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THE METHOD OF PRACTICAL VERIFICATION OF THE MATHEMATICAL MODEL OF GAS ENGINE POWERED BY LPG

Summary. The results of preliminary practical verification of mathematical model of combustion process in gas engine are presented. The single-zone and zero-dimensional mathematical model has been prepared in the previous stage of research program, which has been carried in Transport Institute of Silesian Technical University.
PERKINS AD3.152G gas engine LPG powered was either tested on engine bench test and was an object of simulation. Bench tests have been done on wide range of engine speed and for various throttle openings. Engine test procedure provided also verification of ignition timing and A/F ratio.
Results were used to the verification of the main parameters of mathematical model. Upgraded mathematical model establishes a comparison level necessary for next stage of research program – application of LPG injection system.

METODA PRAKTYCZNEJ WERYFIKACJI MATEMATYCZNEGO MODELU SILNIKA GAZOWEGO ZASILANEGO LPG

Streszczenie. W opracowaniu przedstawiono wyniki wstępnej weryfikacji matematycznego modelu procesu spalania paliwa gazowego w silniku spalinowym. Model ten opracowano w pierwszej fazie badań prowadzonych w Instytucie Transportu Politechniki Śląskiej. Badaniom stanowiskowym i symulacyjnym poddano silnik gazowy PERKINS AD3.152G, zasilany mieszaniną propanu-butanu. Badania stanowiskowe przeprowadzono w pełnym zakresie prędkości obrotowej silnika, przy różnym otwarciu przepustnicy, dla różnych kątów wyprzedzenia zapłonu i odmiennym składzie mieszanki.
Wyniki, które otrzymano w trakcie badań stanowiskowych, wykorzystano do weryfikacji modelu matematycznego.

1. INTRODUCTION

Nowadays, mathematical models are often used to describe processes inside engine chamber. Mathematical models can be divided in two main groups according to heat release profile:

- Individual model – heat release profile is based on indicated pressure.
- Universal model – heat release profile is described by premised function (e.g. Wibe function).

Main advantage of universal model is possibility of realisation of combustion analyses, complete of all parameters. However, in the case it is necessary to premise the angle of combustion beginning and combustion duration.
The proper course to make an universal and reliable model is verification process of all function which represent the combustion process and which have been acquired during tests on the

engine bench. Verification method of the model which is presented in this paper is based on balance of energy.

2. ENGINE STAND AND PROCEDURE OF MEASUREMENT

The bench tests of PERKINS AD3.152G gas engine carried out on bench stand HPS-75, equipped with suitable actuators and sensors. The main information acquired at the stand are:
- indicated pressure,
- mass fuel consumption,
- exhaust gases emission,
- inlet manifold pressure,
- engine speed and engine load,
- temperatures of: air, fuel, mixture, exhaust gases, etc.

Data have been acquired and processed with the use of Keithley Acquisition Card and specialised software environment (ASYST). Bench tests included large range of engine working conditions (91 points measured) with variation of main parameters listed in table 1 [2]:

Table 1

The main parameters variation range

Parameter	Range
Engine speed	850.......2000 rpm
Throttle opening ratio	25...50 75 100%
Air/Fuel ratio	0,8..........1,4
Ignition advanced angle	5°......17° b.T.D.C

3. MATHEMATICAL MODEL AND ITS COMPUTER APPLICATION

The main equation of the single-zone and zero-dimensional mathematical model based on balance of energy was formed as below:

$$dL + dQ_s + dQ_B + dU = 0 \tag{1}$$

Where:

dQ_s – quantity of heat transferred to chamber walls;
$d\,Q_B$ – quantity of heat released in combustion process;
dU – change of mixture and burned gas inner energy;
dL – quantity of mechanical energy.

Combustion process simulation consists of larger part of engine cycle, when valves are closed: compression, combustion and expansion. The average temperature of gases and cylinder pressure are calculated during simulation. Analysing described above model is very simple thanks to computer application named EnComOne resolving model equation in function of engine crank angle. Computer program enables calculation for various engine construction parameters and different working conditions. Results of computer simulation are presented both in a table and graphical on a chart with possibility to save them in a text file.

4. VERIFICATION PROCEEDINGS

Verification process is based on bench test results. The most important is indicate of cylinder pressure profile. Proceedings include verification of:

- variation of total inner energy for fresh mixture and burned gas;
- the heat release rate;
- the wall transfer rate.

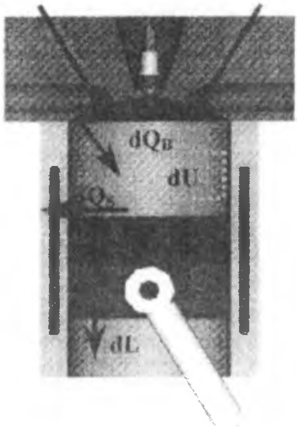

Fig. 1. The balance of energy in the combustion chamber
Rys. 1. Bilans energii w komorze spalania

4.1. Composition proportions and heat capacity of mixture

4.1.1. Molecular mass of mixture[1]

The premised molecular mass of mixture depends on A/F ratio. The mixture inside combustion chamber is consist of: fuel, air and a residual gases burned. The mole proportions were obtained for measured air and fuel mass consumption and A/F ratio. The molecular mass of combustible mixture inside combustion chamber is determined as below:

$$M_M = z_F \cdot M_F + z_A \cdot M_A + z_B \cdot M_B \left[\frac{kg}{kmol}\right] \tag{2}$$

Where:

z – mole proportion,
M – molecular mass,

Index:

F — fuel,
A – air,
B – burned gases,
M – mixture.

$M_F = 51\left[\frac{kg}{kmol}\right]$ - fuel: 50% propane/50% butane , $M_A = 28{,}72\left[\frac{kg}{kmol}\right]$

Table 2

Molecular mass and composition proportions of mixture

A/F ratio - λ	z_F	z_A	z_B	M_B	M_U
0,8	0,04	0,889	0,071	26,951	29,565
0,9	0,035	0,893	0,072	27,724	29,470
1,0	0,032	0,896	0,072	28,174	29,394
1,1	0,029	0,899	0,072	28,212	29,331
1,2	0,027	0,901	0,072	28,244	29,278
1,3	0,025	0,903	0,072	28,271	29,233
1,4	0,023	0,905	0,072	28,294	29,194

4.1.2. Heat capacity of mixture

The mixture is treated as semi-ideal gas. Heat capacity of mixture is determined by listed in table 3 functions of the temperature (range from T=300K to 1000K).

Table 3

The mole heat capacity of mixture

A/F ratio - λ	c_p (T) [kJ/(kmol*K)]
0,8	$1e\text{-}9*T^3 - 7e\text{-}6*T^2 + 0{,}0169*T + 26{,}457$
0,9	$1e\text{-}9*T^3 - 6e\text{-}6*T^2 + 0{,}0153*T + 26{,}567$
1,0	$1e\text{-}9*T^3 - 6e\text{-}6*T^2 + 0{,}0143*T + 26{,}678$
1,1	$9e\text{-}10*T^3 - 5e\text{-}6*T^2 + 0{,}0133*T + 26{,}760$
1,2	$9e\text{-}10*T^3 - 5e\text{-}6*T^2 + 0{,}0126*T + 26{,}815$
1,3	$8e\text{-}10*T^3 - 5e\text{-}6*T^2 + 0{,}0119*T + 26{,}870$
1,4	$7e\text{-}10*T^3 - 4e\text{-}6*T^2 + 0{,}0113*T + 26{,}925$

4.2. The heat release rate

The modelled coefficient of heat release is described by Wibe function [3]:

$$x = 1 - \exp[a \cdot \left(\frac{\varphi_C}{\Delta\varphi_c}\right)^{m+1}] \tag{3}$$

Where:

a – combustion complete parameter (CCP),
m – combustion velocity parameter (CVP),
φ_C – current angle during combustion process [CA deg],
$\Delta\varphi_C$ – combustion period (CP) [CA deg].

CVP and CP are verified. CCP was premised (a=6,908 then x_{max}=0,999).

4.2.1. Combustion period boundary angles

The boundary angles of combustion period were verified of the base on analysis of registered indicated pressure profile.

Where:

φ_V – Ignition advanced angle;

φ_S – Combustion start angle (CSA);

φ_E – Combustion end angle (CEA);

$\Delta\varphi_I$ – Combustion initiation period (CIP);

$\Delta\varphi_C$ – Combustion period (CP).

CSA and CIP are determined by pressure increase rate, dP/dφ. An example showed in table 4.
CEA is obtained by analysing of expansion politropic exponent. This method is based on finding such crank angle after T.D.C till the variation of politropic exponent is less then 1%. While tested politropic exponent is variable more the 1% till is combustion inside the chamber [4].

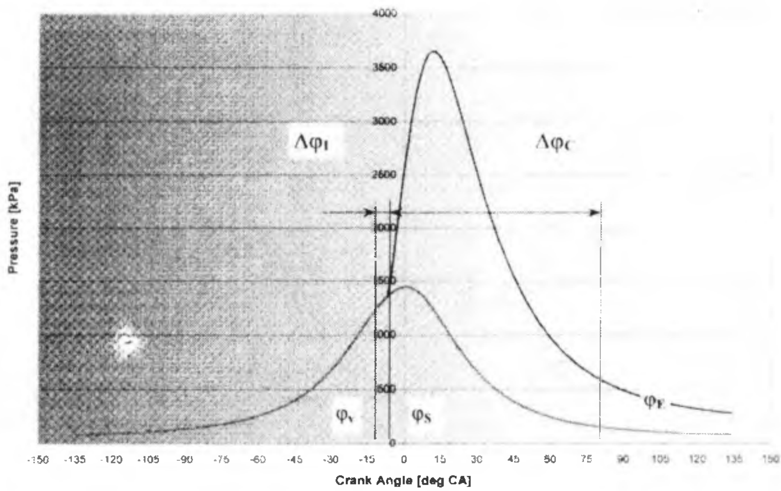

Fig. 2. Combustion period boundary angles, φ_V – ignition point, - φ_S – start of combustion
Rys. 2. Przebieg spalania w funkcji kąta obrotu wału korbowego

$$\Delta\varphi_I = \varphi_V - \varphi_S \tag{4}$$

$$n_i = \frac{\log P_w - \log P_i}{\log V_w - \log V_i} \tag{5}$$

Where:

n – polytropic exponent;

P_w – exhaust valve opening pressure;

P_i – current pressure;

V_w – exhaust valve opening cylinder capacity;

V_i – current cylinder capacity.

Table 4

Pressure increase rate after ignition point

φ [CA deg b.T.D.C]	P [kPa]	dP/dφ [kPa/ CA deg]
[redacted]		
16	1035,182	22,127
15	1084,586	49,404
14	1134,181	49,595
[redacted]		
12	1359,075	112,543
11	1475,242	116,167
10	1591,599	116,358
9	1725,887	134,288
8	1860,366	134,479
7	2011,440	151,074

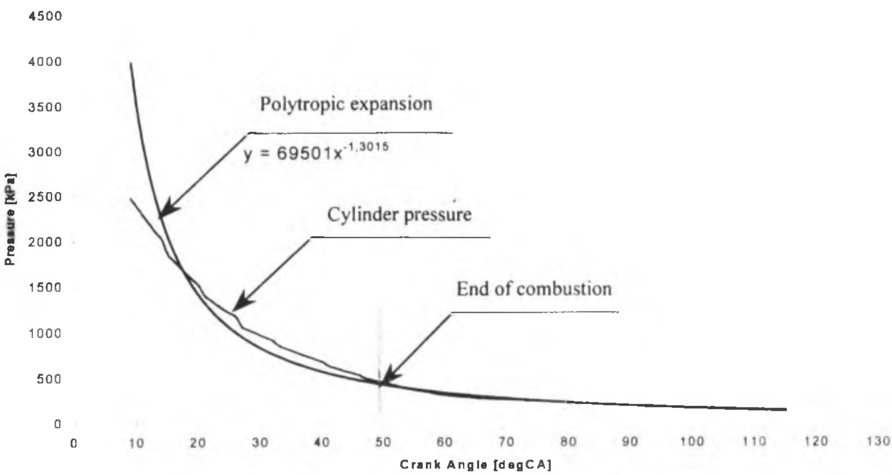

Fig. 3. Polytropic expansion and cylinder pressure as a function of crank angle
Rys. 3. Politropa i przebieg ciśnienia w funkcji obrotu wału korbowego

CP calculation was possible according to:

$$\Delta\varphi_C = \varphi_E + |\varphi_S| \qquad \text{for } \varphi_S < 0 \tag{6}$$

$$\Delta\varphi_C = \varphi_E \qquad \text{for } \varphi_S = 0 \tag{7}$$

$$\Delta\varphi_C = \varphi_E - \varphi_S \qquad \text{for } \varphi_S > 0 \tag{8}$$

4.2.2. The verification of CVP

The most suitable value of CVP is obtained by comparing of indicated pressure and modelled pressure profile. An example is presented on figure 4. The most corresponding value of CVP is obtained by comparison both indicated pressure curves and calculated pressure curves.

4.3. The rate of heat transfer to the cylinder walls

The rate of heat transfer to the cylinder walls is defined by the following equation:

$$\frac{dQ}{d\varphi}=\frac{1}{\omega}*\sum_{i=1}^{3}\alpha_i(\varphi)\cdot F_i(\varphi)\cdot\left[T(\varphi)-T_{W_i}\right]\cdot\left[\frac{J}{\deg CA}\right] \tag{9}$$

Where:

ω – engine angular speed;
$\alpha(\varphi)$ – coefficient of heat transfer (HTC);
$F(\varphi)$ – area of combustion chamber;
$T(\varphi)$ – average gas temperature;
T_W – average wall temperature.

Index:

i=1 – cylinder;
i=2 – piston crown;
i=3 – cylinder head.

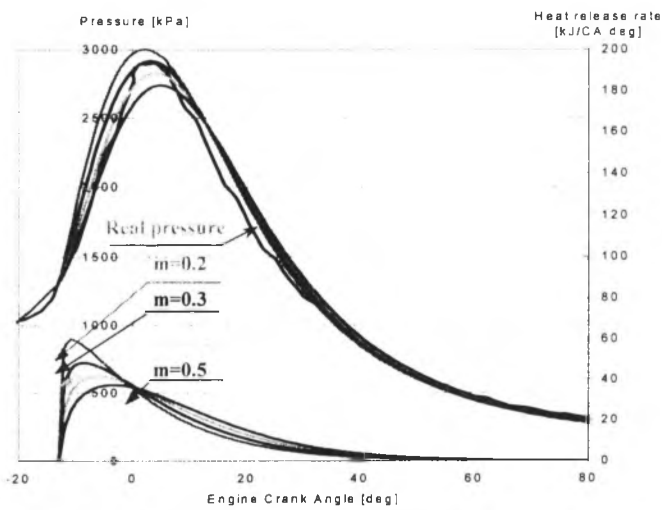

Fig. 4. The CVP influence on cylinder pressure and rate of heat release profiles
Rys. 4. Wpływ prędkości spalania na ciśnienie wewnątrz cylindra i przebieg współczynnika przejmowania ciepła

The results of calculations in which only HTC is verified are presented below. The influence of HTC on pressure was observed for optimised CVP (Fig. 5). Both, the cylinder gas temperature curves and of heat transfer rate to the cylinder wall curves, are presented on chart (Fig. 6). These curves were made for HTC by Serafimov hypothesis, because this gives the best results among tested HTC cases. There have chosen five hypothetical profiles of HTC [W/(m^{2}K)]:

1. Woschni $\alpha_W = 0{,}068c^{0,78}\ D^{-0,22}\ p^{0,78} / T^{0,52}$ (10)

2. Eichelberg $\alpha_E = 0{,}0067\ (p\ T\ c)^{0,5}$ (11)

3. Hohenberg $\alpha_H = 0{,}013\ V^{-0,06}\ p^{0,8}\ T^{-0,4}\ (1{,}4\ c)^{0,8}$ (12)

4. Eckert $\alpha_{Ec} = 46{,}33\ (p\ c)^{0,58}$ (13)

5. Serafimov $\alpha_S = 4{,}1868\ (4{,}32+9{,}85\ p)\ T^{1/2}\ c^{1/3}$ (14)

Where:

c – average piston velocity [m/s];
D – piston diameter [m];
p – cylinder pressure [kPa];
T – gas temperature [K];
V – combustion chamber capacity [m^3].

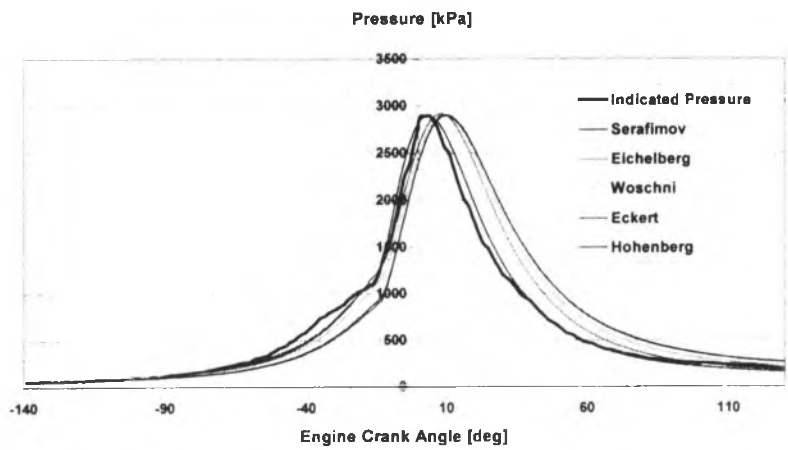

Fig. 5. The cylinder pressure curves for tested HTC hypothesises
Rys. 5. Przebieg ciśnienia dla wybranych funkcji przejmowania ciepła

5. DISCUSSION OF RESULTS

The results of mathematical model verification are showed that the calculated cylinder pressure profile is overlap with indicate pressure which has been registered on bench tests of engine.

Some of examples of using elaborated are presented below. The distribution of supplied and consumed energy are showed on figures 7 and 8. Energy balance of consumed energy during combustion process is presented on figure 9.

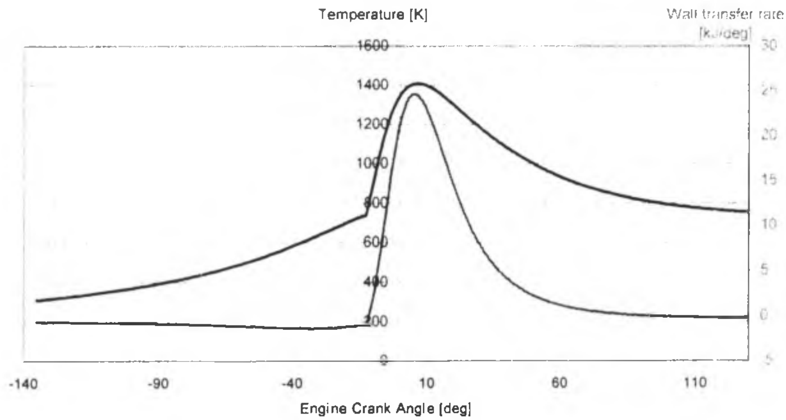

Fig. 6. The cylinder gas temperature and the rate of heat transfer to the cylinder walls for HTC by Serafimov
Rys. 6. Temperatura gazów wewnątrz cylindra i współczynnik przejmowania ciepła przez ścianki cylindra dla hipotezy Serafimowa

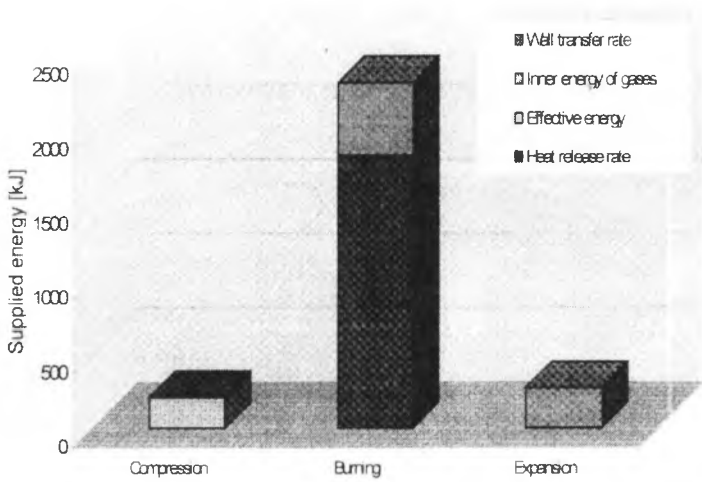

Fig. 7. Distribution of supplying energy
Rys. 7. Rozkład dostarczonej energii

6. CONCLUSIONS

A single-zone and zero-dimensional mathematical model has been supplied to simulate combustion of LPG in a three cylinder SI internal combustion of gas engine. The point of modelling can position of test bench and calculated results.

The model has shown to prediction of cylinder pressure and heat release shape in good agreement with experimental data under wide range engine speed and load conditions:

- The model gives possibilities to observe the influence of any parameters on combustion process and analysing direction of energy flow during this process.
- The possibilities of constant control of all parameters creates a particular image of all analysing processes and allows for whole cycle efficiency.

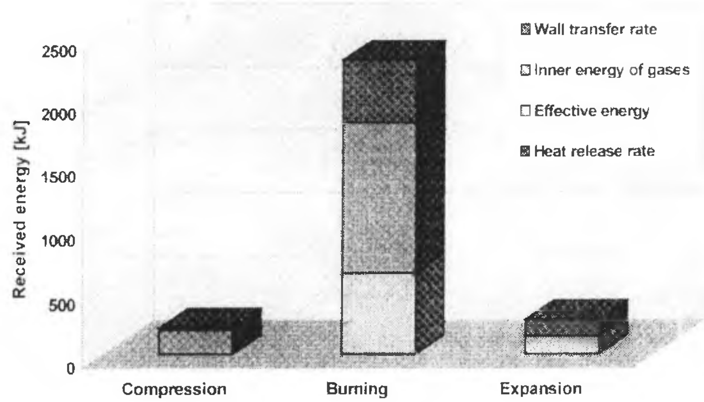

Fig. 8. Distribution of consuming energy
Rys. 8. Rozkład zużycia energii

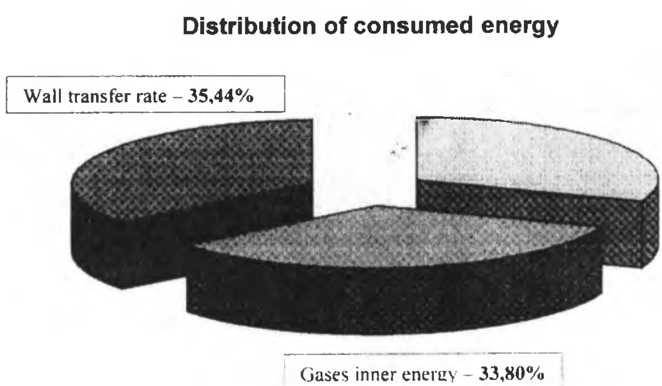

Fig. 9. Distribution of consumed energy in whole cylinder cycle
Rys. 9. Rozkład energii zużytej w komorze spalania

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