INFLUENCE OF GEOMETRIC AND HYDRO-DYNAMIC PARAMETERS OF INJECTOR ON CALCULATION OF SPRAY CHARACTERISTICS OF DIESEL ENGINES

by

Ivan M. FILIPOVIĆ [*]***, Boran D. PIKULA, and Dževad Š. BIBIĆ***

Department for IC Engines and Vehicles, Faculty of Mechanical Engineering,
University of Sarajevo, Sarajevo, Bosnia and Herzegovina

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The main role in air/fuel mixture formation at the IC Diesel engines has the energy introduced by fuel into the IC engine that is the characteristics of spraying fuel into the combustion chamber. The characteristic can be defined by the spray length, the spray cone angle, and the physical and chemical structure of fuel spray by different sections. Having in mind very complex experimental set-ups for researching in this field, the mentioned characteristics are mostly analyzed by calculations. There are two methods in the literature. The first based on use of the semi-empirical expressions (correlations) and the second, the calculations of spray characteristics by use of very complex mathematical methods. The second method is dominant in the modern literature.

The main disadvantage of the calculation methods is a correct definition of real state at the end of the nozzle orifice (real boundary conditions). The majority of the researchers in this field use most frequently the coefficient of total losses inside the injector. This coefficient depends on injector design, as well as depends on the level of fuel energy and fuel energy transformation along the injector. Having in mind the importance of the real boundary conditions, the complex methods for calculation of the fuel spray characteristics should have the calculation of fuel flows inside the injector and the calculation of spray characteristics together. This approach is a very complex numerical problem and there are no existing computer programs with satisfactory calculation results.

Analysis of spray characteristics by use of the semi-empirical expressions (correlations) is presented in this paper. The special attention is dedicated to the analysis of the constant in the semi-empirical expressions and influence parameters on this constant. Also, the method for definition of realistic boundary condition at the end of the nozzle orifice is presented in the paper. By use of this method completely avoid a use of the coefficient of total losses inside the injector. At the same time, semi-empirical expressions have the universal constant that does not depend on the injector design.

Keywords: *injector, spray, diesel fuel, loss coefficient, modeling*

Introduction

The combustion process in the direct fuel injected Diesel engines (DI) mostly depends from quality of the air/fuel mixture formation. The air/fuel mixture process can be accepted and analyzed by total energy of the air and the fuel introduced into the DI. At DI,

* Corresponding author; e-mail: filipovic@mef.unsa.ba

especially at DI with greater displacement, the most important role is played by the fuel energy. In all realistic conditions, the fuel energy can be manifested through spray characteristics. The spray characteristics, in the ambient where the spray develops, can be expressed in the following form:

– the spray dimensions like: the spray length, the spray cone angle, parameters after spray impact with wall, *etc*. as a function of the time, and
– the physical and chemical spray structure, where the physical spray structure is determinate mostly by mean Sauter diameter of droplets in different spray cross-sections. On the other hand, a chemical structure in different cross-sections of the spray and as a function of the time is interesting in case of multi components fuels, like diesel fuel. The basic principle in case of these fuels is that the heavy component of C_XH_Y are concentrated in the middle of the spray and have a better penetration, while lighter C_XH_Y are located on the periphery of the spray and have a better vaporization.

In the air/fuel mixture formation analysis, the spray shape characteristics (spray dimensions) are mentioned mostly. Based on these characteristics, one can make conclusions regarding the air/fuel mixture homogeneity in the whole ambient of the combustion chamber, the cross overlying next sprays, the influence of the combustion chamber walls, *etc*. In this context, the majority of the researchers analyzes diesel fuel sprays and recommends influence parameters for spray dimensions and shapes. Generally, the different spray shapes are mostly presented in the fig. 1, where the free shape fuel spray, injected into free ambient without walls, is presented in fig. 1(a).

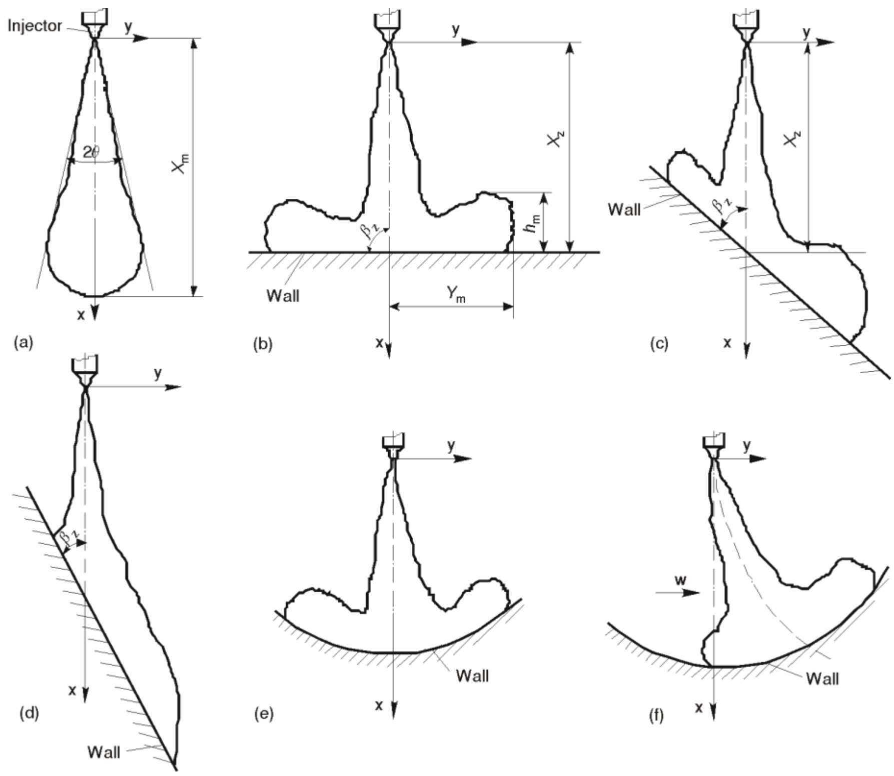

Figure 1. The characteristic spray shapes in different ambient

Furthermore, the spray shapes injected into free ambient impact to different wall shapes and are presented in figs. 1(b) to 1(e). The shapes of sprays in impact with flat wall in free ambient on distance x_z and under angle β_z are presented in figs. 1(b) to 1(d). In the same time, shape of the spray in impact with curved wall in free ambient is presented in fig. 1(e). The most realistic case to real working conditions in the Diesel engine is the shape of the spray impact with curved wall, fig. 1(f), where air speed w exists. The mentioned cases represent the most common cases used in analysis of the spray characteristics. It is very important to notice, the non-stationary values of ambient pressures and temperatures, the different types of ambient air flows (swirl, tumble, *etc*.) and variable space of the ambient where the fuel is injected should be taken into consideration. For mentioned conditions, the spray analyses become more complex and there are not present in the literature.

Influence of geometric and hydro-dynamic parameters of the injector will be presented in the example of spray length as function of time $X_m = f(t)$, as well as spray dimensions after impact on wal, (Y_m, $h_m = f(t)$.

Analysis of fuel spray characteristics

Simple model of fuel spray

The fuel spray characteristics are mostly defined by:

– the numerical solution of the continuity, the motion and the energy equations improved with additional models for definition of some phenomena like: break-up zone, turbulence effect, vaporization, droplets impact, *etc*. and
– the different semi-empirical expressions (correlations).

The first method of calculation is more applied in the literature [1] and [2]. In order to make realistic definition of boundary conditions, the calculation of fuel flows inside the injector together with the calculation of air/fuel mixture formation should be necessary. In this case, boundary conditions are: fuel state in high pressure pipe before the injector (entrance) and state of the ambient where the fuel is injected (an exit on the nozzle orifice). Having in mind the complexity of spray characteristics calculation, this method is not used in practice. In case of the calculations of spray characteristics based on the numerical solutions, researchers accept boundary conditions at the nozzle orifice as a known value. However, the researchers do not discuss are realistic these values (pressure and velocity) and how they have been determined. Based on these facts one can conclude that the greatest mistakes for further calculations were done just on the start of the calculations.

It was the reason to make analysis of influence of the injector design and influence of hydro-dynamic fuel characteristics through injector on spray length by use of the second method for defining fuel spray characteristics. According to the literature, there are a great number of different semi-empirical expressions to calculate the fuel spray characteristics. Some characteristics, like X_m, Y_m, and h_m, and firstly the spray length X_m out from the break-up zone, have almost the same form of the semi-empirical expressions with different constants considered by the majority of researchers.

All researchers in this field take a known physical model presented in fig. 2(a) as the basis for its research and use the mathematical models for stationary sprays. The basic equation used for their calculation of spray length is the momentum equation. The spray in the free ambient (there is no air motion) – air with known conditions (p_c, T_c) will be used for the analysis. In the same time, there is no vaporization in the free ambient while air pressure is equal to the atmosphere pressure. Drag force due to motion of fuel spray in the free ambient is taken in to consideration, as well. Based on the knowledge of the fluid mechanics and based on momentum of fluid flows in the characteristic cross-sections over the reasonable length, the simple mathematical model for calculation of the spray length can be developed.

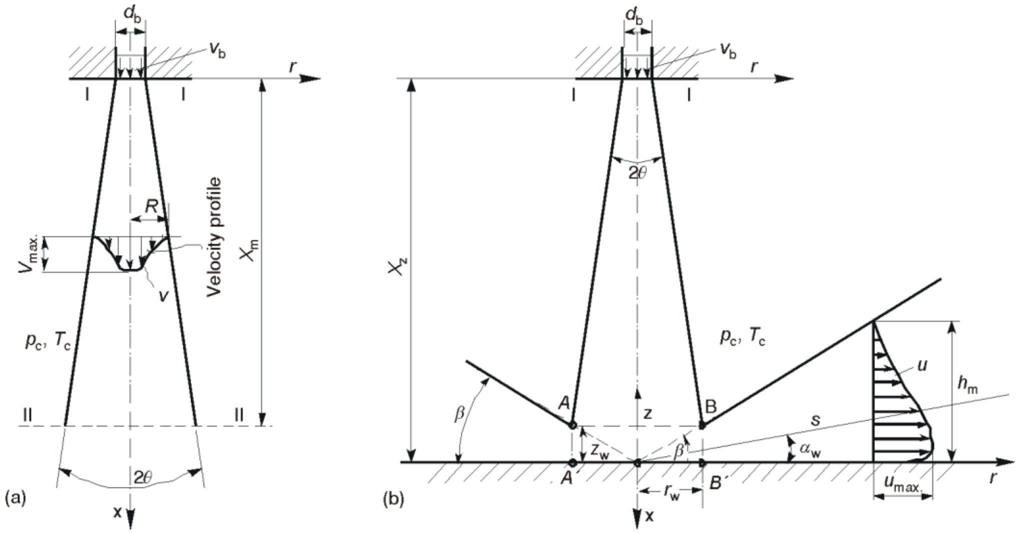

Figure 2. Physical models of free spray (a) and spray with wall impact (b) in free ambient

Based on the physical model presented in fig. 2(a), the mathematical model between cross-section I-I and II-II can be expressed by use of momentum of fluid flows as:

$$K_{\text{I-I}} = K_{\text{II-II}} \tag{1}$$

Expression for the momentum of fluid flow in cross-section I-I can be written as:

$$K_{\text{I-I}} = \rho_\text{L} \frac{d_\text{b}^2 \pi}{4} v_\text{b}^2 = \frac{\pi}{2} c_\text{b}^2 d_\text{b}^2 \Delta p \tag{2}$$

where the velocity at the exit from the injector orifice is calculated by Bernoulli equations for referent locations: before the injector, cross-section A-A, fig. 3(a) and the ambient conditions (p_c, T_c), like:

$$v_\text{b} = c_\text{b} \sqrt{\frac{2\Delta p}{\rho_L}} \tag{3}$$

Pressure before the injector is calculated as the average value of the pressure at the end of the high pressure pipe during fuel injection process. Some authors calculate the average value of this pressure for the first 0.5 ms of injection duration [3].

Expression for momentum of the fluid flow, without drag of spray motion, in the cross-section II-II can be written as:

$$K_{\text{II-II}} = \int_0^R \rho_\text{m} v^2 2\pi r \,\text{d}r = 2\pi \rho_\text{m} v_{\text{max.}}^2 R^2 \varphi_1 \tag{4}$$

where $\varphi_1 = \int_0^1 f^2(\eta)\eta \,\text{d}\eta$.

For this analysis one assumption is adopted. The velocity profile (v) in the spray, out from potential core of the spray, has the following function $v = v_{\mathrm{max.}} f(\eta)$, where $\eta = r/R$. The dimensionless function of the air/fuel mixture velocity profile $f(\eta)$ in different cross-sections mostly response to the Gauss error function. Other functions like $f(\eta) = (1 - \eta^{3/2})^2$ presented in [4] and [5], or $f(\eta) = (1 - \eta^2)$ presented in [6] are used also for analysis. However, taking the drag of spray motion into consideration, which is expressed by the coefficient c_{m}, the final expression for the momentum in the cross-section II-II has the form:

$$K_{\text{II-II}} = 2\pi\varphi_1\rho_{\mathrm{m}}v_{\mathrm{max.}}^2 R^2 + \frac{1}{2}c_{\mathrm{m}}\,2\pi\varphi_1\rho_{\mathrm{m}}v_{\mathrm{max.}}^2 R^2 = 2\pi\varphi_1\left(1+\frac{c_m}{2}\right)\rho_{\mathrm{m}}v_{\mathrm{max.}}^2 R^2 \tag{5}$$

Assuming that spray width is proportional to spray length [7], in arbitrary section $R = c_1X_{\mathrm{m}}$, as well as the local fuel density is calculated as:

$$\rho_{\mathrm{m}} = \frac{\mathring{m}_{\mathrm{L}} + \mathring{m}_{\mathrm{z}}}{\mathring{V}_{\mathrm{m}}} = \frac{\mathring{m}_{\mathrm{L}} + \rho_{\mathrm{c}}\left(\int_0^1 2\pi R^2 v_{\mathrm{max.}}\, f(\eta)\mathrm{d}\eta - \dfrac{\mathring{m}_L}{\rho_{\mathrm{L}}}\right)}{\int_0^1 2\pi R^2 v_{\mathrm{max.}}\, f(\eta)\mathrm{d}\eta} \tag{6}$$

It can be assumed that, out of the spray potential core, $\rho_{\mathrm{m}} \approx \rho_{\mathrm{c}}$ and the eqs (1), (2), and (5) offer the following relation:

$$\frac{\pi}{2}c_{\mathrm{b}}^2 d_{\mathrm{b}}^2\Delta p = 2\pi\varphi_1\left(1+\frac{c_{\mathrm{m}}}{2}\right)\rho_{\mathrm{c}}v_{\mathrm{max.}}^2 c_1^2 X_{\mathrm{m}}^2 \tag{7}$$

i. e.

$$v_{\mathrm{max.}} = \frac{1}{2}\frac{c_{\mathrm{b}}c_1^{-1}\varphi_1^{-0.5}}{\left(1+\dfrac{c_{\mathrm{m}}}{2}\right)^{0.5}} d_{\mathrm{b}}\left(\frac{\Delta p}{\rho_{\mathrm{c}}}\right)^{0.5}\frac{1}{X_{\mathrm{m}}} \tag{8}$$

Based on the fact that the maximum velocity ($\mathbf{v}_{\mathrm{max.}}$) can be expressed as $v_{\mathrm{max.}} = \mathrm{d}X_{\mathrm{m}}/\mathrm{d}t$, the spray length can be calculated from eq. (8) as:

$$X_{\mathrm{m}} = \left[\frac{c_{\mathrm{b}}c_1^{-1}\varphi_1^{-0.5}}{\left(1+\dfrac{c_{\mathrm{m}}}{2}\right)^{0.5}}\right]^{0.5} d_{\mathrm{b}}^{0.5}\left(\frac{\Delta p}{\rho_{\mathrm{c}}}\right)^{0.25} t^{0.5} = C_1 d_{\mathrm{b}}^{0.5}\left(\frac{\Delta p}{\rho_{\mathrm{c}}}\right)^{0.25} t^{0.5} \tag{9}$$

where C_1 is the constant defined as:

$$C_1 = \frac{c_{\mathrm{b}}^{0.5}c_1^{-0.5}\varphi_1^{-0.25}}{\left(1+\dfrac{c_{\mathrm{m}}}{2}\right)^{0.25}} \tag{10}$$

By using the physical model of spray impact in to the wall, fig. 2(b), with assuming ome simplifies like in the previous model, the characteristic spray parameters after impact into wall (Y_{m}, h_{m}) can be defined. For a simply analysis, the equation of continuity for control volume bordered by surfaces *AB* and *A'B'*, and envelope A-A' – B-B' was used. The equation has the form:

$$\int_0^{r_w} 2\pi\rho_c v_{\max.} f\left(\frac{y}{r_w}\right) y\mathrm{d}y = \int_0^{z_w} 2\pi\rho_c u_{\max} f_1\left(\frac{z}{z_w}\right) r_w \mathrm{d}z \tag{11}$$

where the velocity profile along the wall is assumed as $u = u_{\max.}f_1(z/z_w)$ and f_1 is the mathematical function that represent the shape of realistic velocity profile in this part of the spray approximation. Assuming that the nozzle orifice diameter, marked in fig. 2(b), is significantly smaller regarding other geometrical dimensions, the following simple trigonometry relations can be written:

$$\left.\begin{aligned} &z_w = \frac{x_z \mathrm{tg}\theta\, \mathrm{tg}\beta}{1+\mathrm{tg}\theta \mathrm{tg}\beta}; \quad r_w = \frac{x_z \mathrm{tg}\theta}{1+\mathrm{tg}\theta \mathrm{tg}\beta} \\ &r = s\cos\alpha; \quad z = r\,\mathrm{tg}\alpha; \quad h = r\,\mathrm{tg}\beta \end{aligned}\right\} \tag{12}$$

By use of eqs. (11) and (12) the velocity profile of spray (u), near the wall, can be defined, as well as the location of the co-ordinate s and the spray dimensions $Y_m = r$ and $h_m = h$ as function of time. After detail analysis of the spray dimensions Y_m and h_m one can make conclusion that the influenced parameters for these characteristics are the constant c_b, the spray drag coefficient c_m, the functions of velocity profiles $f(r/R)$ and $f_1(z/h)$, the spray angle 2θ and the angle β, the pressure difference Δp, the nozzle orifice diameter d_b, and the time $(t - t_z)$, where the t_z is the impact time. The intensity is different to the intensity expressed in the eq. (9). Similar models for spray characteristics can be developed in case of air speed presence or in case of the spray with variable ambient conditions (p_c, T_c) with fuel vaporization.

In further analysis, the attention will be dedicated to the spray characteristics during injection into the free ambient and partly to fuel spray characteristics after impact into the wall.

Fuel spray characteristics defined by correlation expressions

Spray length during injection into the free ambient without air speed is calculated by use of semi-empirical (correlation) expressions based on expression (9). There are several correlation expressions in the literature where the different values have only the constant C_1. Values for the constant C_1 can be: $C_1 = 3.01$ as presented in [8], $C_1 = 3.8$ as presented in [6] and finally $C_1 = 3.9$ as presented in [6].

The spray characteristics after impact on the wall can be defined the similar way. According to the literature, the following expressions can be found:

$$\left.\begin{aligned} Y_m &= C_2 \Delta p^{a_1} \rho_c^{a_2} (t - t_z)^{a_3} \\ h_m &= C_3 \Delta p^{a_4} \rho_c^{a_5} (t - t_z)^{a_6} \end{aligned}\right\} \tag{13}$$

The following parameters exist in aforementioned expressions (9) and (13): the fuel pressure difference (Δp), the density of air (ρ_c), nozzle orifice diameter (d_b), and constants C_1, C_2, and C_3. These constants contain the constant c_b which represents the injector loss coefficient. Analyzing the expression (9) for the spray length during injection into the free ambient and the constant C_1, one can easily see a significant dissipation of this constant in range, according to the literature, from 2.95 to 3.9. By using expression (10) and making an analysis of every single value like c_b, c_m, c_1, and φ_1, it is obvious that constants c_m, c_1, and φ_1 have almost the same values. In the same time, the constant c_b, at the same fuel pressure before the injector (p_A) and the same nozzle orifice diameter (d_b), can have different values depending on injector design. This leads to the conclusion that the main goal for different values of the constant C_1 is the injector design, what is the topic for further analysis.

Results of research for known injector

Model for calculation of hydro-dynamics characteristics

The previous analysis leads to the conclusion that fuel velocity on the nozzle orifice exits (v_b), expressed by eq. (3), and it is the most important parameter for definition of initial and boundary conditions for spray development. Having in mind the fact that two parameters were used to define value of fuel velocities (v_b) as:

– the fuel pressure difference (Δp), defined by the fuel pressure before the injector (p_A) on the location the easiest for measuring and by the pressure of the ambient (p_c), and
– the coefficient (c_b) that takes into account all losses inside the injector (friction, minor loss, fuel contraction),

most researchers based their researching on the measurement of the fuel pressure before the injector (p_A), making average value (p_Asr) and for known ambient pressure (p_c), calculate the fuel pressure difference $\Delta p = p_\mathrm{Asr} - p_\mathrm{c}$. In the same time, the literature was used for adopting the injector loss coefficient (c_b). Taking into the consideration the different injector designs (some examples are shown in fig. 3), as well as the different fuel pressures before the injector, *i. e.* fuel velocities inside the injector the coefficient c_b can obviously have significantly different values. These different values mostly influence the constant C_1, as well as the constants C_2 and C_3 in many semi-empirical expressions. Also, it is very important to notice fuel properties, especially viscosity that can influence on the coefficient c_b. This fact is especially important in case of use of different fuels, like biodiesel, blends, *etc*. Further analysis will be directed to total losses in the injector, from cross-section A-A to the cross--section I-I, fig. 3(a), *i. e.* the coefficient of total losses through injector (c_b).

By using the eq. (3) and with known fuel pressure in the cross-sections A-A and I-I (fig. 3), current value of the loss coefficient c_b^* and the mean value of the loss coefficient can be written in the form:

$$c_\mathrm{b} = \sqrt{\frac{p_\mathrm{Isr} - p_\mathrm{c}}{p_\mathrm{Asr} - p_\mathrm{c}}}; \quad c_\mathrm{b}^* = \sqrt{\frac{p_\mathrm{I} - p_\mathrm{c}}{p_\mathrm{A} - p_\mathrm{c}}} \tag{14}$$

adopting the fact that ambient pressure p_c = const. and ignoring the losses structure through injector. The pressure p_I cannot be measured directly and it's measuring, indirectly from injection rate measuring, is very complex. For this reason, the calculation of hydro-dynamics characteristics inside the injector is used and definition of the injector loss coefficient is based on this calculation. The structure of losses the best can be seen in form of Bernoulli equation between cross-section A-A and I-I, fig. 3(a):

$$p_\mathrm{A} + \frac{\rho_\mathrm{L} v_\mathrm{A}^2}{2} = p_\mathrm{I} + \frac{\rho_\mathrm{L} v_\mathrm{b}^2}{2} + \sum_j \xi_j \frac{\rho_\mathrm{L} v_j^2}{2} + \sum_i \lambda_i \frac{\rho_\mathrm{L} v_i^2}{2} \tag{15}$$

where all minor losses, all friction losses of the hydro-dynamics characteristics, at first velocities v_i and v_j, should be known in different cross-sections.

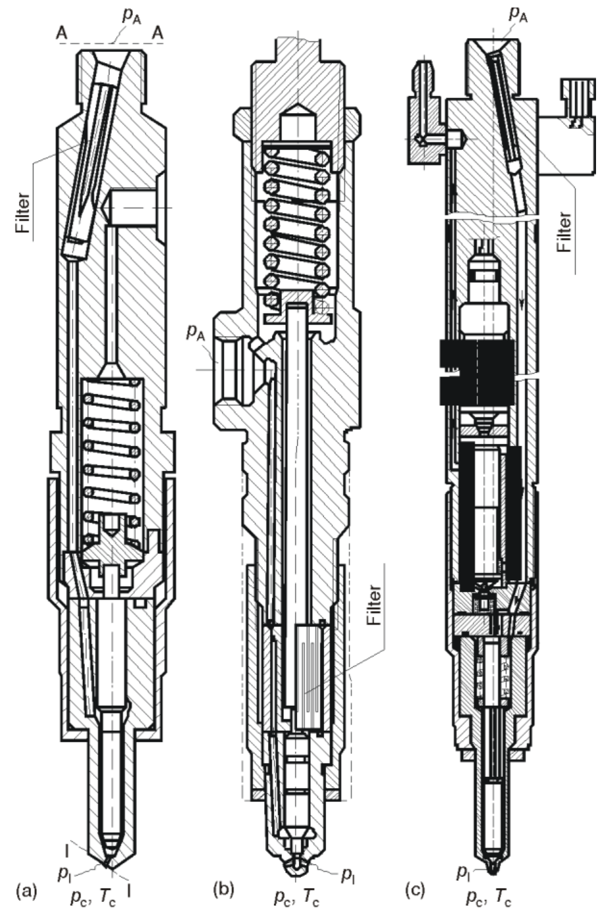

Figure 3. Different injectors design
(a),(b) – conventional injectors with hydraulic control of needle lift, (c) – injector with piezoelectric actuator with electronic control of needle lift

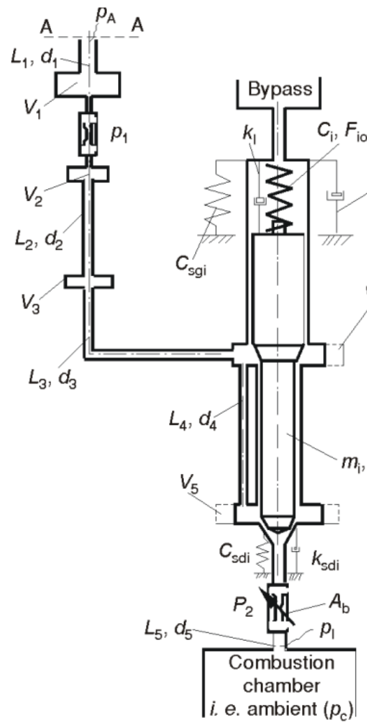

Figure 4. Physical model of the injector presented in fig. 3(a)

Solving the hydro-dynamics characteristics of fuel inside the injector, according to the state-of-the-art modeling in this field, should be combination of dimensionless and 1-D mathematical models. For this analysis, one conventional injector, shown in fig. 3(a) is taken as an example. The injector is the Bosch manufactured injector which consists from an injector holder of KDAL 80S20/129 type and an injector body of DLL 25S834 type, with only one nozzle orifice (diameter d_b = 0.68 mm and length l_b = 2 mm). More information regarding the injector can be found in [9]. The physical model of used injector, convenient for dimensionless and 1-D model, consisting of pipes, spaces with constant and changeable volumes, thin wall orifices with constant and variable flow cross-sections (P), combined with the dynamic model of needle movement is presented in fig. 4.

A detail review of characteristics of injector physical model is presented in [9].

The basic equations for fuel flows calculations are:

(a) for the pipes with dimensions length L_i and diameter d_i

– continuity equation

$$\frac{\partial p}{\partial t}+\rho a^2\frac{\partial v}{\partial x}=0 \tag{16}$$

– momentum equation

$$\frac{\partial v}{\partial t}+\frac{1}{\rho}\frac{\partial p}{\partial x}+\frac{\lambda_i v|v|}{2d}=0 \tag{17}$$

(b) for volumes (V_j), a continuity equation is used in the form:

$$\frac{V_j}{E}\frac{\mathrm{d}p}{\mathrm{d}t}=\sum_{m=1}^{m_k}Q_m \tag{18}$$

The solutions of eqs. (16) and (17) for 1-D isentropic fluid flows in pipes can be found by use of method of characteristics. At the same time, it has been necessary to solve continuity eq. (18) for volumes by setting numerical method convenient for coupling with method of characteristics. The principles for solving the mentioned equations in case of different combinations of pipes and volumes, by use of the Newton-Raphson numerical method, as well as the thin wall orifices are presented in [9] and [10]. The coefficients of friction losses in eq. (17) and minor losses, expressed by Q_m in eq. (18) are taken from the [11] and [12]. The equation of the needle dynamic motion is expressed in the form:

$$m_\mathrm{i}\frac{\mathrm{d}^2h_\mathrm{i}}{\mathrm{d}t^2}+k_\mathrm{i}\frac{\mathrm{d}h_i}{\mathrm{d}t}+c_\mathrm{i}h_\mathrm{i}=\sum F \tag{19}$$

For this purpose eq. (19) is assumed as a linear non-homogenous differential equation of second order. Equation (19) has transformed into two differential equations of first order which solution have been found by the common fourth-order Runge-Kutta numerical method.

In brief presented physical and mathematical models with appropriate numerical solutions have been verified by the experimental results. The experiments are original and done in Laboratory for IC Engines at Mechanical Engineering Faculty, University of Maribor, Slovenia. The piezoelectric sensor of KISTLER manufacturer of 6227 type, with measurement range of 0-2000 bar in wide temperature range –50 °C up to 200 °C and sensitivity 2.5 pC/bar has been used for measurement of fuel pressure at the end of the high pressure pipe p_A immediately before the injector. The needle lift has been measured by inductive sensor AP 5,5/1,0, while the pressure at the nozzle orifice exit p_I has been calculated from the pressure obtained by piezoelectric sensor of KISTLER manufacturer, of 7031 type, used for definition of the injection rate (law).

The examples of calculated and measured values of the fuel pressures and the needle lift, for adopted injector at the full load and speed regime of the high pressure pump speed at 1100 min.$^{-1}$, are shown in fig. 5.

The calculation results of the characteristic parameters show there are no cavitation in the fuel injection system that is confirmed by the experiments. Also, the cavitation phenomena did not exist in injector nozzle that is analyzed in computational fluid dynamics software by AVL Fire and shown in [9]. A similar conclusion can be obtained by the analysis of the results presented in [13].

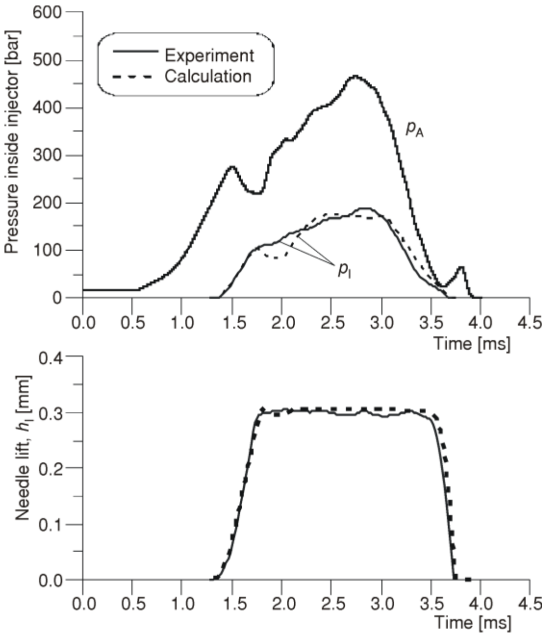

Figure 5. Diagram of fuel pressures (p_A and p_I) and needle lift (h_i) in function of time

Having in mind results presented in fig. 5, one can conclude an excellent matching between experimental and calculation results. In this way, the model for calculation of the hydro-dynamics characteristics of fuel flows inside the injector is verified. Similar matching of experimental and calculation results were achieved on other speed regimes of the high pressure pump. By having the presented and verified models, it is possible to make detail analysis of total losses, as well as for every single loss inside the injector, that depends from injector designs and fuel parameters, cross-section A-A, fig. 3(a).

Analysis of the loss coefficient c_b

Based on the great number of complex experimental measurements of pressure difference $p_{Asr} - p_c$ and $p_{Isr} - p_c$ and with the measurements of the injection rate $\dot{q}_c$, the loss coefficient c_b can be defined by use of the eq. (14). However, according to the literature the loss coefficient c_b can have the following values: $c_b = 0.6\text{-}0.8$ as shown in [14], $c_b = 0.7\text{-}0.75$ as shown in [4], $c_b = 0.65$ as shown in [6], *etc*. These values leads to conclusion that values of the loss coefficient c_b can be found in a wide range. An explanation for the wide range can be found in different injector designs, hydro-dynamics flows, quality of metal processing, *etc*. In the same time, this is the answer for significant influence on the constant C_1 (eq. 10), as well as on the constants C_2 and C_3 in semi-empirical expressions (9) and (13). In order to better understand influence of the loss coefficient c_b and its variations with different injector parameters, in the rest of this paper and based on the experimental results and on the model presented in the section *Model for calculation of hydro-dynamics characteristics*, on the basis of expression (14) the analysis of influence parameters has been done. The analysis of influence of the intake fuel pressures on the loss coefficient c_b, in case on the mentioned injector (DLL 25S834) in known fuel injection system, is done and the results are shown in fig. 6. Mean values of fuel pressure before injector (p_{Asr}) were in a range from 175 to 342 bar, while the values of loss coefficient c_b were in a range from 0.523 to 0.593.

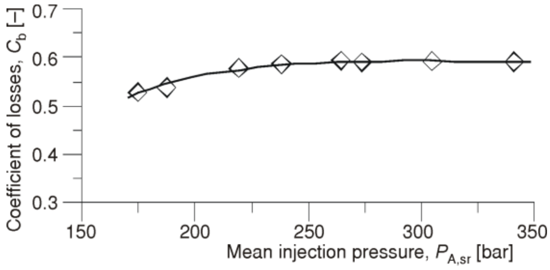

Figure 6. Trend of the loss coefficient as function of mean fuel pressure $p_{A,sr}$ during injection

Having in mind the range of values of the loss coefficient one can conclude that dissipation of the values is up to 13%. The lower values of the loss coefficient, *i. e.* the greater losses, have been arhived in case of the lower mean fuel pressure (p_{Asr}) before the injector. Although lower values of mean fuel pressures before the injector mean lower fuel velocities, the main reason for this trend is not full needle lift. The presented results are related to the average values of loss coefficient c_b during injection. It is very interesting to make analysis of the current value of loss coefficient c_b^*, during injection, as well. The example of curves that show the trend of current values of the loss coefficient for two different engine speeds (n = 700 min.$^{-1}$ and n = 1100 min.$^{-1}$) are presented in the fig. 7. These variations of current values of the loss coefficient are more expressed with different design solutions on the injectors.

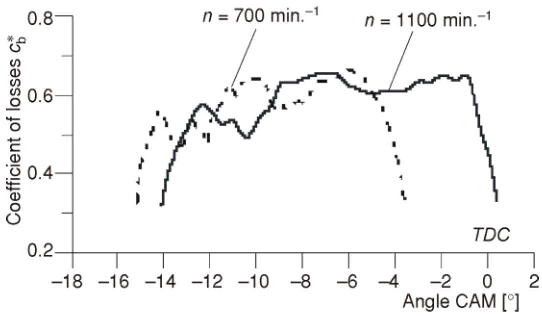

Figure 7. Trend of the current value of the loss coefficient c_b^* during injection

Besides the general trends of the loss coefficient, the short analysis of influence of different cross-sections inside the injector has been done. This analysis takes into the consideration the locations of:
– the filter inside the injector, and
– the needle seat.

The original design of the filter flow area in the injector is 2.58 mm^2. During exploitation, with presence of dirt, changing a filter temperature and flow contractions, the flow area can be changeable. Having in mind these situations, the analysis has been done in case of different flow areas in the filter, from 0.785 mm^2 to 3.1 mm^2. Results with values of the loss coefficient c_b in case of the max of speed regime (n = 1100 min.$^{-1}$) are presented in fig. 8. The arrow in fig. 8 represents the original flow area of the filter inside the injector.

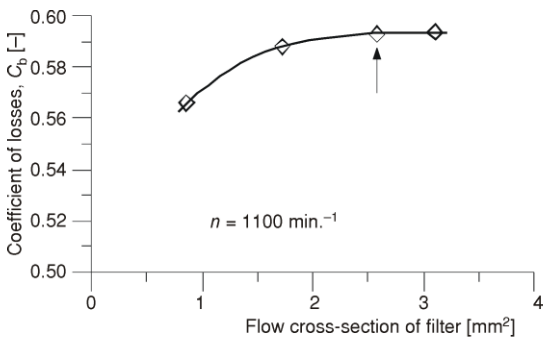

Figure 8. Trend of the loss coefficient c_b as function of the filter flow area

Since the values of the loss coefficient c_b are in the range from 0.561 to 0.595 one makes conclusion that injector filter design can have a significant influence on the loss coefficient, almost up to the 6% on one speed regime. In the same time, it is very easy to see that filter flow area has the optimal value (the minimum losses) because further increasing of the filter flow area keeps constant value of the loss coefficient.

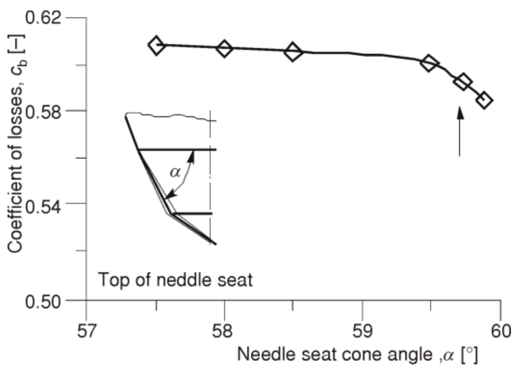

Figure 9. Influence of needle cone angle at needle seat (α) on the loss coefficient c_b

The second parameter which has an influence on the loss coefficient is the location of the needle seat. In order to make an analysis, the different needle cone angles α have been used, as it shown in fig. 9. The characteristic results (values are in range from 0.585 to 0.608) are presented also in fig. 9, where the arrow represents the original needle cone angle. It is interesting to notice that increase of needle cone angle $\alpha > 59.3°$ will reduce the loss coefficient. On the other side, decreasing of needle cone angle does not have any influence on the loss coefficient. The mentioned decreasing of needle cone angle can produce additional wearing of needle seat. Since the needle cone angle is important for flow losses from one side, as well as for wearing of the needle seat from the other side, it is very hard to find the optimal solution and always presents a compromise. The original needle cone angle has value $\alpha = 59.72°$.

All presented analyses show the loss coefficient c_b as a parameter that cannot be adopted from the literature and use in the semi-empirical expressions like (9). The main goal is contained in facts that injector designs, hydro-dynamics flow conditions inside the injector, physical properties of fuel and finally dynamics characteristics of the needle have a significant influence on the loss coefficient. Having in mind all facts, as the most acceptable procedure is the following one. Based on the measured fuel pressure before the injector (p_A), the fuel pressure at the nozzle orifice exit (p_I) is calculated by use of the model presented in the section *Model for calculation of hydro-dynamics characteristics*. In this case, the loss coefficient is c_b = 1. The examples of the calculated pressure at the exit from nozzle orifice (p_I), based on the measured fuel pressure (p_A), for different high pump speeds n = 1100, 900, 700, and 500 min.$^{-1}$, are shown in fig. 10.

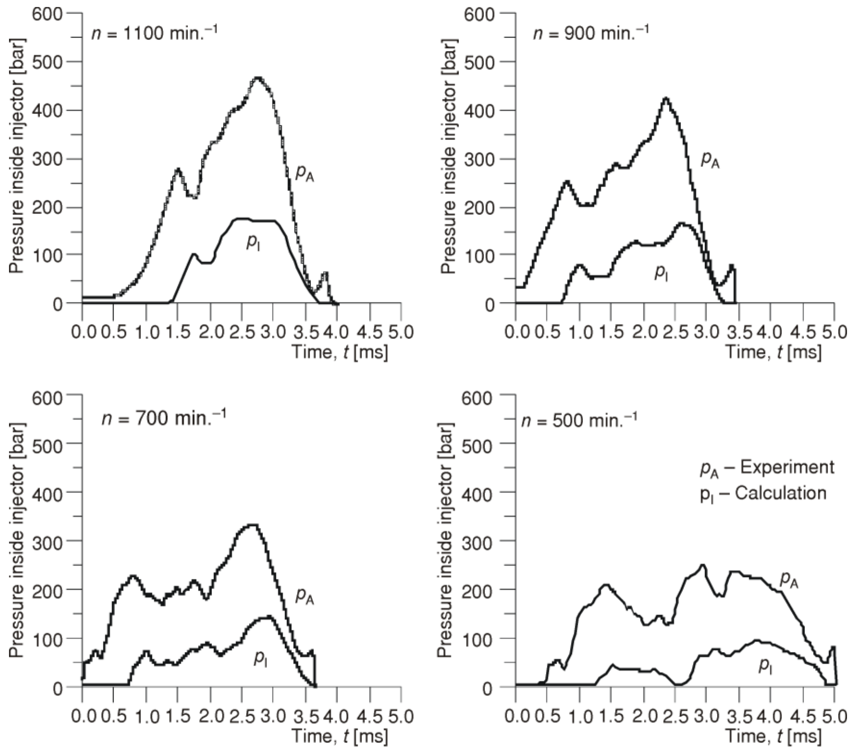

Figure 10. Measured fuel pressure (p_A) and calculated fuel pressure (p_I) for different engine speeds of high pressure pump

All research and analyses have been done on the full load regime of the fuel injection system, *i. e*. on the same geometric amount of fuel moved by the piston of the high pressure pump. Also, all values of the loss coefficient of the injector (c_b) are related to the mean loss coefficient during one injection process, except values for c_b^* that is related to the current value of loss coefficient presented in fig. 7.

By using this method for calculation of the fuel pressure at the nozzle orifice exit (p_I), eqs. (9) and (10) transform to the following form:

$$X_m = C_1^* d_b^{0.5} \left(\frac{\Delta p^*}{\rho_c} \right)^{0.25} t^{0.5} \tag{20}$$

$$C_1^* = \frac{c_1^{-0.5} \varphi_1^{-0.25}}{\left(1 + \frac{c_m}{2} \right)^{0.25}} \tag{21}$$

where C_1^* is the real constant and have the same value for known ambient conditions. In the same time, pressure difference Δp^* is calculated on the following way $\Delta p^* = p_{Isr} - p_c$. Using this methodology, all influence parameters from the injector and hydro-dynamics flow conditions inside the injector are eliminated, which is shown in eq. (20).

Conclusions

Based on the literature review of semi-empirical expressions for calculation of the spray length the significant dissipation of the results is ensured up to 30%. By detail analysis of the physical processes in spray development and methods for definition of semi-empirical expressions one can conclude that the greatest influence on final form of semi-empirical expressions has the injector loss coefficient. The coefficient depends on the following parameters:
- the speed and load regimes of the diesel fuel injection system,
- the injector geometry and design,
- the hydro-dynamics parameters of fuel inside the injector, and
- the physical fuel properties, especially viscosity.

On this way it is shown that the injector loss coefficient is not single-valued defined, even for the same injector and full load regime of the diesel fuel injection system. For these reasons, avoiding of use of semi-empirical expressions for calculation of spray length is recommended. The greatest influence has the injector loss coefficient and the following procedure is suggested.
- Application of simple models which combine dimensionless and 1-D models for fuel flows inside the injector. Using the models will make possible a correct definition of pressure and velocity values in every single location inside the injector.
- Using calculated value of pressure or velocity at the injector orifice exit (the loss coefficient is equal 1) as the boundary condition for definition of spray characteristics.

The use of this procedure avoids the influence of the injector loss coefficient for definition spray length using semi-empirical expressions. Question of pressure difference between the pressure at the injector orifice exit and ambient where the spray is formatted is not analyzed. Based on the average values of fuel pressure during one injection process the principle of definition of pressure difference is retained.

Nomenclature

A_b – geometrical flow area located below injector needle cone, [m^2]
a – speed of sound through fuel, [ms^{-1}]
$a_1, a_2, \ldots, a_6$ – constants, [–]
C_{sdi}, C_{sgi} – stiffness of upper and lower needle seats, [Nm^{-1}]
C_1, C_2, C_3 – constants, [–]
C_1^* – constant, [–]
c_b – coefficient of total losses in the injector, [–]
c_i – needle spring stiffness, [Nm^{-1}]
c_m – coefficient of movement resistance, [–]
c_1 – constant of the proportionality, [–]
d_b – diameter of the nozzle orifice, [m]
d_i – pipe diameter, [m]
F – force, [N]
F_{io} – force of the injector opening, [N]
h_i – injector needle lift, [m]
h_m – spray high on wall, [m]
K_{I-I}, K_{II-II} – momentum in characteristic cross-sections, [kgm^{-3}]
k_i – damping coefficient of needle movement through fuel, [Nsm^{-1}]
k_{sdi}, k_{sgi} – damping coefficient (upper and lower) of needle movement on needle seats, [Nsm^{-1}]
L_i – pipe length, [m]
l_b – nozzle orifice length, [m]
m_i – mass of the needle with moving parts, [kg]
$\mathring{m}_L$ – fuel flow on cross section of the spray, [kgs^{-1}]
$\mathring{m}_Z$ – air flow on cross section of the spray, [kgs^{-1}]
n – high pressure pump speed, [$min.^{-1}$]
p_A – instantaneous value of fuel pressure before the injector during injection process, [bar]
p_{Asr} – mean value of fuel pressure before the injector during injection process, [bar]
p_c – pressure of injection ambient, [bar]
p_I – instantaneous value of fuel pressure at the end of nozzle orifice, [bar]
p_{Isr} – mean value of fuel pressure on the nozzle orifice exit, [bar]
Δp – pressure difference from the injector fuel entrance to the end of the nozzle orifice, [bar]
P_1, P_2 – blends
Q_m – volumetric fuel flows, [m^3s^{-1}]
R – spray radius in the spray cross-section, [m]
r – co-ordinate, [m]
s – co-ordinate of spray movement after wall impact
t – time, [s]
T_c – temperature of the ambient where fuel is injected, [K]
u – spray velocity near the wall, [ms^{-1}]
u_{max} – max velocity in spray cross-section near the wall, [ms^{-1}]
$V_{max.}$ – maximum of the spray velocity, [ms^{-1}]
$V_1, V_2, \ldots, V_j$ – volumes, [m^3]
$\mathring{V}_m$ – volumetric fuel flow of air-fuel mixture – in spray, [ms^{-1}]
v – velocity in spray cross-section, [ms^{-1}]
v_A – fuel velocity before the injector, [ms^{-1}]
v_b – fuel velocity at the end of the needle orifice, [ms^{-1}]
v_i, v_j – fuel velocity on location *i*, *i. e.* location *j*, inside the injector, [ms^{-1}]
X_m – spray length, [m]
X_z – distance between wall and the injector, [m]
x – co-ordinate, [m]
Y_m – spray length near the wall, [m]

Greek symbol

α – needle seat cone angle, [°]
α_w – angle of the co-ordinate s, [°]
β – spray angle on the wall, [°]
λ_i – hydraulic friction coefficient, [–]
ρ_c – ambient density, [kgm^{-3}]
ρ_L – fuel density, [kgm^{-3}]
ρ_m – air/fuel mixture density, [kgm^{-3}]
θ – half value of spray angle, [°]
ξ_i – coefficient of the minor loses, [–]

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