the indicated work.

$$p_i = \frac{\sum_{180}^{540} \frac{p_n + p_{n+1}}{2} (V_{n+1} - V_n)}{V_s}, \quad (6)$$

where: p_n, p_{n+1} are instantaneous values of the pressure in the cylinder [MPa], V_n, V_{n+1} are instantaneous values of the cylinder volume [m³], V_s is displacement volume [m³].

The average value of the indicated efficiency, expressed in % is equal to:

$$\eta_i = \frac{p_i V_s}{Q} 100\%, \quad (7)$$

where: Q is total heat supplied to the engine [MJ].

The heat supplied to the engine cylinder:

$$Q = \frac{V \rho W}{0.5nt}, \quad (8)$$

where: V is volume of gasoline delivered to the engine cylinder [m³], ρ is density of gasoline [kg/m³], W is calorific value of gasoline [MJ/kg], n is speed engine [rpm], t is time consumption of gasoline delivered to the engine cylinder [min].

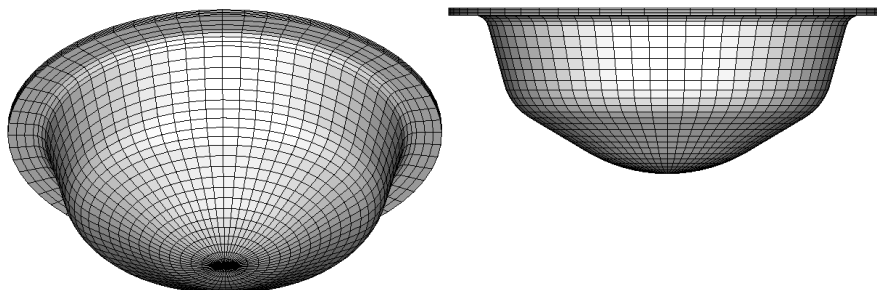


Fig. 1. The computational mesh for combustion chamber

Table 1. Modelling parameters

Engine rotational speed	1000 rpm
Cylinder bore	120 mm
Stroke	160 mm
Connecting-rod length	275 mm
Squish	2 mm
Initial pressure for 180 deg before TDC	0.9 MPa
Initial temperature for 180 deg before TDC	310 K
Lambda	1.2
EGR	0-12.5%
Fuel	C_7H_{13}
FIRE sub-models	k-zeta-f
Turbulence model	Coherent Flame Model ECFM
Combustion model	Extended Zeldowich Model
NO formation model	AnB
Knock model	

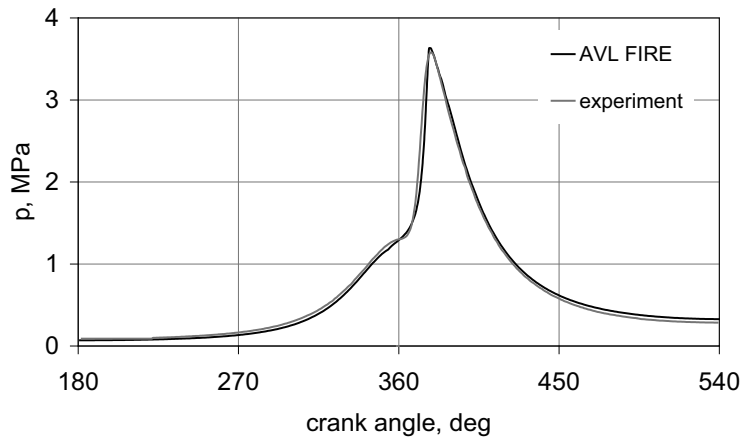


Fig. 2. Courses of pressure obtained by indicating of real test engine and by modeling using FIRE software

Computations were conducted for the angle range from -180 deg CA before TDC (top dead center) to 180 deg CA (crank angle) after TDC. The mesh of the modelled combustion chamber (Fig. 1) of the S320ER test engine consisted of nearly 30000 computation cells. Two-layered wall boundary layer was considered. Model tests were carried out in FIRE software. In the first stage of model calibration was made by comparing the changes of pressure courses from the real test engine with the course obtained by the FIRE software modeling. Result of calibration of the model is presented in Fig. 2. The variation of combustion pressures is shown, obtained as a result of indicating the real test engine and modelling with AVL FIRE software, respectively, for the same initial conditions and settings and spark advance timing. Designated engine efficiency is the gross efficiency, the modelling does not include charge exchange loop.

Fig. 3 presents the results of engine thermal cycle optimization in terms of advance angle, for the conditions without EGR.

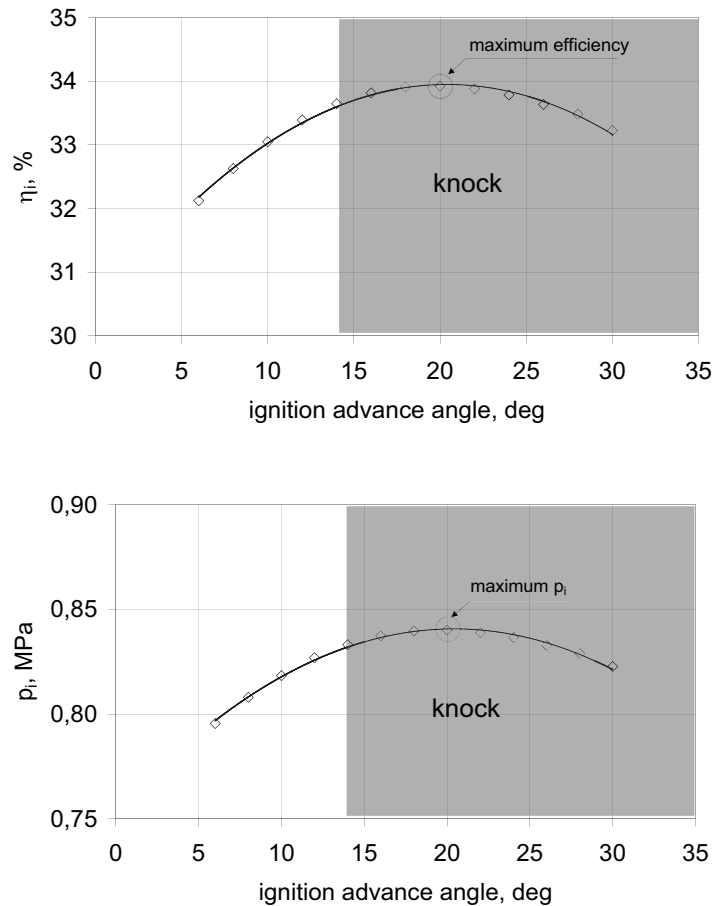


Fig. 3. Indicated thermal efficiency and mean indicated pressure of modelled test engine with marked knock area, without EGR

The gray area indicates the conditions for the occurrence of knock in the test engine. Maximum efficiency possible to obtain, without the danger of knock, was 33.7% and mean indicated pressure is 0.83 MPa. The test engine, thus, can operate with a maximum angle of ignition advance equal to 14 deg before TDC. Further enhancing of the value of the angle of ignition advance causes knock, which can contribute to engine damage.

Fig. 4 shows a comparison of the indicated thermal efficiency curves obtained for a different EGR ratio and for a fixed angle of ignition advance, equal to 10 deg before TDC, ignition timing for maximum efficiency and conditions limited by knock onset. It turned out that for the modelled test engine, the curve that determines knock limit is quite close to the curve that determines the maximum engine efficiency.

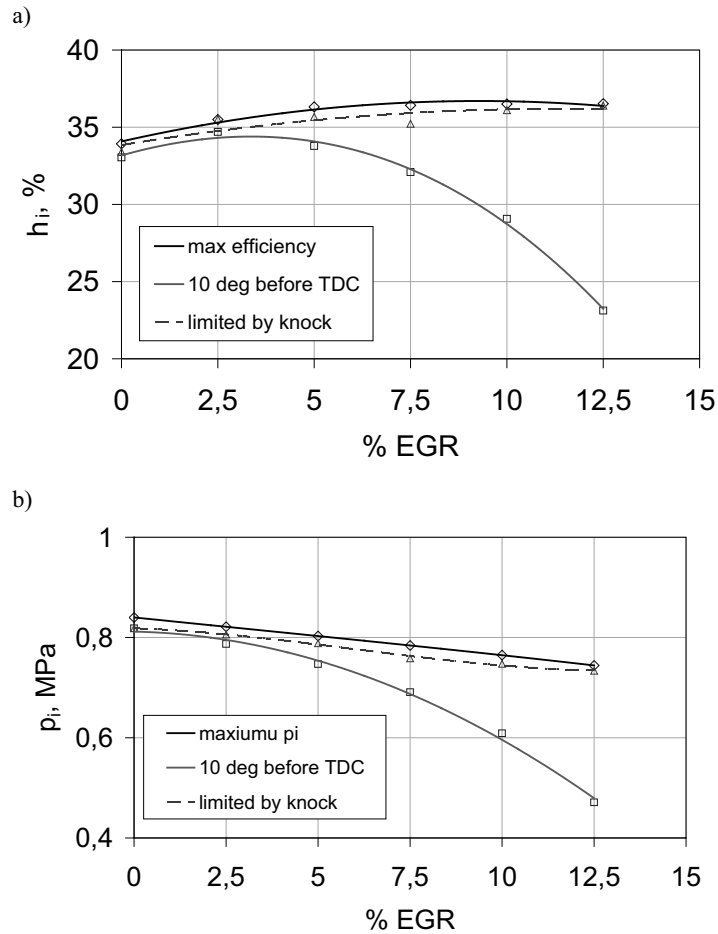


Fig. 4. a i b. Indicated thermal efficiency and mean indicated pressure

The test engine, of course, cannot work under conditions where there is danger of knock phenomena occurrence. For maximum EGR ratio, equal to 12%, an increase of indicated thermal efficiency was observed from 23% do 36.4%. This was due to change in the angle of ignition advance. For maximum efficiency, at 12.5% EGR, the ignition advance should be changed to 60 deg before TDC. Unfortunately, this resulted in the occurrence of knock phenomenon. The maximum ignition timing for these conditions could be below 50 deg before TDC. Optimizing the ignition angle has also a positive effect on the value of the mean indicated pressure. For the constant ignition angle equal to 10 deg and 12.5% EGR ratio, the value of the mean indicated pressure was 0.47 MPa and for the optimized ignition angle equal to 50 deg, TDC reached 0.74 MPa.

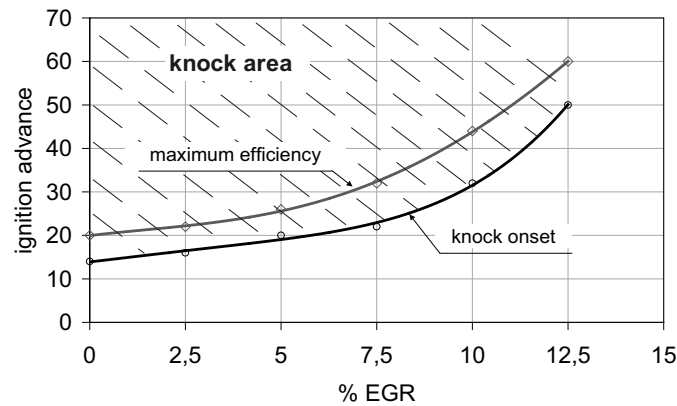


Fig. 5. Curves of spark ignition advance for the maximum value of thermal indicated efficiency and limited by knock phenomenon onset

Fig. 5 shows the area of knock combustion of the test engine. The impact is illustrated of ignition advance angle and EGR ratio on the test engine performance. The curve called „knock onset” refers to the critical angle of ignition at which knock in the combustion process of the test engine begins. The second curve defines the ignition angles for which the maximum efficiency could be achieved but because of the danger of knock phenomenon it cannot be achieved.

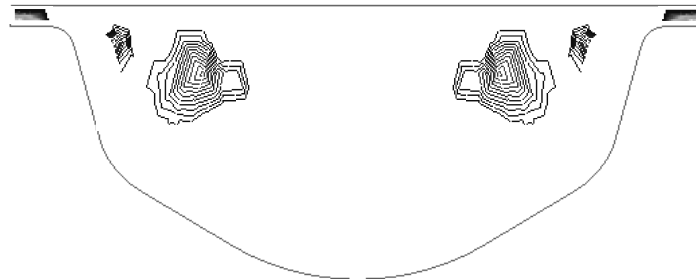


Fig. 6. Places of knock onset in the combustion chamber of the test engine
 $\lambda = 1.2$ for and ignition advance 20 deg before TDC

Fig. 6 shows the areas for the formation of a knocking in the modeled combustion chamber for ignition advance angle of 20 deg after TDC. These areas are located near the surface of the piston squeezing surface.

CONCLUSIONS

A small exhaust gas ratio in a fresh load causes an increase in engine thermal efficiency. For the modelled test engine 2.5% EGR ratio caused an increase in the efficiency of about 1.7%. Increasing share of EGR in the fresh charge decreased value of mean indicated pressure. Limit value of EGR ratio for the modelled engine was 12.5%. With greater participation of EGR ratio combustion process slowed down too much and thus operation of engine was no longer effective.

Changing the timing advance caused a considerable improvement in engine operating parameters. Limit values were determined by the occurrence of knock process. Exhaust gas recirculation not only has a positive effect on the reduction of NO in exhaust gases, but also reduces the possibility of knock onset. Engine equipped with exhaust gas recirculation system should be able to change the ignition angle, depending on the ratio of the exhaust gases recirculation.

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NUMERYCZNA ANALIZA WPŁYWU EGR NA GRANICĘ SPALANIA STUKOWEGO W BADAWCZYM SILNIKU SPALINOWYM

Streszczenie. W pracy przedstawiono wyniki modelowania obiegu cieplnego tłokowego silnika spalinowego z wykorzystaniem programu FIRE. Modelowano obieg silnika o zapłonie iskrowym z procesem recyrkulacji spalin. Analizowano wpływ udziału EGR na zmniejszenie toksyczności spalin oraz na ograniczenie obszaru pracy silnika, w którym występuje spalanie stukowe. Okazało się, że dzięki optymalizacji obiegu silnika z EGR pod względem kąta wyprzedzenia zapłonu, w badanym silniku, osiągnięto takie wartości kąta zapłonu, dla którego nie występuje jeszcze spalanie stukowe a jednocześnie osiąga się parametry zbliżone do optymalnych.

Słowa kluczowe: silnik spalinowy, obieg cieplny, badanie, spalanie stukowe, recyrkulacja spalin.