МАШИНОСТРОИТЕЛЬНЫЕ КОМПОНЕНТЫ

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ANALYTICAL AND EXPERIMENTAL MODEL OF THE DIESEL INJECTOR

An analytical-experimental model has been proposed to determine sac volume pressure changes and fuel doses injected into the diesel cylinders. The model takes into account friction forces and the mechanical properties have been identified. Engine performance results have been analysed statistically. The results of the Lilliefors, Pearson, Shapiro-Wilk and Jarque-Bera tests have provided evidence to reject, under certain engine operating conditions, the hypothesis about the compatibility of the measurement data with the normal distribution. Measurement uncertainty for the injector needle lift has been estimated. The difference between the measured and predicted fuel dose values have amounted to approximately 3 %.

Keywords: *injector, modelling, fuel expense, uncertainty of measurement*

Introduction. Conventional internal combustion engines fitted with mechanical injection pumps are stilled used in off-road vehicles. Their robustness and relative insensitivity to fuel properties and quality are attractive features. Elements of those conventional engines, after some re-engineering, are integrated in the common rail injection systems in which parameters are adjusted electronically [3]. Analysis of fuel pressures in cross-sections perpendicular to the injector axis [8] reported lateral motion of the needle in addition to its axial motion. Due to non-symmetric fuel pressure distribution on both sides of the clearance, the needle is pressed against the body orifice or even rotated about its axis. Reference [7] discussed fuel pressure distribution in the entire injection system focusing on the effect of fuel compressibility on the injected fuel dose. The authors proposed a model of the injector needle movement, which took account of viscous friction. A two-mass vibro-impact model was proposed in [5]. The authors of [4] presented a one-degree-of-freedom viscous elastic model and analysed the effect of the injector design on the needle motion. They observed variation in the flow coefficient even in the case of the same injector, this being dependent on the operating conditions and the fuel physical parameters. The influence of the injection pressure on the fuel spray parameters, and the coefficient of variation for the fuel dose were determined in [1]. In this study, the model of the injector needle motion proposed in [2] was modified to allow viscous friction and other resistive forces (denoted as friction *T*) to be taken into account. The identification procedure for the mechanical properties of the model parameters was changed and the simulations of the needle lift, pressure in the injection sac and the fuel dose injected were conducted. Results of the experiments for the diesel- or bio-fuelled engine were analysed statistically, which allowed the experimental data distributions to be compared with the normal distribution. Then the selected parameters of the model were identified and the measurement uncertainty for the injector needle lift maximum was determined. The model was validated through the comparison of the calculated and the measured values of the needle lift.

Experimental facilities. The test engine used for these measurements was a 3-cylinder compression-ignition Perkins AD3.152 UR. Selected parameters of its performance were tested at the Laboratory of Internal Combustion Engines, Kielce University of Technology [9] on the test stand capable of measuring: pressure in the combustion chamber, pressure in the injection pipe, injector needle lift, crank angle.

The experimental studies [9] conducted using an engine dynamometer included measurements of the pressure in the injection pipe, in-cylinder pressure and the needle lift. Values from 50 full engine working cycles were recorded as a function of the crank angle with an increment of 1.4° for each parameter, which gave 512 measurement points within one engine working cycle. This study used the results obtained for the engine working at full load condition at the variable rotational speeds from 1000 to 2000 rpm. The engine ran on diesel or biofuel (FAME — methyl esters of fatty acids).

Analysis of experimental results. The first step in the development of an analytical-experimental model of an injector was to carry out a statistical analysis of the tests results for checking stationarity and compatibility of the variables distribution with normally distributed data. Once the signals are found to be stationary and their distributions are compatible with normal distribution, measurement uncertainties can be determined [1], and their synchronically averaged values can be used in further analysis. If the signals are found to be non-stationary and their distribution is not normal, a different procedure should be chosen [10].

The analyses [1] rendered the signals stationary. Autocorrelation function was analysed in this study to confirm the stationarity of the signals. The Lilliefors and the Pearson tests were carried out to determine unequivocally whether the measurement dataset distribution could be said to be compatible with the normal distribution. The analysis of the compatibility of the maximum pressure in the combustion chamber distribution [1] showed that only when n = 2000 rpm, the Lilliefors tests provided evidence to reject the null hypothesis (H0) about the compatibility of the measurement dataset distribution with normally distributed data at the significance level α = 0,05. However, the results of the additional Shapiro-Wilk test provided no reason to reject this hypothesis. Analysis of the results using the Jarque-Bera test confirmed also the earlier conclusions – there was no reason to reject the null hypothesis H0.

Evaluation of the injection pressure stationarity and conformity of this measurand distribution to the normal distribution employed the same procedures as those used for pressure in the combustion chamber [1]. The results showed stationarity of the signal. The Pearson and the Lilliefors tests indicated that only for speed n=2000 rpm there was a reason to reject the null hypothesis H0. This result was confirmed in the Shapiro-Wilk test, where the p-value = 0,01 (the Jarque-Bera test did not provide any evidence to reject the hypothesis H0).

Analysis of the maximum values of the needle lift recorded for the consecutive engine working cycles indicates that the signal is stationary [1]. The statistical measures relating to the shape of the distribution, together with the results from the Lilliefors and the Pearson tests which help evaluate the goodness of fit (for all the speeds investigated) show that for the speeds 1200, 1400, 1600 and 1800 rpm, at least one of the tests provides evidence to reject the hypothesis H0.

The following methodology was used for estimating the uncertainty of measurand maximum values: first-identification of uncertainty sources, next-evaluation of type A standard uncertainty, after this-evaluation of type B standard uncertainty and finally-determination of expanded uncertainties at the 95 % interval of confidence.

Standard uncertainty determined in type A evaluation is the estimate of the standard deviation of the mean expressed with the dependence [6]:

$$u_A = \sqrt{\frac{\sum_{i=1}^{n}\left(x_i - \bar{x}\right)^2}{n(n-1)}} \tag{1}$$

where as a measured quantity estimate x_i is an arithmetic mean from n observations. To estimate the boundaries of the confidence interval, which contains the unknown real value of the quantity measured with the adopted confidence level $\propto$ an extended uncertainty u_{CA} has to be determined:

$$u_{CA} = k_A(\propto)\cdot u_A, \tag{2}$$

standard uncertainty corresponding to the value of $k_A(\alpha) = 1$ is determined for the confidence level $\propto = 0{,}6827$.

Type B evaluation of standard uncertainty can be defined as [6]:

$$u_B = \frac{\Delta g}{\sqrt{3}}, \tag{3}$$

where Δg is a measurement error resulting from the class of the measurement equipment used. In the dependence above, a uniform distribution of the true value probability in interval $\pm\Delta g$ is assumed. The extended value of type B uncertainty can be calculated from following equation:

$$u_{BC} = k_B(\propto)\cdot u_B, \tag{4}$$

where $k_B(\propto)$ denotes the standardised variable of certain probability distribution (for instance, for uniform distribution $k_B(\propto) = \propto\sqrt{3}$). The measurement uncertainty of the injector needle lift is affected by: a displacement sensor error (δ_c = 1 %), a charge amplifier error (δ_w = 0,1 %) and a A/C transducer error (δ_{ac} = = 0,024 %). The total value of the relative error relating to the type B uncertainty [6] of the injector needle lift was approximately δ = 1 %, whereas the absolute error referred to the maximum measurement range (2 mm) was $\Delta h_i = \Delta h_i$ = 0,02 mm. The type B uncertainty was measured with a prior assumption that the distribution of probability of occurrence of the true value in the range Δh_i was uniform, which gave the same values of type B standard uncertainty for all rotational speeds:

$$u_B = \frac{\Delta h_i}{\sqrt{3}} = 0{,}012 \text{ mm}.$$

Dependence (1) was used (50 maximum values h_i recorded in consecutive working cycles were used as input quantities) to evaluate the type A uncertainty of the maximum injector needle lift. An example of type A standard uncertainty for n = 2000 rpm was u_A = 0,005 mm.

Test results for compatibility of the distribution of maximum values of the needle lift with the normal distribution were taken into account while measuring expanded uncertainty at the confidence level of 95 %.

In some cases, these results provide enough evidence to reject the hypothesis H0. Therefore it is necessary to find the values of expansion factor $k_B(\infty)$ which should take standardised values for the probability distribution type defined by the probability distribution of the variables. In this paper, value $k_B(\infty) = 2$ was adopted for the interval of confidence of 95 %. Uncertainty of measuring the mean needle lift reaches considerably higher values when it is evaluated using the type B method. Type B standard uncertainty for the mean maximum needle lift was about 0,01 mm for all the rotational speeds.

The fuel dose used during one working cycle can be calculated from:

$$V_{wtr} = m\int_{t_{pw}}^{t_{kw}} U_p \pi d_r^2 dt, \tag{5}$$

where initial velocity U_p of the fuel injection is defined by:

$$U_p = \mu_r \sqrt{\frac{2\Delta P(OWK)}{\rho_l}}, \tag{6}$$

other denotations include: t_{kw} — injection end time, t_{pw} — injection start time, d_r — diameter of holes through which the fuel is injected, μ_r — flow coefficient, ΔP (OWK) — pressure difference between the injector and the cylinder, ρ_l — fuel density, m — number of holes through which the fuel is injected.

In calculations of the fuel dose injected using dependence (5), parameter ΔP is used relating to the pressure difference between, most often, the injector pipe and the cylinder. This approach leads to an overestimation of the results as in reality, the dose is dependent on the difference between pressure in the sac and in the chamber. It is thus warranted to attempt building a model that will help track pressure changes in the sac volume and the dose of the fuel injected. For that purpose, analytical dependencies were formulated and experimental data were used [9]. Then selected parameters of the injector model were identified and computer simulations of the injector needle lift and the fuel dose injected were made.

Analytical-experimental model of injector. The mathematical model of the injector needle motion can be expressed in the form of equations (7) and (8). The flow continuity equation for the sac volume can be written as [2]:

$$\frac{V_s}{E_s}\frac{dp_s}{dt} = \operatorname{sgn}(p_w - p_s)\mu_g A_g \sqrt{\frac{2}{\rho_l}|p_w - p_s|} - \\ -\varepsilon_s \mu_r A_r m \sqrt{\frac{2}{\rho_l}|p_s - p_c|} - \frac{dV_s}{dt}. \tag{7}$$

Dynamic equation of the injector needle motion will have the form:

$$m_w \frac{d^2 h_i}{dt^2} = -\beta_w \frac{dh_i}{dt} - k_{sw}(h_i + h_{w_0}) + \\ + p_w(A_{ip} - A_i) + p_s A_i - T, \tag{8}$$

where $T = f(p_s)$ — other resistive forces affecting the needle motion (friction forces T). It was assumed that the sum of these forces is proportional to the pressure in the sac volume $T = f(p_s) = -sgn\left(\frac{dh_i}{dh}\right)\cdot \mu p \cdot p_s$. Other denotations: V_s — the volume of the nozzle cavity; E_s — elasticity modulus, p_w, p_s, p_c — pressure in the intake tube, sac volume and engine cylinder; μ_g — flow factor within the injector socket; A_g — cross-sectional flow area of the seat; $\varepsilon_s = 1$ for $p_s \geq p_c$ and $\varepsilon_s = 0$ for $p_s < p_c$; A_r — flow area of hole through which the fuel is injected; m_w — mass of injector needle; h_i — injector needle lift; β_w — viscous friction coefficient; k_{sw} — spring constant; A_i — cross-sectional area of the injector needle; A_{ip} — cross-sectional area of the injector needle gib part. Equations (7) and (8) are non-linear due to the relationships that occur among the following parameters: $V_s = f(h_i)$, $A_g = f(h_i)$, $E_s = f(p_s, T)$ and $\rho_l = f(p_s, T)$. Because E_s and ρ_l are unknown, it was assumed that the temperature, elasticity modulus E_s and density ρ_l of the fuel in the sac are constant during the injection and are independent of the pressure changes. Also, it was assumed that the values: μ_g — within the injector socket and μ_r — through the injector orifices are constant and independent of the injector needle lift. These assumptions simplify the physical pattern of the processes [4]. It has to be noted, however, that the experiments were carried out after the engine operation stabilised.

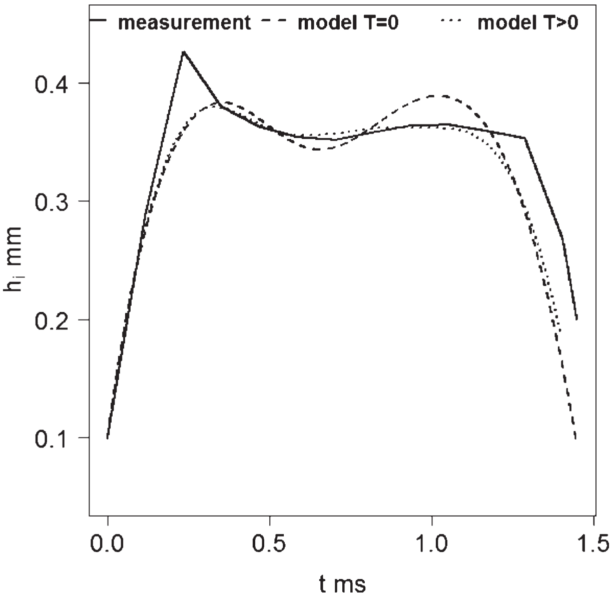
The variables used in calculations are the measured values of pressures in the intake tube p_w, in the engine cylinder p_c, and the injector needle lift h_i. All these parameters, although connected with the cyclic process, take different values in consecutive engine working cycles [1]. The model proposed here helps determine the values of pressure in the sac volume p_s, the fuel dose injected and the needle lift h_i. The last of these parameters is used to validate the model. The unknown variables to be identified are the viscous friction coefficient β_w spring constant ksw and coefficient μp. Identification and determination of the changes p_s and the fuel dose injected should be performed within the crank angle range selected for analysis. The range for which injection occurs is determined based on the measured lift of the injector needle. It was assumed that the injection occurs in the range of crank angles for which the value of the injector needle lift satisfies the condition $h_i \geq 0{,}04$ mm.

System of equations (7) and (8) was solved numerically with use of MATLAB/Simulink package. The model was validated by comparing the values of the injector needle lift calculated theoretically with the lift value recorded during the measurements. The qualitative agreement was satisfactory. The quantitative agreement was found unsatisfactory when friction forces $T = 0$. The parameters of the model β_w, k_{sw} and μp were determined based on the authors' own injector model built in Simulink with the use of *fminsearch* function from MATLAB/Optimization toolbox. This procedure allows the minimum of a multivariate function to be found by means of the simplex algorithm of the Nelder-Mead method. Lack of constraints as to the values of the parameters investigated is the disadvantage of the method; its advantage lies in the ease of implementation. The application of *fminsearch* function involves building a criterion dependent on identified parameters and then determining such values of these parameters for which this criterion achieves the minimum value. The following objective function was used as a criterion for optimum selection of β_w, k_{sw} and μp values:

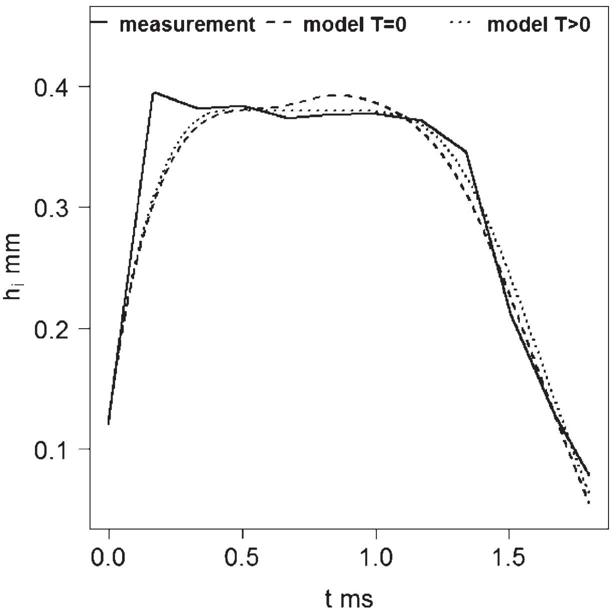
$$K_r = \int_{t_{pw}}^{t_{kw}} \left| h_i\left(t, \beta_w, k_{sw}, \mu p\right) - h_{pom}(t) \right| dt, \tag{9}$$

where $h_{pom}(t)$ — the measured value of the injector needle lift. The criterion adopted in this way allows finding the values of β_w, k_{sw} and μp for which the model correctly describes the injector operation within the entire fuel injection range. In the process of identification the crank angle range changes in each working cycle and so does the simulation time for which the identification is performed. The simulation time is selected individually for each cycle so that condition $h_i \geq 0{,}04$ mm is fulfilled, which corresponds to the injection range. Initial conditions of displacement and the needle speed are also defined individually for each cycle on the basis of the actual change history $h_i(t)$. The values of the other parameters in equation (8) are the same for all other working cycles being identified. Identification and validation of the model were performed based on the synchronically averaged changes in pressures p_w and p_c and the needle lift h_i. The identification results of the spring coefficient of elasticity values indicate that they change only slightly for a given velocity of the crank angle and do not exceed 5 %. It has to be noted that the constant of the spring of the injector being investigated, approximately $3{,}2 \cdot 10^5$ N/m, was also determined in the static compression test carried out using a universal testing machine MTS at the Fracture Mechanics Laboratory, Kielce University of Technology. Figure 1 shows a needle lift curve determined experimentally and analytically according to equations (7 and 8) following the identification for the engine running at 2000 rpm on diesel and at 1400 rpm on FAME. The comparison of the results indicates that the model correctly determines the injector needle displacement. Taking into account the remaining needle motion resistive forces improved the accuracy of calculations. Attention needs to be paid to a considerable decrease in parameter K_r value, which is a measure of discrepancy between the model and the experimental data after identification and with friction forces $T > 0$ taken into account. The accuracy of viscous friction coefficient identification is unsatisfactory. This is a result of too small a number of measurement points (making up the investigated plot) when the needle moves upwards — as a result, the value of the start speed of the needle is determined inaccurately. Examples of simulation results for the needle lift are shown in figure.

The maximum difference between the pressure values in the intake tube determined experimentally and the value of pressure in the sac determined analytically for an engine running at a speed of $n = 1400$ rpm is approximately 14 %.



a



б

Figure — Model and experimental curves expressing the dependence of the injector needle lift on time for: *a* — an engine supplied with diesel; rotational velocity of the crankshaft n = 2000 rpm; *b* — an engine supplied with biofuel FAME; rotational velocity of the crankshaft *n* = 1400 rpm

Determining pressure changes in the sac volume allows the modification of the algorithm for determining a fuel dose. The fuel dose has to be computed using dependencies (5) and (6). The pressure difference ΔP can be calculated based on the measurement data as $\Delta P = p_w - p_c$. It is worth taking into consideration that fuel doses can be determined using unaveraged data for particular engine working cycles. The mean values of the injected fuel doses ranged from $5 \cdot 10^{-8}$ to $6 \cdot 10^{-8}$ m^3

depending on the crank angle velocity, and their standard deviation varied from $4 \cdot 10^{-10}$ to $6 \cdot 10^{-10}$ m^3. But, knowing the computed pressure changes in the sac volume, it is better to determine ΔP as $\Delta P = p_s - p_c$. The fuel dose results derived from the pressure differences between measured p_w or computed p_s (for averaged variations) were compared with those determined experimentally [9]. For example, for an engine running at n = 1400 rpm, fuel consumption reported in the literature is 5,67 kg/h, whereas the same parameter calculated from model was 5,53 kg/h. The discrepancy amounts to approximately 3 %. The fuel dose determined based on the injection pressure measured in the intake tube may be overestimated by about 12 %.

Simulation of the injector operation indicated that the friction force T in equation (8) changes nonlinearly as a function of the injector needle path. For the diesel-fuelled engine and working at a speed of n = 1400 rpm, its maximum value is approximately 20 N. The value of work performed by friction forces T was $66 \cdot 10^{-4}$ J for n = 1400 rpm and $74 \cdot 10^{-4}$ J for n = 2000 rpm.

Examples of simulation results in the form of the model and experimental curves illustrating the dependence of the injector needle lift on time for the FAME-fuelled engine are shown in figure *b*.

Final conclusions. The analytical-experimental injector model proposed correctly determines the sac volume pressure changes and the needle displacements. The differences between the values obtained for the intake tube and the sac volume amount to approximately 14 %. This causes an overestimation of the calculated fuel dose volume of about 12 %. The calculated dose of the injected fuel, after identification of the proposed injector model parameters, varies from the theoretical dose by approximately 3 %. The mean value of the viscous friction coefficient changes irregularly with the increase in the crankshaft rotational speed. The mean value of the elasticity coefficient determined during the identification process increases with the increase in the crankshaft rotational speed. Viscous friction forces and other resistive forces incorporated in the model allowed for a better fit of the experimental curves to the model-predicted curves. Standard uncertainty of measurement of the injector needle maximum lift mean computed using the type B evaluation is considerably higher for all the rotational speeds than that determined using the statistical method.

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Расчетно-экспериментальное моделирование процесса впрыска в дизельном двигателе

Предложена аналитическо-экспериментальная модель впрыска в цилиндр дизельного двигателя, в целях определения изменения давления в камере впрыска и расхода впрыскиваемого топлива. В модели учтены силы трения и проведена идентификация механических свойств. Выполнена статистическая обработка результатов экспериментальных исследований двигателя. Результаты тестирования по методам Lillieforsa, Pearosona, Shapiro-Wilka, а также Jarque-Bera позволили опровергнуть общепринятую гипотезу о достоверности измерений при нормальном распределении. Определена недостаточность измерений сечения впрыска. Разница между измеренными и расчетными значениями расхода топлива составила 3 %.

Ключевые слова: *впрыск, моделирование, расход топлива, погрешность измерения*

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