ASPECTS OF VOLUMETRIC EFFICIENCY MEASUREMENT FOR RECIPROCATING ENGINES

by

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The volumetric efficiency significantly influences engine output. Both design and dimensions of an intake and exhaust system have large impact on volumetric efficiency. Experimental equipment for measuring of airflow through the engine, which is placed in the intake system, may affect the results of measurements and distort the real picture of the impact of individual structural factors. This paper deals with the problems of experimental determination of intake airflow using orifice plates and the influence of orifice plate diameter on the results of the measurements. The problems of airflow measurements through a multi-process Otto/Diesel engine were analyzed. An original method for determining volumetric efficiency was developed based on in-cylinder pressure measurement during motored operation, and appropriate calibration of the experimental procedure was performed. Good correlation between the results of application of the original method for determination of volumetric efficiency and the results of theoretical model used in research of influence of the intake pipe length on volumetric efficiency was determined.

Key words: *internal combustion engine, flow measurement, power, volumetric flow rate, volumetric efficiency*

Introduction

Output engine parameters like power, torque, fuel consumption, *etc*. essentially depend on characters of processes that develop during exhaust and intake strokes. Design conception and dimensions of intake-exhaust engine system have a large influence over the flow processes in pipes and characters of both exhaust and intake processes development. Thus, in serial intake systems, up to 10% larger torque may be obtained by its optimization [1-3]. This fact is known from the early days of engines. Scientists have engaged in investigations of flow phenomena in intake-exhaust system since 1927, when Capetti [4] established a simple resonant wave action theory. The first calculations were based on determination of geometrical parameters of the intake system in order to achieve dynamical effects in pipelines at certain engine speeds [4]. In all these calculations, contribution to increase of volumetric efficiency could be determined only by experiment [5, 6].

With development of computer techniques, possibilities for using complex 1-D, 2-D or multi-dimensional models occurred, based on which, a number of programs for calculation of gas exchange process in engines, like PROMO-4 1-D and FIRE (3-D) were developed. These

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models are distinguished by the fact that improvement effects on engine (increase of volumetric efficiency, increase of torque …) may be determined directly from calculations [6-9]. Modern measuring methods (LDA – laser-Doppler anemometry, corona flow measurement, *etc*.) may be applied for experimental verification of the results [10].

Objectives of the paper are to summarize the results of own investigations in the subject area and to show the original method for indirect determination of volumetric efficiency. Conventional methods of measurement of volumetric efficiency are based on flow measurement devices that disturb the airflow and change dynamic phenomena in the engine intake-exhaust system. Based on own research of the intake airflow and pressure in the cylinder without combustion, the original method for determination of the volumetric efficiency was formed. The method enables the analysis of the influence of design characteristics of intake-exhaust system on volumetric efficiency values without disturbing the gas flow by measuring devices.

Since the research on multi-processing Otto/Diesel engine is conducted at the Faculty of Engineering from Kragujevac, special attention was given to the problems of determination of volumetric efficiency of the engine due to corresponding special conditions for volumetric efficiency variation in all engine operating regimes.

Measurement of airflow into the engine

Theoretically, mass of fresh charge in each cycle (m_{th}) is equal to the product of air density evaluated at atmospheric conditions outside the engine and volume displaced by the piston. However, because of the short cycle time available and the flow restrictions presented by the air cleaner, carburettor (if any), intake manifold and intake valve(s), less than this theoretical amount of fresh charge enters the cylinder. It is equal to the mass of fresh charge in one cylinder at the start of compression (m_{IVC}). Volumetric efficiency is defined as:

$$\eta_v = \frac{m_{\mathrm{IVC}}}{m_{\mathrm{th}}} \qquad m_{\mathrm{th}} = \rho_{\mathrm{o}} V_{\mathrm{dl}} \tag{1}$$

where η_v is the volumetric efficiency, m_{IVC} – the mass of fresh charge in one cylinder at the start of compression , m_{th} – the theoretical mass of fresh charge, ρ_{o} – air density evaluated at atmospheric conditions outside the engine, and V_{dl} – the volume displaced by the piston (one cylinder displacement).

The above definition is applicable only to the naturally aspirated engine. In the case of supercharged engine, however, the theoretical mass of fresh charge should be calculated for conditions of pressure and temperature prevailing in the intake manifold.

In real conditions, it is easier to measure air consumption level or, according to Jankov [11], fresh air or fresh mixture consumption coefficient in Otto engines. The air consumption level is defined as ratio between real mass airflow through the engine (Q_{m}) and theoretical mass airflow (Q_{mth}), which is calculated in regard to volume of the cylinder having environmental pressure and temperature.

The difference between the volumetric efficiency and the air consumption comes from the fact that the air consumption includes air losses in the cylinder before of the start of compression (first of all during valve overlap) while the mass of fresh charge in one cylinder at the start of compression does not include air losses in the cylinder. Therefore, the air consumption level is always higher than the volume efficiency. Since, in four-stroke engines, air losses in the cylinder are most frequently smaller than 4%, it may be concluded that there is no significant difference between the two parameters, so their difference is not emphasized and the same designation (η_v) is used.

The paper further deals with volumetric efficiency determined by experimental measurement of airflow through the naturally aspirated engine:

$$\eta_{\mathrm{v}} = \frac{Q_{\mathrm{m}}}{Q_{\mathrm{m\,th}}}$$
$$Q_{\mathrm{m\,th}} = \rho_o \frac{n}{2} V_{\mathrm{d}} \tag{2}$$

where Q_{m} is the actual steady-state mass airflow into the engine, $Q_{\mathrm{m\,th}}$ – the calculated mass airflow into the engine, ρ_{o} – the air density evaluated at atmospheric conditions outside the engine, n [rpm] – the engine speed, and V_{d} – the engine displacement.

Beside determination of volumetric efficiency, known airflow also enables calculation of average value of an excess air factor (λ), as well as of specific emissions of combustion products. In this paper, special attention is given to problems of measurement of airflow through the engine with sharp edged orifice plates. The basic prerequisite for precise measurement of flow with sharp edged orifice plates is steady-state flow through an orifice. On the other hand, piston IC engine, as cyclic machine, generates distinctive, unsteady – impulsive airflow through intake manifold, followed by corresponding dynamical phenomena. Since airflow-measuring device becomes a part of engine intake manifold, these phenomena also transfer to the measuring part of installation. By placing the air receiver tank between the engine and the measuring part of installation, a part of installation with large volume and cross-section of passage is obtained which, in mathematical sense, integrates the impulse function generated by the engine. In ideal case, a constant flow at tank's entrance or through the measuring part of installation would be obtained as the result. In general, large ratio between volume of air receiving tank and cyclic air consumption ratio and high frequency of flow excitation are in favour of the achievement of this goal.

Figure 1 shows a laboratory device for measurement of airflow, which consists of intake manifold, air receiver tank and measurement installation.

Intake manifold is made of PVC (poly-vinyl chloride) HTU DN 50 elements (standard EN1451) and it connects engine intake system to air receiver tank.

Air receiver tank has 200 L of volume and it is made of sheet metal barrel with corresponding attachments.

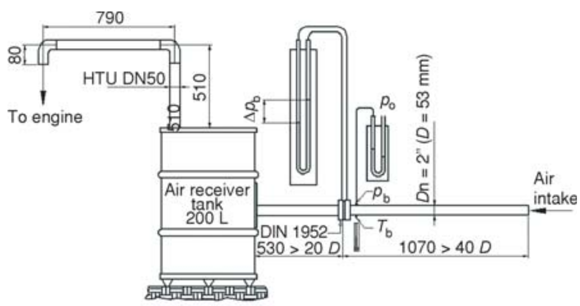

Figure 1. Scheme of device for airflow measurement (*D* – internal pipe diameter)

Basic part of the measuring installation is a sharp edged orifice plate located in a device with ring chambers for differential pressure measurement. All elements are dimensioned and made according to DIN1952 and ISO 5167-2 standards. Standard's recommendations regarding pipeline's length in front of and behind the orifice plate eliminate the hazard of occurrence of measurement errors related to flow disturbances due to change of cross-section of passage and pipeline configuration. Thereat, condition of steady-state flow is still valid.

Considering that the measuring principle itself causes irretrievable pressure drop in installation, several orifice plates with diameters of 10, 15, 20, 25, and 30 mm have been built. Selection of orifice plates was conducted in such way that, for minimal flows, pressure drop at orifice plates would be $\Delta p_b \geq 30$ mm H_2O and, for maximal flows, the irretrievable pressure drop would be smaller than 300 mm H_2O. Such criteria for selection of orifice plates provide satisfactory measurement accuracy with no significant influence on engine performance.

According to DIN1952 standard, volumetric flow (Q_v) and mass flow (Q_m) of compressible fluid are determined using the following equations:

$$
\begin{gathered}
Q_v = \alpha k_\varepsilon A_0 \sqrt{\frac{2\Delta p_b}{\rho}} \\
Q_m = \rho Q_v = \alpha k_\varepsilon A_0 \sqrt{2\rho \Delta p_b}
\end{gathered} \tag{3}
$$

where α is the coefficient of discharge, k_ε – the expansibility factor, A_0 – the cross-sectional area of orifice, Δp_b – the differential pressure on the orifice, and ρ – the air density.

The density can be evaluated from conditions upstream the orifice plate.

DIN1952 and ISO 5167-2 standards enable design of measuring device with sufficient accuracy, with no additional calibration. The main disadvantage of the procedure for flow calculation is complicated determination of orifice plate's coefficient of discharge $\alpha = f(\text{Re}, d/D)$ (interpolation of table data in function of Reynolds number and geometrical parameters of the orifice plate) and expansibility factor $k_\varepsilon = f(\Delta p_b/p_b, \kappa)$.

Consequence of earlier stated criteria for selection of orifice plates is their operation in tolerance limits, where coefficient of discharge of the orifice plate is $\alpha \neq$ const. On the other hand, due to small pressure drops at the orifice plates, expansibility factor (taking into consideration the compressibility of air) is $k_\varepsilon \approx 1$ (for $d/D < 0.5$ and $\Delta p_b < 400$ mm H_2O, $k_\varepsilon > 0.988$) (ISO 5167-2).

Calibration of airflow measuring device

In order to simplify the calculations and take into consideration other parameters influencing the measuring accuracy (imprecision machining of orifice plate's, roughness of measuring pipes …), calibration of all orifice plates was performed in mentioned measuring regimes. Thus, the eq. (3) for mass flow, Q_m, was transformed into the following form:

$$
\begin{gathered}
Q_m = \alpha k_\varepsilon A_0 \sqrt{2\rho \Delta p_b} = K_b \sqrt{\rho \Delta p_b} \\
K_b = \sqrt{2} \alpha k_\varepsilon A_0
\end{gathered} \tag{4}
$$

where K_b is the coefficient of orifice, Δp_b – the differential pressure on the orifice H_2O, and ρ – the air density evaluated from conditions upstream the orifice plate.

In this case, the coefficient of orifice (K_b) takes into account small changes of coefficient of discharge (α) and expansibility factor (k_ε) and it is determined during calibration procedure for each orifice plate separately.

Formula used during calibration was obtained from eq. (4) by solving it by K_b:

$$
K_b = \frac{Q_m}{\sqrt{\rho \Delta p_b}} \tag{5}
$$

Calibration gauge with "hot wire" for mass airflow, "DEGUSSA deguflow 8740-1313", was positioned at the entrance of the laboratory device. A four-stroke, four-cylinder Otto engine having 900 cm^3 of cubic capacity was used as flow generator. Minimal engine speed during calibration was 900 rpm, which means that the frequency of suction was always above 30 Hz.

Air density (ρ) was determined as follows:

$$\rho = \rho_0 \frac{p_\mathrm{b}}{p_0} \frac{T_0}{T_\mathrm{b}} = 1.225 \frac{p_\mathrm{b}}{1013.25} \frac{288.15}{T_\mathrm{b}} \tag{6}$$

where p_b is the air pressure upstream the orifice plate, T_b – the temperature of air upstream the orifice plate, ρ_0 – the air density at standard reference conditions (ISO 13443), 1.225, p_0 – the air pressure at standard reference conditions (ISO 13443), 1013.25 mbar, and T_0 – the air temperature at standard reference conditions (ISO 13443), 288.15 K.

The results of calibration of coefficient of orifice, K_b, are shown in fig. 2. It may be seen that coefficients of orifice plate have slightly linear decreasing trend with the increase value of differential pressure on the orifice plate (that is, with flow increase). This occurrence is more distinct at orifice plates having bigger cross-sectional area. Based on comparison of the results, it was concluded that empirically determined linear approximation using the least squares method of coefficient of orifice, $K_\mathrm{b} = \mathrm{f}(\Delta p_\mathrm{b})$, may sufficiently enough take into account variations of standard coefficient of discharge (α) and expansibility factor (k_ε), when $\Delta p_\mathrm{b} < 350$ mm H_2O.

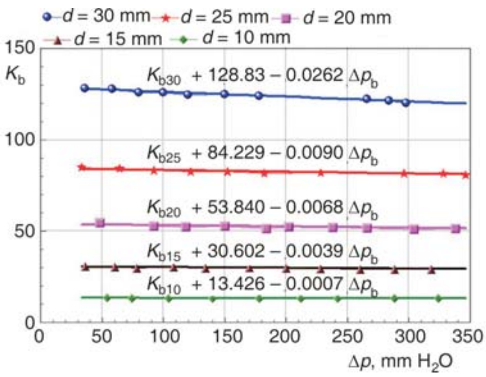

Figure 2. Results of calibration of coefficients of orifice plates (*d* – orifice plate diameter)

Thus, expressions from eq. (4) were further used to calculate the airflow, with the following coefficients of orifice:

$$K_\mathrm{b10} = 13.462 - 0.0007\Delta p_\mathrm{b}, \quad K_\mathrm{b15} = 30.602 - 0.0039\Delta p_\mathrm{b}, K_\mathrm{b20} = 53.840 - 0.0068\Delta p_\mathrm{b},$$
$$K_\mathrm{b25} = 84.229 - 0.0090\Delta p_\mathrm{b}, K_\mathrm{b30} = 128.83 - 0.0262\Delta p_\mathrm{b} \tag{7}$$

Expressions form eq. (7) are valid for 30 mm $H_2O < \Delta p_\mathrm{b}, < 350$ mm H_2O and for temperature of water in U tube manometer of 20 °C ± 5 °C. The number NN in index of K_bNN is denotes the diameter of the orifice given in mm.

Figure 3 shows charts of mass flow (Q_m) change in function of pressure drop (Δp_b) at the orifice plates, calculated according to eq. (4) and (7), for air density determined during calibration process (device calibration curves). In addition, discrete values measured on that occasion (calibration points) were entered into charts.

For small values of the differential pressure at the orifice (flow), there are distinguished "ends" in orifice characteristics. In this area, even small errors in the differential pressure on the orifice readings (differential manometer with water column) may lead to significant mea-

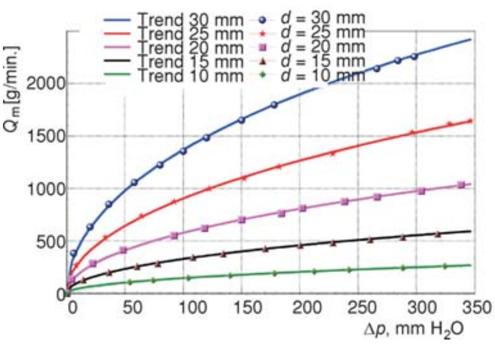

Figure 3. Calibration curves of airflow measuring device (*d* – orifice plate diameter)

surement error. It should be noted that the accuracy of reading of the water column height is 1 mm. There is a similar situation with application of modern differential pressure gauges with integrated electronics. All measurements in the area below 1/10 of nominal range are problematic due to inherent zero drift. Thus, it was adopted that lower limit of measurement range is determined by the following condition: $\Delta p_b \geq 30$ mm H_2O.

For large values of the differential pressure on the orifice (flow), characteristics are "linearized" and relative error of water column height reading becomes negligible. Unfortunately, there comes the increase of permanent pressure drop downstream the orifice plate. It was shown that permanent pressure drop (measured in air receiver tank) was typically 0.9 Δp_b. Thus, it was adopted that the upper limit of measurement range of the orifice plate is determined by the following condition: $\Delta p_b \leq 350$ mm H_2O.

Particularity of air mass flow measurement on multi-processing Otto/Diesel engine

Conventional Otto engine with premixed charge has low efficiency at partial loads and low particulate matter (PM) and NO_x emissions (stoichiometric homogenous air-fuel mixture and three-way catalytic converter + Exhaust gas re-circulation) at high-loads. Conventional Diesel engine with non-premixed charge has high efficiency and high PM emission at full-load and low PM emission at low-loads. An ideal engine, which has the best characteristics of both Otto and Diesel engines, is multi-processing Otto/Diesel engine. At low loads and idling, the engine operates with compression ignition of inhomogeneous air-fuel mixture and without a throttling (Diesel engine), while at full load and at higher loads, the engine operates with spark ignition of homogenous air-fuel mixture and with a throttling (Otto engine) [12].

Volumetric efficiency measurements on multi-processing Otto/Diesel engine were conducted on experimental engine developed at the Laboratory for IC engines of the Faculty of Engineering from Kragujevac [13, 14]. The experimental engine is the result of the reconstruction of the basic DI-Diesel – 3 LDA 450 engine, with basic characteristics given in tab. 1.

Table 1. Basic engine data

Manufacturer	"21 May Belgrade"
Type	DI-Diesel – 3 LDA 450
Number of cylinder	1
Bore/stroke	85 mm/80 mm
Engine displacement	454 cm^3
Compression ratio	$\varepsilon = 17.5$
Power	7.3 kW
Engine speed	3000 rpm
Torque	28 Nm
Intake valve opening (IVO)	16° CA bTDC
Intake valve closing (IVC)	40° CA aBDC
Exhaust valve opening (EVO)	40° CA bBDC
Exhaust valve closing (EVC)	16° CA aTDC

During preliminary tests, it was determined that experimental multi-processing engine obtains flows from 36 g/min (Otto mode at idle speed) to 660 g/min (Diesel mod at maximal engine speed).

In addition, it was established that the static pressure variation upstream the orifice plate is almost immeasurable. Relatively large internal pipe diameter (D = 53 mm) and small length of the measuring pipeline upstream the orifice plate, provided that there were no significant stream losses at this section, even at maximal flows. Therefore, during testing on multi-processing engine, it was adopted that absolute air pressure upstream the orifice plate is equal to environmental pressure.

Total flow resistance on airflow measuring device may be maintained at levels of resistance in factory air cleaners (not being used at this occasion) with appropriate selection of the orifice plates.

Things are quite different regarding dynamical phenomena. Large differences in lengths and diameters of the intake manifold, as well as differences in volumes of devices with respect to factory intake system, certainly had influenced the engine performance. In order to quantify these effects, determination of speed characteristics of volumetric efficiency (η_v) was conducted in Diesel mode (fired), using the device shown in fig. 1, with the two orifice plates (15 and 20 mm). Figure 4 shows measurement points (spline interpolation) and trends of variation (third order polynomial) of the obtained results.

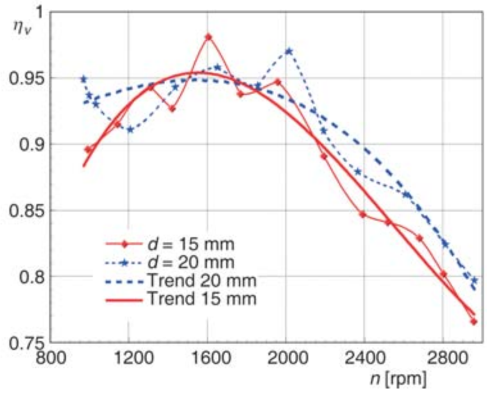

Figure 4. Orifice plate diameter (*d*) and volumetric efficiency (η_v)

Comparison of trends of variation of volumetric efficiency (η_v) shows the following:

- maximal values, in both cases, are obtained around 1600 rpm,
- with the increase or decrease of engine speed from 1600 rpm the volumetric efficiency decreases more rapidly when orifice plate of 15 mm is used (large permanent pressure drop caused by orifice plate), and
- the largest differences in the volumetric efficiency are found below 1250 rpm.

Differences in measured values of flow, specially the amplitudes bias of individual measuring points from trends of variation (up to 8%) may be attributed to appearance of standing waves and resonances of individual parts of measuring device, as well as to insufficient stiffness of the walls of the air receiver tank. During tests, the frequency of suction was in the range of approximately 7-24 Hz, while the cyclic quantity of air was always maximal. Thus, flow pulsations were transferred in smaller amount to measuring part of installation, in spite of the fact that the volume of the air receiver tank was 440 times larger than the engine capacity.

The general conclusion is that, in view of accuracy, the most unfavourable conditions of flow measurement are at low engine speeds, in Diesel mode. Thus, an orifice plate of 15 mm was used for testing the engine in Diesel mode and in most of the regimes in Otto mode. An orifice plate of 10 mm was used only for idle speed regimes and low engine loads in Otto mode.

Indirect determination of volumetric efficiency

Experimental determination of volumetric efficiency during optimization of intake-exhaust system is a specific problem. In order to establish the real influence of designs of these systems, it is necessary to use measuring instruments that do not interfere with airflow (LDA, Corona flow measurement, *etc*.).

The other solution is indirect determination of volumetric efficiency by using a difference of pressure at the end of compression and pressure at the start of compression during compression stroke, when engine was motored. These conditions may be achieved either by application of external drive (DC motor) or by short-term cutting out of ignition and/or injection (only in one or in several cylinders) at selected engine operating regime. This paper will present the first variant, where experimental installation shown in fig. 5 was used, while the engine characteristics are given in tab. 1 [13, 14].

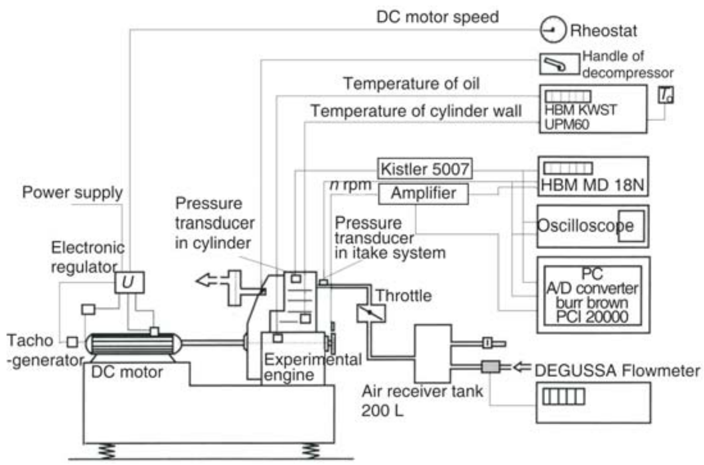

Figure 5. Scheme of experimental installation

Dependence between pressure at the end of compression and volumetric efficiency may be determined analytically, by application of the basic equation of engine thermodynamics [10], as it follows from expressions:

$$\begin{aligned} p_1 &= \frac{1}{\kappa(\varepsilon-1)+1}\left[p_\mathrm{r} + p_\mathrm{o}\kappa(\varepsilon-1)\eta_v \frac{T_\mathrm{o}+\Delta T}{T_\mathrm{o}}\right] \\ p_2 = p_1\varepsilon^{n_1} &= \frac{1}{\kappa(\varepsilon-1)+1}\left[p_\mathrm{r} + p_\mathrm{o}\kappa(\varepsilon-1)\eta_v \frac{T_\mathrm{o}+\Delta T}{T_\mathrm{o}}\right]\varepsilon^{n_1} \end{aligned} \tag{8}$$

where p_o is the environmental pressure, p_1 – the pressure at the start of compression, p_2 – the pressure at the end of compression, p_r – the pressure of exhaust gas residual, T_o – the environmental temperature, ΔT – the temperature increase of the fresh charge, ν_1 – the polytropic exponent for compression, κ – the ratio of specific heats for air (1.41), and ε – the compression ratio.

In conditions of external engine drive, operation with no combustion and with standard intake-exhaust system, the following values may be adopted: $n_1 = 1.375$, $p_\mathrm{r} = 1.2$ bar, $p_\mathrm{o} = 1.0$ bar, $T_\mathrm{o} = 293$ K, and $\Delta T = 10$ K. When these values are entered into eq. (8), the following theoretical linear dependence is obtained:

$$p_2 - p_1 = 2.48 + 49.76\eta_v \tag{9}$$

Corresponding measurements at several different engine speeds were conducted at the Laboratory for IC engines of the Faculty of Engineering from Kragujevac. In stationary regime of the engine, airflow through the engine was varied using a throttle valve at the entrance of the intake manifold, fig. 5. At each partial closing of the throttle valve, corresponding airflow through the engine and the cylinder pressure were measured (for determination of difference of pressures, $p_2 - p_1$). The airflow was measured by device shown in fig. 1, using the orifice plate diameter of 20 mm. Based on known flow, mass of fresh charge that enters the engine was obtained and, then, volumetric efficiency was determined from eq. (2). Experiments were conducted for engine speed range from 2000 rpm to 2800 rpm with the step of 200 rpm. For each engine speed, one dependence curve was obtained, fig. 6. Curves are almost linear and close to each other. Thus, it is possible to set the corresponding trend by using the least squares method (trend). Trend equation, acquired by experiment, is:

$$p_2 - p_1 = 7.13 + 41.86\eta_v$$
$$\eta_v = \frac{p_2 - p_1 - 7.13}{41.86} \tag{10}$$

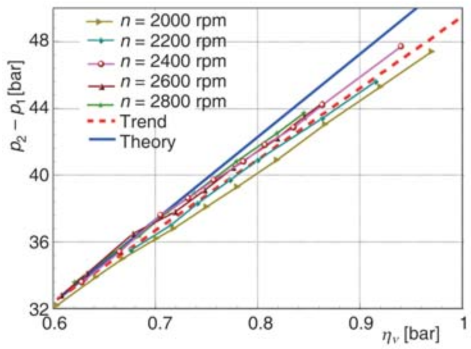

Figure 6. Difference of pressures in the cylinder and volumetric efficiency

Equation (10) may be used for indirect measurement of volumetric efficiency in investigations of the influence of design parameters of intake-exhaust system. Coefficients of this equation are determined with standard intake system and classic intake airflow gauges. Afterwards, devices for intake airflow measurement are removed and cylinder pressure transducer is maintained. Now, influence of design parameters of the intake-exhaust system on volumetric efficiency may by analyzed, without disturbances from flow measuring devices.

Figure 6 presents theoretical dependence between difference of pressures ($p_2 - p_1$) and volumetric efficiency, defined by eq. (9). It may be seen that theoretical values of difference of pressures are larger than experimentally determined values, for the same value of volumetric efficiency. In other words, for one value of difference of pressures, the theoretical dependence gives smaller values of volumetric efficiency than experiment. The reason for this is that experimental values of volumetric efficiency contain corresponding losses of fresh charge, first of all during valve overlap (this engine has relatively large valve overlap of 32° CA). The difference between theoretically and experimentally obtained volumetric efficiency is increasing with the increase of the airflows through the engine. Also, the difference is increasing with the decrease of the engine speeds, because the losses during valve overlap are higher. It may be said that values of theoretically determined volumetric efficiency are more suitable to the original definition (when they are determined by the quantity of the fresh charge in the cylinder at the beginning of compression), while experimentally determined values are more suitable to air consumption level because they were determined in that manner.

From fig. 6, it may be seen that experimentally determined difference of pressures ($p_2 - p_1$) has larger values than theoretically determined value at low values of volumetric efficiency. This is explained by lower pressures in the cylinder during valve overlap, when, due to higher pressure in the exhaust system, gas from the exhaust system is carried back into the cylinder. It results in much larger quantity of gas in the cylinder at the beginning of compression and accordingly higher pressure of compression. This does not mean that the actual volumetric efficiency, determined by this method using difference of pressures in the cylinder, is larger because there is no large quantity of fresh charge in the cylinder, but more exhaust gases. This is important for analysis of gas exchange process at Otto engine at low loads and at low engine speeds.

Comparison between calculated and measured volumetric efficiency

Dynamic unsteady flows in intake manifolds of internal combustion engine have already been subject of numerous contributions. Pressure wave produced by the piston and valves motion have a significant effect on the cylinder gas exchange and, consequently, on the volumetric efficiency [13-16].

Experimental installation in fig. 5 was also used for analysis of dynamic effects in intake/exhaust system of the engine. Accordingly, pressure in the intake manifold directly above the intake valve was measured. In addition to experimental analysis of the dynamic effects in the intake/exhaust system, corresponding theoretical analysis was also conducted using PROMO-4 software. Calibration of theoretical model (determination of the valve coefficients of flow) was done for the case of engine having standard intake and exhaust systems without flow measuring devices. The flow measuring device was removed in order to avoid its influence on dynamics in intake-exhaust system of the engine. Theoretical and experimental researches were conducted after calibration. These researches were conducted on experimental engine with standard intake-exhaust system replaced by pipes with corresponding diameters and lengths. Based on these investigations, it may be concluded that the most important characteristics of the flow of low pressure in the intake pipe are the amplitude and angular position of the maximum pressure during the intake phase, namely, the position of maximum pressure in relation to the IVC moment. Also, wave phenomena in the exhaust system do not significantly influence volumetric efficiency of engines, especially when the valve overlap is small. The character of flow in the intake /exhaust system depends on its geometrical parameters. If the valve overlap is small, oscillations of pressure in the exhaust pipe do not affect the exchange process of the working substance, *i. e*., the volumetric efficiency. Also, the size of flow losses in the exhaust pipe has no significant influence on the volumetric efficiency. The oscillations of pressure in the intake pipe significantly affect the volumetric efficiency [15, 16].

Figure 7(a) shows values of volumetric efficiency of the experimental engine with standard intake-exhaust systems for different engine speeds. Experimental values are gained using eq. 10, which is new original method, while theoretical values were calculation results obtained by PROMO-4 software. It may be noticed that both curves have the same shape and that experimental values are somewhat smaller. The difference between experimental and theoretical values increases with decrease of engine speed. The cause was found in deviation of experimental from theoretical pressure course in the intake manifold immediately before IVC [13]. These values may not be compared with those from fig. 4, because there is no influence of flow measuring device.

Values of volumetric efficiency of the experimental engine for three different lengths of the intake manifold (300 mm, 600 mm, and 1500 mm) and for different engine speeds were shown in fig. 7(b), fig. 7(c) and fig. 7(d). In these cases also, curves have the same shape and experimental values are somewhat smaller. Also, the difference between theoretical and experimental values is somewhat larger at lower engine speeds.

Oscillations caused by the intake process do not stop when the intake valve is closed. Backward wave remains for the next intake process which has a positive or negative effect. The frequency of these oscillations is equal to the frequency of oscillation of unilaterally closed pipe with slightly decreasing amplitude (damped oscillations).

At low engine speeds, residual wave comes to a complete damping inside the intake pipe so it has no influence on the next intake process. Also amplitude of pressure oscillations are small – for the longest tube (1500 mm) at 1000 rpm the amplitude of pressure is only 0.013 MPa [13]. Intake phase has long duration time, so during the intake phase there are more minimums and maximums of the pressure. Residual oscillations in the intake pipe, after the intake phase, are highly frequented and thus quickly damped. Therefore, at lower engine speeds (1000 and

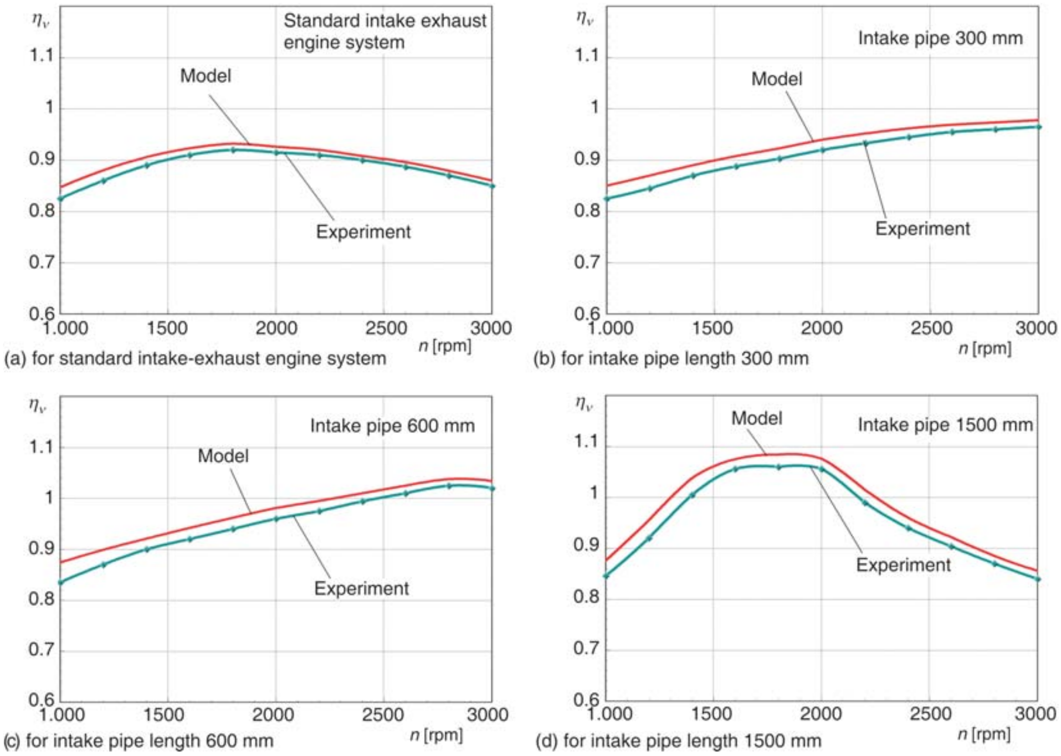

Figure 7. Values of volumetric efficiency obtained by new experimental method and model

1200 rpm) modified lengths of the intake pipe have a small effect on the change in the volumetric efficiency, fig. 8.

With increasing the engine speed, the influence of the length of the intake pipe on the volumetric efficiency also increases. At 1500-2500 rpm, the effects of supercharge in long pipes (1500 mm, 1200 mm, and 900 mm) are noted. The values of the volumetric efficiency ranged around and above 1. The volumetric efficiency has a pronounced maximum due to the stronger supercharge effect induced by oscillations in pipes, fig. 8.

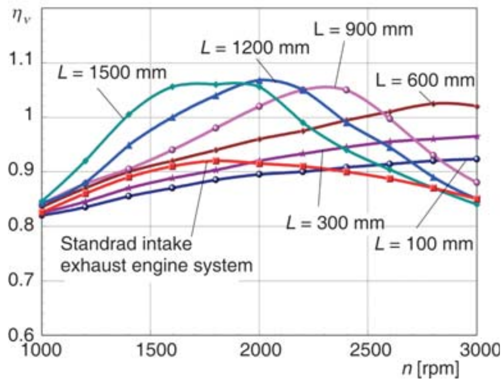

Figure 8. Volumetric efficiency for different length of the intake pipe (experimental data)

Figures 9(a) and 9(b) show examples of intake pipe relative pressure (compared to the environmental pressure $p_o = 1.0$ bar) oscillations at the intake valve location, for pipe length of 1500 mm and for the two engine speeds: 1600 rpm and 2400 rpm, respectively. From the diagrams, it may be noticed that there is larger amplitude at higher engine speed fig. 9(b), than at lower engine speed, fig. 9(a), while volumetric efficiency is smaller, fig. 7(d). The reason for this is inadequate moment of IVC, fig. 9(b).

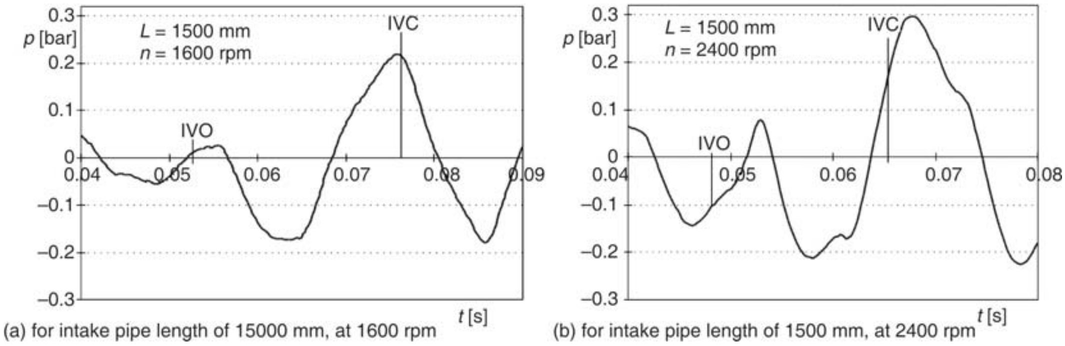

Figure 9. Relative pressure in the intake pipe before the intake valve (experimental data)

With increasing length of pipe and the engine speed, the amplitude of oscillation increases. With increasing amplitude, volumetric efficiency increases if closing moment of the intake valve is tuned.

Conclusions

The problems of experimental determination of intake airflow using orifice plates and of influence of orifice plate diameter on the results of the measurements were investigated. The problems of measurements of airflow through a multi-process Otto/Diesel engine were analyzed. Original method for determination of the volumetric efficiency based on measurement of pressure in the cylinder without combustion is developed and appropriate calibration of experimental method has been conducted. The results of application of the original method compared to the results of the theoretical model, during research of the influence of the intake pipe length on volumetric efficiency, have been also shown. The following conclusions can be derived.

- The basic prerequisite for precise measurement of flow with sharp edged orifice plates is the steady-state flow through an orifice. On the other hand, piston IC engine, as a cyclic machine, generates distinctive, unsteady – impulsive airflow in intake manifold, followed by corresponding dynamical phenomena. Since airflow-measuring device becomes a part of engine intake manifold, these phenomena also transfer to the measuring part of installation. By placing the air receiver tank between the engine and the measuring part of installation, a part of installation with large volume and cross-section of passage is obtained, which, in mathematical sense, integrates the impulse function generated by the engine. In ideal case, a constant flow at the tank's entrance or through the measuring part of installation would be obtained as the result.
- Large differences in lengths and diameters of the intake manifold equipped with the measurement devices, as well as differences in volumes of measuring devices with respect to factory made intake system, had influenced the dynamical phenomena in the engine intake-exhaust system. Corresponding differences between measured values of flow through the engine were registered for different orifice plate diameters. Differences in measured values of flow, specially the amplitudes bias of individual measuring points from trends of variation (up to 8%) may be attributed to the appearance of standing waves and resonances of individual parts of the measuring device, as well as to insufficient stiffness of the walls of the air receiver tank.
- When dynamical phenomena in intake-exhaust system of the engine are analyzed, devices for airflow measurement must not disturb dynamical processes in the system. One of possible solutions is indirect determination of volumetric efficiency by measurement of

pressure in the cylinder without combustion. The difference between theoretically and experimentally obtained volumetric efficiency is increasing with the increase of the airflows through the engine. Also, the difference is increasing with the decrease of the engine speeds, because the losses during valve overlap are higher.

- Diagrams of volumetric efficiency of the experimental engine for the three different lengths of the intake manifold (300 mm, 600 mm, and 1500 mm) and for different engine speeds have the same shape and experimental values are somewhat smaller. Also, the difference between theoretical and experimental values is somewhat larger at lower engine speeds. At low engine speeds, residual wave comes to a complete damping inside the intake pipe so it has no influence on the next intake process. At lower engine speeds (1000 and 1200 rpm), modified length of the intake pipe have a small effect on the change in the volumetric efficiency. With the increase of the engine speed, the influence of the length of the intake pipe on the volumetric efficiency also increases.
- At 1500-2500 rpm, the effects of supercharge in long pipes (1500 mm, 1200 mm, and 900 mm) are noted. The values of the volumetric efficiency ranged around and above 1. The volumetric efficiency has a pronounced maximum due to the stronger supercharge effect induced by oscillations in pipes.

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Nomenclature

A_0 – cross-sectional area of orifice, [m^2]
CA – crank angle, [°]
D – internal pipe diameter, [mm]
d – orifice plate diameter, [mm]
EVC – exhaust valve closing, [°]
EVO – exhaust valve opening, [°]
IVO – intake valve opening, [°]
IVC – intake valve closing, [°]
K_b – coefficient of orifice, [–]
k_ε – expansibility factor, [–]
L – pipe length, [mm]
m_{IVC} – mass of fresh charge in one cylinder at the start of compression, [kg]
m_{th} – theoretical mass of fresh charge, [kg]
n – engine speed, [rpm]
n_1 – polytropic exponent for compression, [–]
p_b – air pressure upstream the orifice plate, [Pa]
p_o – environmental pressure, [Pa]
p_r – pressure of exhaust gas residual, [Pa]
p_0 – air pressure at standard reference conditions (ISO 13443), [mbar]
p_1 – pressure at the start of compression, [Pa]
p_2 – pressure at the end of compression, [Pa]
Q_m – actual steady-state mass airflow into the engine, [gmin^{-1}]
$Q_{m\,th}$ – calculated mass airflow into the engine, [gmin^{-1}]
Q_v – volumetric flow, [m^{3}s^{-1}]
Re – Reynolds number, [–]
T_b – temperature of air upstream the orifice plate, [K]
T_o – environmental temperature, [K]
T_0 – air temperature at standard reference conditions (ISO 13443), [K]
V_{d1} – volume displaced by the piston (one cylinder displacement), [m^3]
V_d – engine displacement, [m^3]

Greek letters

α – coefficient of discharge, [–]
Δp_b – differential pressure on the orifice, [Pa]
ΔT – temperature increase of the fresh charge, [K]
ε – compression ratio, [–]
η_v – volumetric efficiency, [–]
κ – ratio of specific heats for air [–] (1.41)
λ – excess air factor, [–]
ρ – air density, [kgm^{-3}]
ρ_o – air density evaluated at atmospheric conditions outside the engine [kgm^{-3}]

Acronyms

LDA – laser-Dopler anemometry
BDC – bottom dead centre
TDC – top dead centre

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