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# NUMERICAL SIMULATION OF THE EFFECT OF INTAKE PIPE LENGTH ON THE AIR MASS REMAINING IN COMPRESSION-IGNITION ENGINE CYLINDER AFTER INLET VALVE CLOSURE

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**Summary.** The paper presents a general description of numerical model for combustion engine intake system. A one-dimensional model was developed, allowing for energy conservation law. Model computing possibilities have been presented, as well as the analysis of simulation results for the effect of intake pipe length on the air mass remaining in the engine cylinder after inlet valve closure.

**Key words:** intake system, numerical model, cylinder charge mass, simulation

## INTRODUCTION

A piston combustion engine works according to the appropriate thermal cycle. These cycles are continually repeated during an engine's operation. At each thermal cycle a charge exchang as well as compression and expansion processes occur. Charge exchange in a piston combustion engine consists in removing the residues from the engine cylinder after a preceding thermal cycle and delivering a fresh charge into it. The process of delivering a fresh charge is usually called the filling process [1] [5] [8] [9] [14].

The filling process is induced by a piston movement from top dead centre (TDC) to bottom dead centre (BDC). At that time, a negative pressure is being produced in the cylinder owing to which the charge flow from environment to the engine cylinder is possible. It should be noted here that the charge flow is accompanied by a number of phenomena, usually unfavourable, which cause a smaller cylinder filling with a fresh charge than it would result from displacement volume.

Construction solutions of the intake system, dimensions of its components, manufacture accuracy, surface condition and distribution of system components may fundamentally affect engine operation parameters, such as power, torque and, in particular, its rotational speed characteristics, fuel consumption and engine response. The aforementioned parameters are important from the point of view of vehicle user because they influence motor-car traction properties and fuel economy of driving. The intake system may also affect the emission of exhaust gas toxic components, such as carbon and nitrogen oxides, hydrocarbons and particulate solids.

In order to assure a sufficient efficiency of the filling process by delivering a suitable mass of charge (air) into engine cylinders without the necessity of using additional supercharging devices (compressors), geometric charge exchange system parameters and timing gear system parameters should be properly matched.

The most important parameters of the intake system should include:
- intake pipe length,
- intake pipe cross-sectional area,
- intake pipe shape.

Due to the fact that the course of charge vibrations in the intake pipes is a time function, intake system parameters should be matched allowing for a specific frequency of charge vibration constraints in the intake pipe. The frequency of vibration constraints depends, however, on the speed of engine crankshaft and therefore geometric intake system parameters are always matched for a specific engine rotational speed or a range of engine rotational speeds.

The intake system parameters can be determined as a result of engine test bed examinations. These examinations are carried out by the method of successive approximations and hence their execution is laborious and expensive. Therefore, application of the methods of modelling dynamic phenomena course in the charge exchange process seems to be purposeful.

In order to develop a complex model of combustion engine, it is necessary to consider the following processes:
- charge exchange,
- compression,
- air-fuel mixture formation and combustion,
- expansion.

Therefore, questions referring to wave phenomena in the intake and outlet systems and connected with the formation and combustion of air-fuel mixture in engine cylinder should be solved. These questions should be solved at the same time, which makes a computational model to require – due to its complexity – application of computing tools with a very large capacity. For this reason, computational models of the charge exchange process are developed using specific simplifying assumptions. The applied simplifications significantly affect the computation speed and the faithfulness of the simulated modelling processes.

## NUMERICAL MODEL OF THE ENGINE FILLING PROCESS

The so-far-developed models of phenomena in engine intake systems and the computational models of charge exchange process can be classified into one of the four groups that differ in the scope of adopted simplifications [13]:
- zero-dimensional models,
- one-dimensional models, not allowing for energy conservation law, based on the acoustic theory,
- one-dimensional models, allowing for energy conservation law,
- multi-dimensional models.

The zero-dimensional model does not allow for the selection of intake system parameters for a required efficiency of dynamic supercharging. In more complex zero-dimensional models, the model is used for determining the parameters of a medium in total cylinder volume under defined conditions of flow through inlet an outlet valves [2][7].

One-dimensional method allowing for energy conservation law was developed by Seifert [12]. This method is more commonly known under the name of PROMO approach. For computations of dynamic processes in the exchange process, the PROMO method uses basic principles of mechanics. They allow to describe a general case of non-stationary gas flow through the charge exchange system. Based on them, it is possible to derive a system of non-homogeneous quasi-linear differential equations that enable analysis of the course of wave phenomena in the intake or the outlet system. This method allows for charge wall friction, heat exchange with environment and intake pipe cross-sectional area change. Equations in the PROMO method are solved using the finite difference method [3]. The results of simulation examinations obtained through the application of the PROMO method are concurrent with those of engine test bed examinations. Its application allows for the selection of geometric intake system parameters with sufficiently great accuracy, in particular in case of systems with individual intake pipes.

A method most accurately modelling the phenomena in the engine intake system is the multi-dimensional method. It is based on non-stationary gas flow equations made for a two-dimensional or three-dimensional solution of the model of dynamic phenomena in combustion engine charge exchange system. The group of multi-dimensional methods includes a package of AVL BOOST computer simulation programmes from AVL List GmbH. Programmes being part of the package enable execution of complex calculations for the whole cycle of two-stroke or four-stroke engine, particularly for the process of charge exchange both in unsupercharged and supercharged engine. The package enables three-dimensional calculations together with visualisation of flow phenomena in the intake system, engine cylinder and the outlet system. In the simulations being carried out, it is possible to take into account the presence of all elements of the intake and outlet system, i.e. air filter, compressor or turbo-compressor, charge air cooler, outlet [exhaust] pipe, exhaust silencer and catalyst. The equations allow for heat exchange with environment, charge wall friction and system spatial dimensions. A shortage of the AVL BOOST package is its high price and high requirements with regard to the computer speed and its memory capacity.

When selecting simulation methods, one should not forget that the final selection of geometric charge exchange system parameters is possible only as a result of engine test bed examinations performed with the charge exchange system – most frequently a prototype – under examination due to the fact that the full modelling of actual processes in simulation models is impossible.

In his further considerations, the author examined a one-dimensional model allowing for energy conservation law.

Air flow in the intake system can be described with basic equations which were formulated for a general fluid model and resulting from three fundamental laws of mechanics, i.e. [11]:

- mass conservation law:

$$\frac{\partial \rho}{\partial t} + div(\rho u) = 0, \qquad (1),$$

- momentum and angular momentum conservation law:

$$\rho \frac{du}{dt} = \rho F + divS, \qquad (2),$$

- and energy conservation law:

$$\rho \frac{d}{dt}\left(T C_v + \frac{u^2}{2}\right) = \rho F u + \rho q + div(\Gamma \, grad \, T) + div(S u), \qquad (3),$$

where: t – time, ρ – density, S – stress tensor, T – temperature, Γ – thermal conductivity, u – speed, F – body force, $C_v$ – specific heat at constant volume, q – specific yield of internal heat source.

During the process of cylinder filling with air, physical phenomena are divided into two groups. Within the first groups, an elastic flow through the intake pipe is examined and described by equations (1), (2), and (3). The second group refers to heat phenomena in the cylinder. When analysing the air flow through the engine intake system, phenomena in engine cylinder are described with zero-dimensional models which ignore a movement of charge in the cylinder space. Hence, momentum and angular momentum conservation law is not allowed for. Also mass conservation and energy conservation laws can be presented in a simplified form.

When developing a physical model of engine cylinder filling process, a flow of charge through four systems connected in a series was assumed, i.e. environment, intake pipe, cylinder, and environment.

As it can be seen, the first and the last system out of the aforementioned ones is the same system, characterised by the same parameters (ambient temperature $T_a$, pressure $p_a$ air density $\rho_a$) which are constant both in time and space.

The above parameters refer to conditions that are present prior to the intake pipe. When looking upon the intake pipe as the first system element, it is possible to assume that every device found before the intake pipe will be considered as environment. Hence, the parameters of a charge on the exit from this device will be ambient parameters for the model under examination. Such a device placed before the intake pipe can be a supercharger.

In the other two connected systems, there are phenomena relating to the process of dynamic changes in charge parameters.

The four systems connected in the series in question cooperate with each other in reciprocal charge exchange.

When making a number of assumptions, general equations (1, 2, and 3) of mass and momentum and angular momentum conservation laws as well as that of energy conservation law assume the following form:

$$\begin{aligned} \frac{\partial u}{\partial t} &= -\frac{1}{\rho}\frac{\partial \rho}{\partial x} - u\frac{\partial u}{\partial x} - u \cdot k_t, \\ \frac{\partial p}{\partial t} &= u^2 \cdot k_t \cdot \rho(\chi - 1) - u\frac{\partial p}{\partial x} - \chi \cdot p\frac{\partial u}{\partial x}. \end{aligned} \tag{4}$$

The most important numerical question which occurs during the solving of a model is to reduce partial differential equations to ordinary differential ones. The equations reduced to this form can be solved with the finite difference method using computer calculations. When solving the numerical question of fluid flow, a method of characteristics or a method of lines is being used. Some researchers attempted to use the finite difference method to solve questions relating to fluid and gas flow but the same authors found that these examinations were time-consuming and had no significant advantages in relation to the method of characteristics or lines.

The numerical solution of this question consists in the partitioning of the whole length of intake pipe into $n$ sections with the same length $k$. The examined cross-section $x_i$ belongs thus to section $x_i \in \langle 0, L_d \rangle$.

When denoting:

$$\begin{aligned} x_i &= i \cdot k \\ \rho_i(t) &= \rho(x_i, t) \\ u_i(t) &= u(x_i, t) \qquad for \quad i = 0,1,2,\ldots,n, \\ p_i(t) &= p(x_i, t) \\ T_i(t) &= T(x_i, t), \end{aligned} \tag{5}$$

and assuming that:

equation; $\frac{\partial\rho}{\partial t}\frac{\partial\rho}{\partial t}+u\frac{\partial\rho}{\partial x}+\rho\frac{\partial u}{\partial x}=0$ is fulfilled when it is fulfilled for $x_0$, $x_1$, $x_2$, ..., $x_{n-1}$, $x_n$,

equation $\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+\frac{1}{\rho}\frac{\partial p}{\partial x}-k_r\cdot u=0$ is fulfilled when it is fulfilled $x_0$, $x_1$, $x_2$, ..., $x_{n-1}$,

equation $\frac{\partial p}{\partial t}=u^2\cdot k_r\cdot\rho(\chi-1)-u\frac{\partial p}{\partial x}-\chi\cdot p\frac{\partial u}{\partial x}$ is fulfilled when it is fulfilled for $x_0$, $x_1$, $x_2$, ..., $x_{n-1}$, $x_n$.

The vector notation of equations is as follows [10]:

$$\begin{aligned} \frac{d\rho}{dt}&=F(\rho,p,u,t) & \rho&=[\rho_1,\ldots,\rho_n]^T \\ \frac{dp}{dt}&=F(\rho,p,u,t)\ \text{where:} & p&=[p_1,\ldots,p_n]^T \\ \frac{du}{dt}&=F(\rho,p,u,t) & u&=[u_1,\ldots u_n]^T\ . \end{aligned} \tag{6}$$

In order to solve the system of equations (4), the MacCormack's explicit method was used [4][11]. This method is particularly useful in solving inviscid gas flows. An earlier attempt to solve the numerical question by means of the method of lines did not rener the desired results due to high instability.

In order to make simulation calculations, computing software Matlab v. R2009a was used.

The developed programme enabled the determination of, as follows:

- $m_c$ [kg] – mass of air remaining in the engine cylinder after completion of the filling process,
- $p_d$ [Pa] – pressure in any point of the intake pipe,
- $T_c$ [K] – temperature in the engine cylinder.

The above parameters have been presented in the form of variation diagrams for a given parameter in the function of crank angle. Additionally, it is possible to visualise the course of pressure changes along the intake pipe with the inlet valve open and closure stage plotted.

As constant data for simulation, engine-related data as well as ambient parameters and basic thermodynamic data were adopted (Table 1).

Table 1 Engine data and ambient parameters adopted for simulation

| Parameter | Value | Parameter | Value |
|---|---|---|---|
| Cylinder diameter D | 0.127 [m] | Combustion chamber surface ratio | $\mu_c$ = 0.7 |
| Crank radius $R_k$ | 0.073 [m] | Ambient pressure $p_o$ | 100000 [Pa] |
| Piston stroke S | 0.146 [m] | Ambient temperature $T_o$ | 293 [K] |
| Connecting rod length $l_k$ | 0.265 [m] | Adiabatic exponential χ | 1.4 |
| Connecting rod length ratio $\lambda_k$ | 0.27 | Universal gas constant MR | 8.3147 [J/mol·K] |
| Cylinder wall temperature $T_w$ | 370 [K] | Air molar mass MM | 28.96 [g/mol] |
| Combustion chamber volume in piston $V_k$ | 10-4 [m3] | Specific heat at constant volume $c_v$ | 20.8 [J/mol·K] |
| Flow ration on inlet valve $\mu_i$ | 0.8 | Specific heat at constant pressure $c_p$ | 29.11 [J/mol·K] |
| Mean piston speed $c_m$ | [m/s] | | |

## RESULTS OF SIMULATION ANALYSIS

When evaluating the effect of intake pipe length on the mass of air in engine cylinder, a number of simulation was performed for different pipe lengths in the function of engine crankshaft rotational speed. Simulations were made for the following lengths: 0.743 m, 0.843 m, 1.143 m, and 1.293 m.

Exemplary diagrams of the variation of air mass in the engine cylinder for an intake pipe length of 0.843 m and rotational speed 1000 1/min are presented in Fig. 1. The intake pipe diameter was determined in this evaluation to 0.07 m.

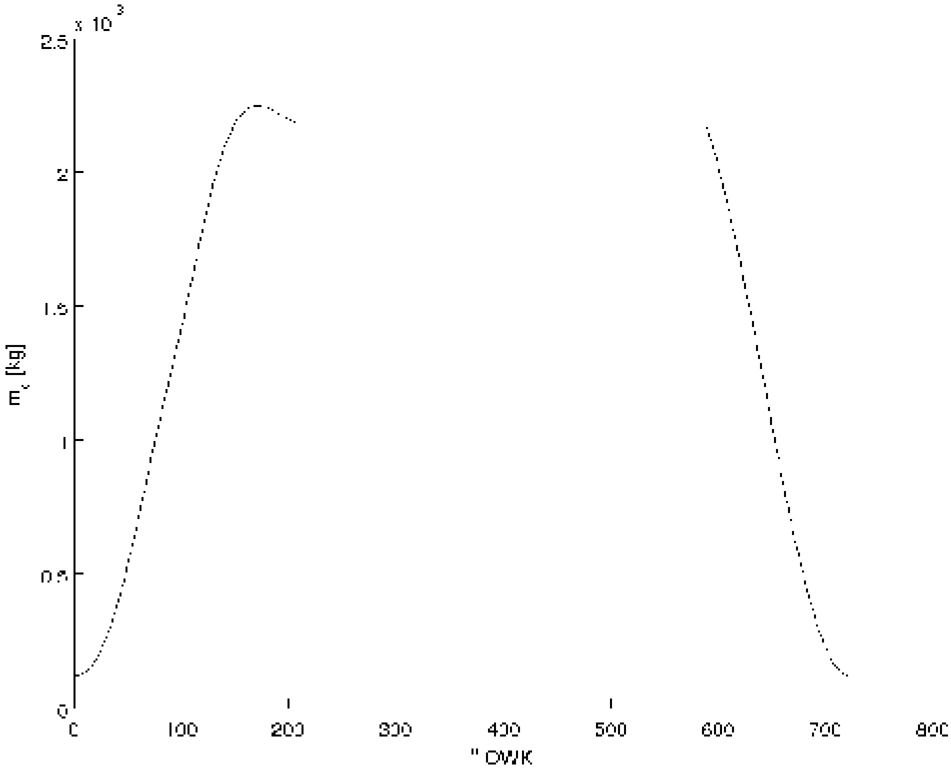


Fig. 1. Air mass in the engine cylinder for the intake pipe length of 0.843 m and rotational speed 1000 1/min

In order to better show the effect of intake pipe length on the air mass in the engine cylinder, mass values for respective pipe lengths at the speeds within the range of useful rotational speeds for the SW-680 engine are presented in Table 2 and Fig. 2.

Table 2. Mass values for respective pipe lengths at the speeds within the range of useful rotational speeds for the SW-680 engine

| | $L_d$ = 0,743 m | $L_d$ = 0,843 m | $L_d$ = 1,143 m | $L_d$ = 1,293 m |
|---|---|---|---|---|
| n | $m_t$ | $m_t$ | $m_t$ | $m_t$ |
| [1/min] | [g] | [g] | [g] | [g] |
| 1000 | 2,213 | 2,185 | 2,236 | 2,225 |
| 1200 | 2,233 | 2,257 | 2,261 | 2,174 |
| 1400 | 2,255 | 2,285 | 2,206 | 2,216 |
| 1600 | 2,257 | 2,211 | 2,226 | 2,601 |
| 1800 | 2,222 | 2,225 | 2,623 | 2,548 |
| 2000 | 2,267 | 2,435 | 2,409 | 2,61 |
| 2200 | 2,433 | 2,272 | 2,671 | 2,265 |

It can be seen in Fig. 2 that within the range of low and medium rotational speeds (from 1000 to 1600 1/min) the largest air mass in the engine cylinder, and at the same the largest value of the filling coefficient, occurred for the intake pipe with the length of 0.843 m at the rotational speed of about 1400 1/min. A slightly smaller air mass, by about 1%, occurred for the intake pipe with the length of 1.143 m at the rotational speed of about 1200 1/min. Undoubtedly, this is an advantage of this pipe length because the expected torque will significantly increase at this speed. The intake pipe with the length of 0.743 m was characterised by air mass comparable with that obtained for the intake pipe with the length of 1.143 m. However, this length is slightly worse from the point of view of earlier arrangements due to the fact that the air mass for this length was obtained at a larger rotational speed, about 1500 1/min.

When examining the variation of air mass in the engine cylinder for the intake pipe with the length of 1.293 m, it can be seen that there is no maximum within the considered range of rotational speeds. It can be concluded from the course of lines that the maximum may be obtained at the rotational speed lower than 1000 1/min but this interval is not used in practice in traction applications.

Within the range of higher rotational speeds, high air mass values can be observed for 1.143, 1.293 and 0.843 m. Within this range of rotational speeds, larger air mass in engine cylinder will affect the value of maximum powers.

Based on the course of lines for the pipe length of 0.743 m, it can be concluded that the largest air mass in the engine cylinder has not reached the maximum yet but this is of no practical importance because the examined engine can not reach rotational speeds higher than 2200 1/min.

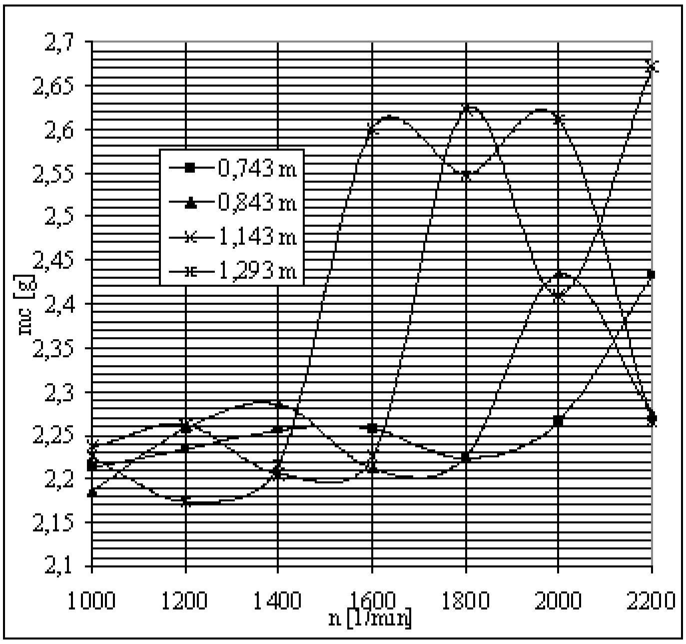


Fig. 2. Cumulative characteristics of air mass in the engine cylinder for the intake pipe lengths under examination

## CONCLUSIONS

A basic advantage of simulation programme is involved in the determination of the most favourable geometric intake system parameters for a newly designed or modernised engine, or in developing the construction of the existing engine intake system with the purpose of improving its operating parameters, or in adaptation of the operating characteristic of engine to its operation conditions.

Making a decision based on the analyses of the results of simulation examinations reduces the costs and time necessary for the determination of the most favourable – under given conditions – parameters of the intake system. When using the numerical methods for the solution of the questions connected with combustion engine cylinder filling, experimental examinations come down to confirmation of the accuracy of the taken decisions.

Construction parameters of the intake system can be determined based on experimental examinations but a great number of parameters will require multiple tests. Considering the fact that the probability of achieving satisfactory results as early as in the first series of tests is practically equal to zero, execution of the whole number of test for a number of variable parameters will be extremely time-consuming and expensive. Therefore, it seems purposeful to estimate provisionally the required parameters and then perform tests confirming the selection accuracy.

The programme presented in this paper enables an evaluation of the effect of basic parameters of the intake system on the engine filling process. Based on the carried out simulation and experimental examinations, the following conclusions can be drawn:

1. A basic parameter of the intake system most affecting the cylinder filling is intake pipe length.
2. The magnitude of cylinder filling is also affected by timing gear system parameters, in particular the time of inlet valve closure. This can be seen in Fig. 1 where the largest air mass occurs before inlet valve closure, after which it decreases rather quickly at the time of its closing. According to the author, it is possible to reduce the outflow of air mass from engine cylinder to the intake system by accelerating slightly the time of valve closure.
3. Numerical simulations allow shortening of the time and reducing the cost of implementation of construction modifications in the inlet system but can not entirely replace the experimental examinations.

During the performed examinations and analyses, the author stated the necessity of carrying out examinations in future that will determine:

- effect of valve timing phase on cylinder filling; and
- effect of ambient conditions (temperature and pressure), which will allow for an application of a numerical programme for the evaluation of the filling process of an engine with a combined supercharging system.

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# NUMERYCZNA SYMULACJA WPŁYWU DŁUGOŚCI PRZEWODU DOLOTOWEGO NA MASĘ POWIETRZA POZOSTAJĄCĄ W CYLINDRZE SILNIKA O ZS PO ZAMKNIĘCIU ZAWORU DOLOTOWEGO

**Streszczenie.** W artykule przedstawiono ogólny opis modelu numerycznego układu dolotowego silnika spalinowego. Opracowano model jednowymiarowy uwzględniający zasadę zachowania energii. Przedstawiono możliwości obliczeniowe modelu i analizę wyników symulacji wpływu długości przewodu dolotowego na masę powietrza w pozostającą w cylindrze silnika po zamknięciu zaworu dolotowego.

**Słowa kluczowe:** układ dolotowy, model numeryczny, masa ładunku w cylindrze, symulacja