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Method for Determining Volumetric Efficiency and Its Experimental Validation

Andrzej Ambrozik[1], Dariusz Kurczyński[2], Piotr Łagowski[3]
[1–3]*Kielce University of Technology, Faculty of Mechatronics and Mechanical Engineering, Poland*

Abstract – **Modern means of transport are basically powered by piston internal combustion engines. Increasingly rigorous demands are placed on IC engines in order to minimise the detrimental impact they have on the natural environment. That stimulates the development of research on piston internal combustion engines. The research involves experimental and theoretical investigations carried out using computer technologies. While being filled, the cylinder is considered to be an open thermodynamic system, in which non-stationary processes occur. To make calculations of thermodynamic parameters of the engine operating cycle, based on the comparison of cycles, it is necessary to know the mean constant value of cylinder pressure throughout this process. Because of the character of in-cylinder pressure pattern and difficulties in pressure experimental determination, in the present paper, a novel method for the determination of this quantity was presented. In the new approach, the iteration method was used. In the method developed for determining the volumetric efficiency, the following equations were employed: the law of conservation of the amount of substance, the first law of thermodynamics for open system, dependences for changes in the cylinder volume vs. the crankshaft rotation angle, and the state equation. The results of calculations performed with this method were validated by means of experimental investigations carried out for a selected engine at the engine test bench. A satisfactory congruence of computational and experimental results as regards determining the volumetric efficiency was obtained. The method for determining the volumetric efficiency presented in the paper can be used to investigate the processes taking place in the cylinder of an IC engine.**

Keywords – **Cylinder filling process, engine working cycle, indicator diagram, piston internal combustion engine, volumetric efficiency.**

I. INTRODUCTION

Working processes in the cylinder of the piston internal combustion engine proceed in cycles. In those processes, the chemical energy contained in the fuel is converted by means of combustion to heat, which causes pressure increase and production of mechanical work, which is transferred to the receiver. The set of changes taking place in the engine cylinder, in which the fuel energy is transformed into the effective work transferred to the engine crankshaft, is called the engine working cycle [1]–[3]. Changes that occur in the cylinder are reflected in the pattern of changes in the pressure of the working medium in the cylinder. The graphic representation of those changes is given as a function of changes in the cylinder volume above the piston or the crankshaft rotation angle, and termed an indicator diagram [4]. The diagram is a basic source of information on working processes proceeding in the engine and makes it possible to assess the quality of those processes. One of the means of reducing a detrimental impact of piston combustion engines on the natural environment is to improve the course of processes that occur in engine cylinders. Such improvements result in lower fuel consumption, decreased emissions of noxious exhaust components and reduced engine noise.

The engine working cycle is performed by the working medium, which is a mixture of gases filling the cylinder volume [5], [6]. During the engine working cycle, the working medium continuously undergoes physical and chemical changes. While the engine cylinder is filled, a fresh charge flows in. The charge, together with exhaust gases left from the previous cycle, is compressed and then burnt. In expansion and exhaust, combustion products are the working medium [1].

In theoretical investigations into piston combustion engines, mathematical modelling is widely used to analyse the processes that occur in the engine cylinder [7]–[9]. In order to assess the impact of various factors on the parameters of engine performance, it is convenient to employ approximate, engineering calculation methods. The state of the working medium in the cylinder is unambiguously defined by four thermodynamic parameters: pressure p, temperature T, volume V and the number of moles of the medium M. In order to determine the quantities mentioned above, five independent equations are used: the first law of thermodynamics, namely the volume balance equation formulated at the assumption that the volume of the working medium in the cylinder is equal to the cylinder changing volume, energy conservation equation, state equation and balance equation of the working medium amount.

The cylinder filling process in four-stroke engines decides on the amount of the working medium delivered to the cylinder [4], [10]. When the piston is positioned at TDC, the space above the piston is filled with exhaust gases at pressure p_{ex}. The value of the pressure of the remaining exhaust gases p_{ex} depends mainly on the flow resistance in the exhaust system, i.e. the passage cross-section of the exhaust valves, length and shape of the pipes, type of mufflers and the exhaust gas flow velocity.

In naturally aspirated engines, the process of volumetric efficiency depends on the action of negative pressure in the cylinder caused by the piston movement towards BDC. The inflow of a fresh charge into the cylinder begins when the leftover exhaust gases from the previous working cycle become expanded to the pressure value lower than the pressure before the intake valves. In supercharged engines, the inflow of the working medium into the cylinder takes place at the pressure, the value of which is higher than the ambient pressure. Higher pressure is obtained with a supercharger or turbocharger by using the energy of exhaust gases.

In the filling process, the pressure in the cylinder varies constantly. Cylinder pressure variation is caused by the changeability in the expenditure of the fresh charge inflow into the volume of the cylinder from the intake system, the changeability in the valve passage cross-section, and the piston velocity [11]. The working medium flowing into the cylinder overcomes hydraulic resistance of the intake system elements, becomes heated by the walls of the cylinder and the exhaust gases remaining there. Consequently, the amount of the fresh charge that is sucked up is smaller than it could, theoretically, be held in the cylinder at the temperature and pressure that prevail before the intake valve.

The filling of the engine cylinder is assessed by the volumetric efficiency η_v [12]–[14]. It is a quotient of the mass of the charge actually delivered to the cylinder of naturally aspirated or supercharged engine by the theoretical mass of the charge that could flow into the cylinder at the thermodynamic parameters prevailing in the intake system:

$$\eta_v = \frac{m_r}{m_t}, \tag{1}$$

where m_r – mass of the charge actually delivered to the cylinder of the piston combustion engine,
m_t – theoretical mass of the charge which could occupy the piston swept volume under the conditions prevailing in the engine intake system.

Volumetric efficiency in a piston combustion engine depends on many factors, which can be classified as those related to the design and service. Design parameters that substantially affect the value of volumetric efficiency include the following [13], [15], [16]:

- combustion chamber geometry and dimensions,
- compression ratio,
- piston sweep-to-diameter ratio S/B,
- piston mean velocity, and related to that, mean flow velocities of the charge through the intake and exhaust ports,
- engine crankshaft rotational speed n,

– geometry of the engine intake and exhaust systems.

Service parameters that influence volumetric efficiency are as follows:

– parameters of the environment in which the engine operates: pressure, temperature, humidity,
– thermal state of the engine,
– engine load.

II. Mathematical Model of the Cylinder Filling Process

For the sake of calculating volumetric efficiency in a four-stroke piston IC engine, comparative thermodynamic Sabathe's engine cycle was employed, which is presented in Fig. 1 in *pV* coordinates. In the figure, point "ou" denotes the beginning of the opening of the intake valve, whereas point "ou'" denotes the exhaust valve closing. The beginning of the opening of the intake valve and the start of the cylinder filling process is denoted by letters "i", the instant of the exhaust valve closing is marked with letters "i'". Calculations of volumetric efficiency are made for the following assumptions [4]:

– cylinder filling process proceeds between piston TDC and BDC positions, at a constant cylinder mean pressure p_{mean}, which is computed,
– during the period both valves are open, the working medium contamination with combustion products is not accounted for,
– the amount of the working medium in the cylinder that corresponds to the conventional point "a" in the compression process line, when the piston is positioned at BDC, is equal to the working medium amount in the cylinder at the beginning of the compression.

Equations that describe the cylinder filling processes include the following:

- the first law of thermodynamics:

$$Q_{\mathrm{w}}^{\mathrm{p\text{-}a}} + H_{\mathrm{i}} = U_{\mathrm{a}} - U_{\mathrm{p}} + \int_{V_{\mathrm{p}}}^{V_{\mathrm{a}}} p\mathrm{d}V = H_{\mathrm{a}} - H_{\mathrm{p}} - \int_{p_{\mathrm{p}}}^{p_{\mathrm{a}}} V\mathrm{d}p, \tag{2}$$

where $Q_{\mathrm{w}}^{\mathrm{p-a}}$ – amount of heat transferred between the working medium and the cylinder walls during the p–a process, H – enthalpy of the working medium at a, d and p points of the engine working cycle presented in Fig. 1, respectively, U – the internal energy of the working medium at a and p points of the engine working cycle presented in Fig. 1, respectively, p – pressure of the working medium in the cylinder, V – volume of the working medium in the cylinder;

- volume balance equation:

$$V(\theta) = V_{\mathrm{c}} + V_{\mathrm{d}}(\theta), \tag{3}$$

where $V(\theta)$ – change in the engine cylinder volume above the piston depending on the engine crankshaft rotational angle, V_{c} – combustion chamber volume. $V_{\mathrm{d}}(\theta)$ – change in engine cylinder volume above the piston that results from the piston movement, which is dependent on the engine crankshaft rotational angle;

- the equation for the amount of the working medium that flowed into the cylinder in the filling process:

$$M_{\mathrm{i}} = \int_{\mathrm{t_p}}^{\mathrm{t_a}} f_{\mathrm{mean}} w_{\mathrm{mean}} \rho_{\mathrm{mean}}\, \mathrm{d}t, \tag{4}$$

where $f_{\mathrm{mean}} = \mu_{\mathrm{i}} f_{\mathrm{i}}$ – minimum cross-section of the working medium jet flowing into the cylinder;

μ_i – coefficient of the outflow through the valve, f_i – geometric passage cross-section of intake valves, w_{mean} – mean gas flow velocity in the minimum jet cross-section, ρ_{mean} – mean density of the working medium in the minimum cross-section of the jet flowing into the cylinder;

- the state equation:

$$pV = \bar{R}MT, \tag{5}$$

where $\bar{R}$ – universal gas constant, M – number of kilomoles of the working medium.

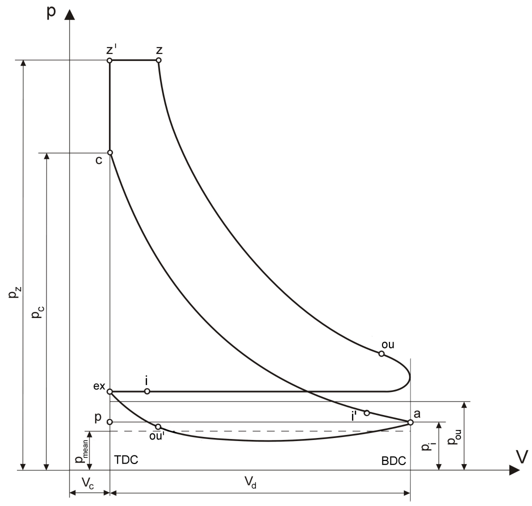

Fig. 1. Comparative thermodynamic Sabathe's engine cycle in the *pV* system [17]: a – the end of the cylinder filling (beginning of compression), i' – closing of the intake valve, c – end of compression, z' – end of isochoric heat delivery process (beginning of isobaric heat delivery process), z – end of isobaric heat delivery process (beginning of expansion), ou – opening of the exhaust valve, i – opening of the intake valve, ex – value of exhaust pressure for piston at TDC, ou' – closing of the exhaust valve, p_i – pressure in the intake system, p_{ou} – pressure in the exhaust system, p_{mean} – mean in-cylinder pressure in the filling process, TDC – top dead centre, BDC – bottom dead centre.

The starting point for determining the volumetric efficiency is the first law of thermodynamics (2). Enthalpies of the working medium in the intake system and those at points "a" and "p" in the comparative engine cycle (Fig. 1) are as follows:

$$\begin{cases} H_i = M_i \bar{c}_{pmean} T_i; \\ H_a = M_a \bar{c}_{pmean} T_a; \\ H_p = M_p \bar{c}_{pmean} T_p, \end{cases} \tag{6}$$

The values of the internal energy of the working medium at points "a" and "p" are calculated from the dependence:

$$U_a = M_a \bar{c}_{vmean} T_a \text{ and } U_p = M_p \bar{c}_{vmean} T_p. \tag{7}$$

Volume change work of the working medium in the filling process is calculated from the formula:

$$\int_{V_p}^{V_a} p\mathrm{d}V = \frac{p_a + p_{mean}}{2}\left(V_a - V_p\right), \tag{8}$$

whereas gas compression work at the end of the cylinder filling process from pressure p_{mean} to pressure p_a is determined from the dependence:

$$\int_{p_{\mathrm{mean}}}^{p_{\mathrm{a}}} p\mathrm{d}V = V_{\mathrm{a}}\left(p_{\mathrm{a}} - p_{\mathrm{mean}}\right). \tag{9}$$

After taking into account the dependences above, the first law of thermodynamics equation (2) takes on the following form:

$$Q_{\mathrm{w}}^{\mathrm{p\text{-}a}} + M_{\mathrm{i'}}\,\bar{c}_{\mathrm{pmean}} T_{\mathrm{i'}} = M_{\mathrm{a}}\bar{c}_{\mathrm{pmean}} T_{\mathrm{a}} - M_{\mathrm{p}}\bar{c}_{\mathrm{pmean}} T_{\mathrm{p}} - V_{\mathrm{a}}\left(p_{\mathrm{a}} - p_{\mathrm{mean}}\right). \tag{10}$$

Numbers of moles of the working medium at characteristic points marked in Fig. 1 are:

$$\begin{cases} M_{\mathrm{i'}} = \eta_{\mathrm{v}} \dfrac{p_{\mathrm{i}} V_{\mathrm{d}}}{\bar{R} T_{\mathrm{i}}}; \\ M_{\mathrm{a}} = \dfrac{p_{\mathrm{a}} V_{\mathrm{a}}}{\bar{R} T_{\mathrm{a}}}; \\ M_{\mathrm{p}} = M_{\mathrm{ex}} = \dfrac{p_{\mathrm{ex}} V_{\mathrm{ex}}}{\bar{R} T_{\mathrm{ex}}}, \end{cases} \tag{11}$$

where $p_{\mathrm{i}} = p_{\mathrm{i'}}$ and $T_{\mathrm{i}} = T_{\mathrm{i'}}$ – pressure and temperature of the working medium before the intake valve, respectively. The temperature at point “p” (Fig 1) is:

$$T_{\mathrm{p}} = T_{\mathrm{ex}} \left(\frac{p_{\mathrm{mean}}}{p_{\mathrm{ex}}}\right)^{\frac{n-1}{n}}, \tag{12}$$

where p_{ex} – is the leftover exhaust gas pressure, n – polytrophic exponent of expansion from pressure p_{ex} to pressure p_{mean}.

When differences in specific heats that occur in eq. (10) are disregarded and this equation is transformed, taking into account dependences (11) and (12), the following is obtained:

$$\eta_{\mathrm{v}} = \frac{T_{\mathrm{i}}}{T}\left[\frac{r_{\mathrm{c}}}{(r_{\mathrm{c}}-1)}\frac{p_{\mathrm{a}}}{p_{\mathrm{i}}} - \frac{1}{(r_{\mathrm{c}}-1)}\frac{p_{\mathrm{ex}}}{p_{\mathrm{i}}}\left(\frac{p_{\mathrm{mean}}}{p_{\mathrm{ex}}}\right)^{\frac{n-1}{n}} - \frac{r_{\mathrm{c}}}{(r_{\mathrm{c}}-1)}\frac{\bar{R}}{\bar{c}_{\mathrm{pmean}}}\frac{(p_{\mathrm{a}} - p_{\mathrm{mean}})}{p_{\mathrm{i}}} - \frac{\bar{R}}{\bar{c}_{\mathrm{pmean}}}\frac{Q_{\mathrm{w}}^{\mathrm{p\text{-}a}}}{p_{\mathrm{i}} V_{\mathrm{d}}}\right]. \tag{13}$$

The two last terms of the expression contained within the square brackets of the dependence above show the influence of the heat transfer between the working medium and the cylinder walls and of the work of the medium compression from pressure p_{mean} to pressure p_{a} on the cylinder-filling level. The impact of the above factors and also of the inflowing fuel-air mixture kinetic energy conversion to heat can be expressed with the temperature increase:

$$\Delta T = \Delta T_{\mathrm{w}} + \Delta T_{\mathrm{k}}, \tag{14}$$

where ΔT_{w} – is the temperature increment caused by heat transfer between the working medium and the cylinder walls [18], whereas ΔT_{k} – is the temperature increment caused by the change of the kinetic energy of the working medium flowing into the cylinder into heat, and the medium compression from pressure p_{mean} to p_{a} [19]:

$$\Delta T_{\mathrm{k}} = T_{\mathrm{i}}\left[\left(\frac{p_{\mathrm{a}}}{p_{\mathrm{mean}}}\right)^{\frac{k_{\mathrm{i}}-1}{k_{\mathrm{i}}}} - 1\right], \tag{15}$$

where k_{i} – isentropic exponent.

Having taken into account the dependences above, eq. (13) takes on the following form:

$$\eta_v = \frac{T_i}{T_i + \Delta T} \frac{1}{r_c - 1} \left[\frac{r_c p_a}{p_i} - \frac{p_{ex}}{p_i} \left(\frac{p_a}{p_{ex}} \right)^{\frac{n-1}{n}} \right]. \tag{16}$$

Values of leftover exhaust gas pressure were calculated from dependence [17]:

$$p_{ex} = p_{ou} \left(1 + a \frac{nS}{T_{ex}} \right). \tag{17}$$

In this empirical dependence, in accordance with [17], the coefficient that accounts for the effect of the resistance of the exhaust gas outflow through the valve is a = 0.3 to 0.5; n – crankshaft rotational speed, S – piston stroke, T_{ex} – temperature of leftover exhaust gases, p_{ex} – gas pressure in the IC engine exhaust system, which is calculated from the dependence:

$$p_{ou} = p_o (1 + \delta), \tag{18}$$

where, in accordance with [17], [20], $\delta = \frac{\Delta p_{ou}}{p_o} = 0.01 \text{ to } 0.03,$ is the relative resistance of exhaust gas flow through the engine exhaust system.

Pressure value at point “a” of the engine working cycle (Fig. 1) is calculated from the formula:

$$p_a = \frac{1}{2} \left(p_i + p_{mean} \right). \tag{19}$$

In order to calculate the amount of the working medium flowing into the cylinder, and then the mean value of cylinder pressure during the filling process, the balance equation of the working medium is used (2). Gas flow velocity in the minimal cross-section of the fuel jet at the intake valve is calculated from the dependence:

$$w_{mean} = \varphi w_t = \varphi \sqrt{2 R_i T_i \frac{\kappa_i}{\kappa_i - 1} \left[1 - \left(\frac{p}{p_i} \right)^{\frac{\kappa_i - 1}{\kappa_i}} \right]}, \tag{20}$$

where φ – velocity outflow coefficient, w_t – theoretical outflow velocity, R_i – air gas constant before intake valves, T_i – air temperature before intake valves, κ_i – adiabatic exponent, p_i – air pressure before intake valves, p – pressure in the engine cylinder.

Density of the air-fuel mixture before intake valves is calculated from the equation of state:

$$\rho_{mean} = \frac{p_{mean}}{R_{AFM} T_{mean}} \tag{21}$$

for $p_{mean} = p$, $T_{mean} = T_i \left(\frac{p}{p_i} \right)^{\frac{\kappa_i - 1}{\kappa_i}}$ and $R_{AFM} = R_i$,

where R_{AFM} – air-fuel mixture constant.

The time, in which the crankshaft rotates by the angle dα at the rotational speed n, is as follows:

$$\mathrm{d}t = \frac{\mathrm{d}\theta}{6n}. \tag{22}$$

Substituting the expressions above, which were employed to determine w_{mean}, ρ_{mean} and dt, in (3) and making transformations, the following is obtained [19]:

$$\delta M_{\mathrm{i}} = \frac{\sqrt{2R}}{\bar{R}6n}\left(\mu_{\mathrm{i}} f_{\mathrm{i}}\right) y_{\mathrm{i}} \frac{p_{\mathrm{i}}}{\sqrt{T_{\mathrm{i}}}} \mathrm{d}\alpha, \tag{23}$$

where $\mu_i = \alpha\varphi$ is the coefficient of the expenditure of outflow through the intake valves, in addition, α is the coefficient of the narrowing of the flowing gas stream, f_i is the geometric plane of the passage cross-section of intake valves, y_i is the function of the expenditure of the working medium flowing into the cylinder.

For the assumptions made above, the dependence for calculating the amount of the working medium flowing into the cylinder in the filling process is obtained [20]:

$$M_{\mathrm{i}} = \int_{\theta=0^\circ}^{\theta=180^\circ} \frac{\sqrt{2R}}{\bar{R}6n}\left(\mu_{\mathrm{i}} f_{\mathrm{i}}\right) y_{\mathrm{i}} \frac{p_{\mathrm{i}}}{\sqrt{T_{\mathrm{i}}}} \mathrm{d}\alpha = \frac{\sqrt{2R}}{\bar{R}6n}\left(\mu_{\mathrm{i}} f_{\mathrm{i}}\right)_{\mathrm{mean}} y_{\mathrm{imean}} \frac{p_{\mathrm{i}}}{\sqrt{T_{\mathrm{i}}}} = a_{\mathrm{i}} y_{\mathrm{imean}}, \tag{24}$$

where

$$a_{\mathrm{i}} = \frac{30\sqrt{2R}\left(\mu_{\mathrm{i}} f_{\mathrm{i}}\right)_{\mathrm{mean}} p_{\mathrm{i}}}{\bar{R}\sqrt{T_{\mathrm{i}}}\, n}, \tag{25}$$

$$y_{\mathrm{imean}} = \left(\frac{p_{\mathrm{mean}}}{p_{\mathrm{i}}}\right)^{\frac{1}{\kappa_{\mathrm{i}}}} \sqrt{\frac{\kappa_{\mathrm{i}}}{\kappa_{\mathrm{i}}-1}\left[1-\left(\frac{p_{\mathrm{mean}}}{p_{\mathrm{i}}}\right)^{\frac{\kappa_{\mathrm{i}}-1}{\kappa_{\mathrm{i}}}}\right]}. \tag{26}$$

The order of solving the system of equations used to compute the parameters of the end of the cylinder-filling process is as follows:

1) p_{mean} value is set and, in accordance with eq. (19), the first approximation of p_a value is computed;
2) value η_v is computed in accordance with dependence (16);
3) with the known value η_v, M_{Zd} is computed in accordance with the first formula of formulas in (11), and a_i is computed in accordance with formula (25), then value y_{imean} is computed from eq. (24);
4) when the computed value y_{imean} is known, pressure ratio (p_{mean}/p_i), and then value $p_{\mathrm{mean}} = p_{\mathrm{i}}\left(p_{\mathrm{mean}} / p_{\mathrm{i}}\right)$ are computed from eq. (26).

If the computed value p_{mean} does not agree with the assumed value, the computations are repeated with a new value p_{mean}, until the stabilisation of value p_{mean} is achieved. When value p_{mean} is known, pressure p_a is computed, and then the final value of cylinder-filling level η_v is computed from (16). Once the above computations are completed, it is necessary to compute:

$$\begin{cases} \gamma = \dfrac{p_{\mathrm{ex}} T_{\mathrm{o}}}{(\varepsilon-1)\eta_{\mathrm{v}} p_{\mathrm{o}} T_{\mathrm{ex}}}; \\[2ex] M_1 = \eta_{\mathrm{v}} \dfrac{p_{\mathrm{i}} V_{\mathrm{d}}}{\bar{R} T_{\mathrm{i}}}(1+\gamma); \\[2ex] T_1 = \dfrac{p_1 V_1}{\bar{R} M_1}. \end{cases} \tag{27}$$

The method for computing the cylinder volumetric efficiency provides a novel means of determining this quantity. The authors of the paper have not found such an approach in the literature on the subject available to them.

III. Experimental Stand

Experimental investigations were conducted at the engine test bench constructed at the Laboratory of Heat Engines of the Kielce University of Technology. The stand is composed of the following modules [19]:

- compression ignition FIAT 1.3 MULTIJET SDE 90KM engine,
- eddy-current brake of EMX – 100/10 000 type by ELEKTROMEX CENTRUM company,
- control cabinet for the engine and brake with the control system by AUTOMEX company,
- the system for taking measurements of the working medium pressure in the cylinder with GH13G12 sensor by AVL company,
- fuel charge meter of 730 Dynamic Fuel Consumption type by AVL company,
- thermal air flow meter SENSYFOLW IG by ABB company.
- PC that makes it possible to control the operation of the engine test bench with PARM 1.7 software by AUTOMEX company, and to diagnose the engine using KTS 540 module and Bosch company software.

The block diagram of the test bench with FIAT 1.3 MULTIJET SDE 90 HP engine is presented in Fig. 2. The control cabinet for the stand included AMX 202 brake power panel, AMX 211 module that controlled the engine-brake system, AMX212 PMO measurement module that made it possible to take measurements of the basic parameters that characterise the engine work and the panel for temperature and pressure measurements. The cabinet also comprised a panel for the control of 730 Dynamic Fuel Consumption AVL fuel charge meter and a panel with LUMEL RE43 temperature regulators.

The investigations involved FIAT 1.3 MULTIJET SDE 90 HP engine, manufactured by FIAT-GM POWERTRAIN Polska company in Bielsko Biała, which complied with Euro IV exhaust emission standard. Basic technical data for the engine under consideration is presented in Table I. The timing gear contains two camshafts. One of them is driven by the crankshaft via a single-strand chain, the other camshaft is driven by the first one via a cogbelt gear, located at the other end of the camshafts. The engine has four valves per cylinder, two intake ones and two exhaust ones. They have the deviation from the vertical of 3°, owing to which it was possible to position an injector between them. The valves are driven by two camshafts located in the head. One camshaft directly drives the Common Rail high pressure pump, the other camshaft drives the negative pressure pump. The cylinder bore is 69.6 mm in diameter, the distance between cylinders is 77 mm., which together with 82 mm of the piston stroke yields the swept volume of approx. 312 cm^3. Piston sweep-to-diameter ratio amounts to 1.18, which accounts for high thermal efficiency of the engine. In the engine intake system, a small VTG-type turbocharger with variable geometry stator vanes is used, which makes it possible to optimise the filling process within the whole range of the rotational speed. The intake system also contains the cooler for the air delivered to the cylinders.

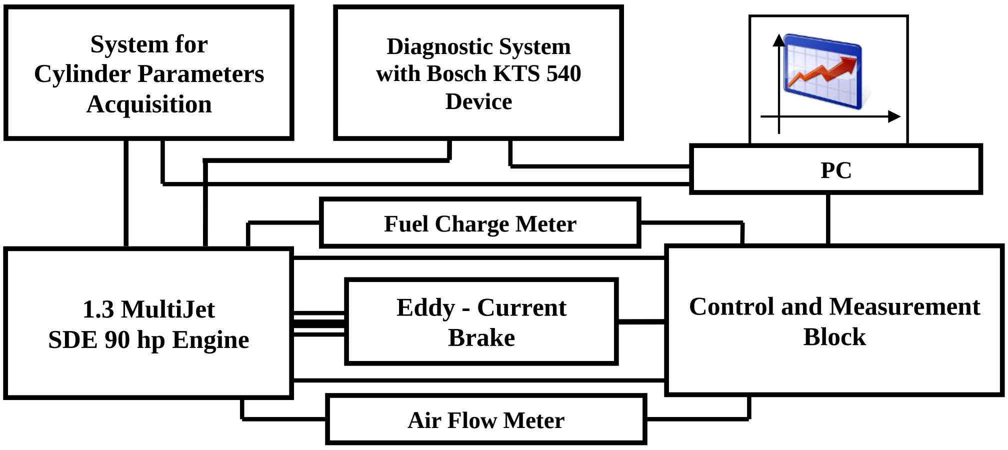

Fig. 2. Block diagram of the engine test bench.

TABLE I

BASIC SPECIFICATIONS FOR COMPRESSION IGNITION FIAT 1.3 MULTIJET SDE 90 HP ENGINE

Parameter	**Unit**	**Value**
Cylinder arrangement	–	in-line
Number of cylinders, c	–	4
Type of injection	–	direct, multi-stage fuel injection
Sequence of cylinder operation	–	1–3–4–2
Compression ratio, ε	–	17.6
Cylinder bore, D	m	$69.6 \cdot 10^{-3}$
Piston stroke, S	m	$82 \cdot 10^{-3}$
Engine displacement, V_{ss}	m^3	$1.251 \cdot 10^{-3}$
Maximum engine power, N_e	kW	66
Maximum power rotational speed, n_N	rpm	4000
Maximum torque, M_e	Nm	200
Maximum torque rotational speed, n_M	rpm	1750
Idle run rotational speed, n_{bj}	rpm	850 ± 20

IV. EXPERIMENTAL METHOD FOR DETERMINING VOLUMETRIC EFFICIENCY

At the engine test bench, FMT500-IG air flow meter by ABB company is installed before the air intake system of the 1.3 SDE 90 HP engine. The meter comprises a measuring sensor located in a tubular structure delivering air to the engine, and a display. Additionally, the reading of the air flow meter is taken by the PC-based system that controls the operation of the test bench. The air is delivered to the tubular structure via an additional air filter. The meter operates like a hot–film anemometer. The flow meter employs hot-film anemometer with continuous control of temperature difference and a platinum resistor, heated to a constant temperature, while the platinum sensor in the gas stream remains unheated. The heating power necessary to maintain the temperature depends on the flow velocity and gas properties.

When the gas composition is known, it is possible to specify the mass flux. The flow meter used for measurements makes it possible to record mass intensity of the amount of the air sucked up by the engine in kilograms per hour with a measurement error of ±0.9 %. The flow meter measuring range is 0 kg/h to 2000 kg/h.

For pre-determined conditions of engine operation at the test bench, which are set by a constant value of the rotational speed of the crankshaft and constant load maintained by the eddy-current brake, air consumption per hour by the engine $\dot{m}_a$, expressed in kilograms per hour, was measured. When the rotational speed of the crankshaft of four-stroke engine under investigation is known, it is possible to calculate the number of engine working cycles completed during one hour:

$$n_c = 30 n N_c, \tag{28}$$

where N_c – cylinder number.

Then, for pre-determined conditions of engine operation, it is possible to determine the actual amount of air delivered to one cylinder of the engine per a working cycle:

$$m_a = \frac{\dot{m}_a}{n_c}. \tag{29}$$

The theoretical amount of air that could be delivered to the engine cylinder under specified known operating conditions can be calculated from the state equation:

$$m_t = \frac{p_i V_d}{R_i T_i}, \tag{30}$$

where p_i – air pressure before intake valves, T_i – air temperature before intake valves, V_d – piston swept volume, R_i – air gas constant.

When values m_a and m_t are known, the value of volumetric efficiency η_{ve} is determined from the dependence:

$$\eta_{ve} = \frac{m_a}{m_t}. \tag{31}$$

V. Results Of Investigations And Calculations

The values of the selected performance parameters of the compression ignition FIAT 1.3 MULTIJET SDE 90 HP engine working at the test bench under full load characteristics are presented in Table II. The table also provides the values of the following work parameters of the engine under consideration: effective power P_e, effective torque T_o, fuel consumption per hour m_f, brake specific fuel consumption *bsfc,* fuel charge per engine working cycle g_c, air consumption per hour m_a, the mass of the air delivered to the cylinder during a single engine working cycle m_a, the volumetric efficiency determined on the basis of the results of experimental investigations η_{ve} and the volumetric efficiency calculated from the mathematical model of the engine filling process presented in the paper η_{vo}.

TABLE II

Performance Parameters of the FIAT 1.3 MULTIJET SDE 90 HP Engine Operating in Accordance With full Load Characteristics

Item	***n*, rpm**	**P_e, kW**	**T_o, Nm**	**m_f, kg/h**	***bsfc*, g/kWh**	**m_f, mg/cycle**	**m_a, kg/h**	**m_a, mg/cycle**	**η_{ve}**	**η_{vo}**
1	1000	8.41	80	2.46	292.51	20.50	40	333.33	0.843	0.963
2	1200	10.88	86	2.96	272.06	20.56	54	375.00	0.882	0.959
3	1400	16.20	110	4.22	260.49	25.12	74	440.48	0.849	0.953
4	1700	34.72	194	8.28	238.48	40.59	135	661.76	0.877	0.932
5	1800	37.92	200	8.88	234.18	41.11	144	666.67	0.883	0.910
6	2000	41.69	198	9.38	224.99	39.08	165	687.50	0.899	0.889
7	2200	46.32	200	10.07	217.40	38.14	180	681.82	0.898	0.878

8	2400	50.79	201	11.08	218.15	38.47	200	694.44	0.907	0.866
9	2600	55.57	203	12.02	216.30	38.53	214	685.90	0.899	0.857
10	2800	58.96	200	12.76	216.42	37.98	225	669.64	0.862	0.858
11	3000	59.38	188	12.81	215.73	35.58	246	683.33	0.905	0.835
12	3200	60.98	181	13.26	217.45	34.53	260	677.08	0.904	0.825
13	3400	63.74	178	13.98	219.33	34.26	271	664.22	0.899	0.812
14	3600	64.05	169	14.21	221.86	32.89	278	643.52	0.880	0.803
15	3800	64.43	161	14.25	221.17	31.25	286	627.19	0.878	0.792
16	4000	63.59	151	14.50	228.02	30.21	293	610.42	0.867	0.781
17	4200	61.02	138	14.92	244.51	29.60	296	587.30	0.849	0.769
18	4400	56.51	122	14.63	258.89	27.71	298	564.39	0.832	0.758
19	4600	53.76	111	13.64	253.72	24.71	303	548.91	0.820	0.748
20	4800	52.06	103	14.00	268.92	24.31	301	522.57	0.781	0.737

VI. Conclusion

The mathematical model of the process of engine cylinder filling presented in the paper makes it possible to determine the basic parameters of the filling process in a four-stroke piston internal combustion engine, including volumetric efficiency. The values of the coefficient determined using the above presented model were validated against experimental data from investigations into the FIAT 1.3 MULTIJET SDE 90 HP engine. On the basis of measurements of air consumption by the engine at the test bench, the values of the volumetric efficiency of the engine operating under full load characteristics were obtained. Satisfactory congruence between the volumetric efficiency determined with the use of the two methods described in the paper was achieved. That indicates that the mathematical model of volumetric efficiency presented in the paper is reliable. The method could be applied to calculations and analysis of working cycles of four-stroke piston internal combustion engines.

The method for volumetric efficiency calculations relies on a number of dependences between engine filling parameters, service conditions and engine design parameters. Thus, the method makes it possible to determine volumetric efficiency and analytically determine the relations between quantities that characterise the process of cylinder filling in piston internal combustion engines.

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Andrzej Ambrozik, engineering professor, graduated from the Faculty of Automotive and Construction Machinery Engineering at Warsaw University of Technology where, in 1970, he received his master's degree as a mechanical engineer with specialisation in automobiles and tractors. He dealt with issues related to cycles processing in piston combustion engines, primarily focusing on compression ignition, including: phenomenological thermodynamics and chemical thermodynamics of processes occurring in an engine cylinder, diagnostics and control of combustion process, efficient heat loss management using exegetic analysis, as well as ecological aspects and searching for alternative types of propellants. He is an author and co-author of 185 publications, five monographs and one patent.
Address: Kielce University of Technology, Department of Automotive Vehicles and Transportation, Aleja Tysiąclecia Państwa Polskiego 7, 25-314 Kielce, Poland.
Phone: +41 3424344
E-mail: silspal@tu.kielce.pl

Dariusz Kurczyński, Doctor of Engineering. A graduate of Kielce University of Technology, Faculty of Mechanical Engineering, field of study: Mechanics and Mechanical Engineering. In 1998, he received his master's degree in engineering. In 2007, he obtained the title of a doctor of technical sciences in construction and exploitation of machines with specialisation in piston combustion engines.
Current place of work: Kielce University of Technology. Faculty of Mechatronics and Machinery Design, department of Motor Vehicles and Transport.
Publications: author and co-author of 84 articles published in national and international conference materials and in scientific and technical journals.
Research interests: piston combustion engines, feed systems of combustion engines, functioning of combustion engines, alternative and conventional propellants used for powering engines, influence of piston combustion engines on the environment.
Address: Kielce University of Technology, Department of Automotive Vehicles and Transportation, Aleja Tysiąclecia Państwa Polskiego 7, 25-314 Kielce, Poland.
Phone: +41 3424332
E-mail: kdarek@tu.kielce.pl

Piotr Łagowski, Doctor of Engineering. A graduate of Kielce University of Technology, Faculty of Mechatronics and Machinery Design, field of study: Mechanics and Mechanical Engineering with specialisation in Automobiles and Tractors. He received his master's degree in engineering in 2002. In 2010, he obtained the title of a doctor of technical sciences in construction and exploitation of machines with specialisation in piston combustion engines.
Current place of work: Kielce University of Technology, Faculty of Mechatronics and Machinery Design, department of Motor Vehicles and Transport.
Publications: author and co-author of seventy-seven articles published in national and international conference materials and in scientific and technical journals, he is also a co-author of one monograph.
Research interests: Diagnostics and control of combustion process in a piston combustion engine using analysis of actual indicator graphs, including heat release characteristics during combustion process. Analytical and experimental research on the influence of types and properties (physical and chemical) of propellants and parameters of fuel injection process on energetic, economic and ecologic performance of engines.
Address: Kielce University of Technology, Department of Automotive Vehicles and Transportation, Aleja Tysiąclecia Państwa Polskiego 7, 25-314 Kielce, Poland.
Phone: +41 3424332
E-mail: p.lagowski@tu.kielce.pl