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ONE DIMENSIONAL MODEL FOR DEFINITION OF FUEL ENERGY TRANSFORMATION INSIDE INJECTOR OF DIESEL ENGINE

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ABSTRACT

Investigation of fuel flows inside the injector is the main topic from early beginnings of development of diesel fuel injection systems. Due to complex inside design of injector, a numerous different experimental and numerical methods have been developed and used in past. In case of 1D numerical modeling, a special attention was dedicated to high pressure pipe, while an injector is most often modeled as a blend with known flow coefficient thanks to the experiment. However, development of 3D numerical method based on use of CFD increased knowledge about fuel flows, but it is generally limited to the final segments of injector, i.e. needle seat, sac and nozzle. This paper presents a universal one dimensional mathematical model for solving fuel flow across the whole inside the injector. One dimensional mathematical model presented in this paper enables solving fuel flows through inside the injector based on simple elements such as pipes, blends and pipe-volume junctions. On that way, fuel flows, as well as fuel energy transformation, have been fully defined in all points of a very complex inner side of the injector and shows characteristic places inside the injector where special care has to be paid during design of new geometric shapes. Considering that all existing research in the spray characteristics were based on a known pressure in front of the injector, very good initial conditions for calculating the characteristics of dispersed fuel can be obtained with this model, which up until now was not the case.

Keywords: diesel engine**,** injector, fuel characteristics, mathematical modelling, fuel energy

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INTRODUCTION

First attempts to define the energy potential of fuel inside the injector are linked to a research from the early 1970s of the last century. Apart from a common measurement of a fuel pressure by using piezoelectric sensors, at the inlet and outlet of a high pressure pipe, according to the research presented in [El-Erian, 1972] and [Wylie et.al., 1993] measurements of a fuel pressure have been carried out inside the injector, at the ring groove inside the injector holder, also at the volume around the needle, but also at the outlet of the nozzle. Considering that there are very narrow channels inside the injector, placing fuel pressure sensor at such locations is very complicated which causes different conditions for fuel flow through the inside of the injector. As a result of that the majority of models for defining fuel flows through an injector take as an initial and boundary condition measured pressure of a fuel at the outlet of a high pressure pipe, i.e. immediately before entrance of fuel into an injector. Modelling of the fuel flows process in an injector, with majority authors, is mainly based on introduction of an equivalent cross section flow rate and a corresponding coefficient of a fluid flow for whole the injector. This is carried out with the aim of defining the basic characteristics of the fuel injection such as: injection characteristics, fuel supply and duration of injection. Up until now all simulation models of a fuel flow through an injector, an inside of an injector have been simplified to that extent that calculated characteristics of a fuel flows in characteristic positions do not agree with the real results. Scientific agreement on this problem can be found in the latest research presented in [Kiijärvi, 2003], where more modern fuel injection system with diesel engines have been analysed. However, as in the previous research defining the fuel energy potential across an injector has remained undefined.

Although currently there are a large number of commercially available numerical programmes based on the finite volumes method (*AVL Fire*, *Fluent*, *STAR-CD*, *KIVA*, etc.), which have been improving all the time, due to its three-dimensional (3D) geometrical complexity with respect to very narrow orifices through fuel flows, existing of moving parts of an injector that requires moving numerical mesh, cavitations, necessary numerical computation time, etc., a solving domain of fuel flows in the injector is mostly based on a sac volume and one or more nozzles for injecting of fuel into the combustion chamber of diesel engine.

Nevertheless, majority of the current models for a fuel flow simulation at an injector are still being based on one dimensional (1D) mathematical models for solving continuity equation and the equation of motion. The advantage of these models which can be found in the commercial programmes, such as *AVL Hydsim*, *AMESim*, etc. [Chaufour et.al., 2004] and [Favennec et.al., 1999], but also personal programmes of individual researches [Kiijärvi, 2003], [Chaufour et.al., 2004], [Gullaksen et.al., 2004], and [Filipovic, 1983], can be seen in the speed of problem solving with great precision, considering that domains of a problem solving for fuel flow, i.e. channels and orifices has one dominant dimension. The research in this field has also been directed towards nozzle shape and defining a coefficient of a fuel flow between the needle and the needle seat, as well as at nozzles.

In order to define hydro-dynamics characteristics of fuel, i.e. pressure and velocity in every single point inside the injector, that is the basis for definition of fuel energy transformation, the universal one dimensional (1D) mathematical model of whole the injector has been developed.

INJECTOR MODELLING PROCESS

With the aim of successful conduction of a fuel flow simulation in an injector, the modelling process has been divided as follows: modelling of design parameters inside the injector, modelling of movements of some parts of an injector and modelling of a fuel flow through the injector with a special attention to fuel flows on different pipe diameters connection, orifices and volumes.

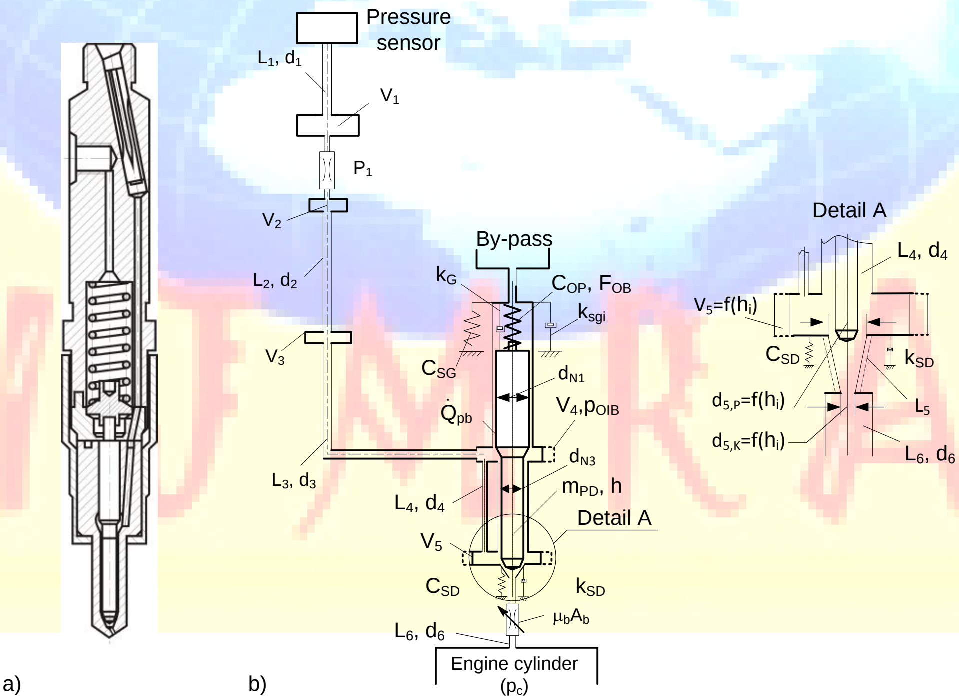
In the research a relatively old injector has been used by a company Bosch which comprises of an injector holder of KDAL 80S20/129 and an injector body of DLL 25S834, that is mainly use for conventional diesel engines with M-Type formation of working mixture and for propulsion of heavy duty vehicles and busses. The main reason for selection of this injector is the possibility for presenting universal modelling process of very complex design parameters inside the injector.

Physical modelling of the injector

Inside the injector (Figure 1.a), through which fuel flows, has been replaced with a model presented in the Figure 1.b). Besides simple construction parts of inside the injector, special attention has been dedicated to modelling of: filter, ring shape volume around the needle, ring volume in the injector body, conical orifice between needle and needle seat, where the latest are characterised by changing volume and diameter due to needle movement.

The filter which is presented in the cross section in Figure 1.a), has a very complex design and comprises from 3 inlet and 3 outlet channels which are connected with narrow orifice.

Considering that fuel which flows through an inlet channel can flow through the two adjacent outlet channels, with this model it is adopted that fuel which enters one inlet channel, through a narrow orifice, flows to one adjacent outlet channel. In this way it is possible to form the simplest model of the filter which is based on pipes of circular cross section whose equivalent area of cross section is equal to the total cross area of 3 inlet and 3 outlet channels. According to the author's experiences, this model does not significantly increase the quality of output parameters from the filter (P1), such that with a great precision of results a model of the filter can be used as presented in Figure 1.b). Similarly, as in the case of the filter, it is possible to form a ring model around an injector needle. Considering that in both cases fuel flows through the pipes which are not of circular cross section, it is required to evaluate hydraulic diameter of pipes in order to determine Reynolds number. This can be done with a known area of a pipe diameter and a known wetted area of a pipe cross section. Thus, the coefficients of losses through these sections can be obtained more precisely.



a) b)

Figure 1. Injector a) and physical model b) injector with basic marks

The greatest attention during modelling process of design parameters inside the injector has been dedicated to a final section of an injector. This comprises an area between conical orifice between needle and needle seat during injection process, as well as to the sac volume and the nozzle, as presented in the Figure 1, detail A. All dotted lines present changes in the characteristic values of $d_{5,P}$, $d_{5,K}$ and V_5.

An overview of all geometrical values of the injector physical model presented in Figure 1.b), (i.e. pipes, channels, orifices, volumes), that are necessary for carrying out planned research can be found in the [Pikula, 2007].

Mathematical modelling of needle moving parts dynamic

The known "elastic needle seat", presented in [Filipovic, 1983], is used for modelling of needle lift. This model enables modelling of needle oscillations which exist at making contact of needle with upper and lower needle seats (contacts on the injector body). Mathematical model of a needle dynamic of an injector can be described in the following expression:

$$m_{PD}\frac{d^2h}{dt^2}+k\frac{dh}{dt}+Ch=F_O+\sum p_iS_i \tag{1}$$

where the parameters, depending on needle lift, have the following meanings:

$$\begin{array}{llll} a)\ \text{for } h<0 & k=k_{SD} & C=C_{SD} & F_O=F_{OB} \\ b)\ \text{for } 0\leq h\leq h_{max} & k=k_G & C=C_{OP} & F_O=F_{OB} \\ c)\ \text{for } h>h_{max} & k=k_{SG} & C=C_{SG} & F_O=F_{OB}+(h-h_{max})C_{SG} \end{array} \tag{2}$$

while p_i is the pressure and S_i is the needle surface at i - location. Exact values of mentioned parameters, based on recommendations in the literature [Cernej, 1987], are given in [Pikula, 2007].

Mathematical modelling of fuel flows inside the injector

In the Figure 1 when analysing physical model of geometric and design parameters of the injector (Figure 1.b) and flow flows inside the injector, it can be shown that solving the fuel flow process should be based on solving each of the following cases:

- fuel flow through various pipes (a high pressure pipe, a pipe which represents inlet channels of filter, a pipe which represents outlet channels of filter, a pipe in injector holder, a pipe in injector body, a ring pipe around needle a pipe which represents conical orifice and a nozzle)
- fuel flow across blends (a blend between inlet and outlet channels of the filter, a blend between conical orifice and sac volume) and

- fuel flow in volumes (a volume before the filter, a volume after the filter, a ring volume inside the injector body, a volume around the needle, sac volume).

This method enables that a very complex design of inside the injector is considered as a modular system made of pipes, blends and volumes. For fuel flow in pipes it is required to define a pressure and a fuel speed in each pipe section, whilst for fuel flow in volumes it is required to define a pressure in a volume. Fuel flow across blends presents a special case of two pipes joint for which as with a joint of pipe-volume-pipe it is required to define a boundary condition.

Considering certain specifics of fuel flow through (across) a section of high pressure pipe and an injector, one dimensional mathematical model (1D) has been developed, where fuel flow has been considered as unsteady, isentropic and compressible flow. Fuel flow specifics are in line with: the model of geometric parameters and small ratio between diameter and length of the pipe ($d << l$), turbulent flow, negligible small radial pressure change when compared with pipe proportions particularly in injector holder and injector body.

In order to find a solution for hydro-dynamics characteristics of fuel inside the injector (pipes) it is necessary to solve an equation of continuity and an equation of motion for any simple control volume. Using approach presented in [Wylie et.al., 1993], these equations can be defined as:

$$W_1 = \frac{\partial p}{\partial t} + \rho a^2 \frac{\partial v}{\partial x} = 0 \tag{3}$$

$$W_2 = \frac{\partial v}{\partial t} + \frac{1}{\rho}\frac{\partial p}{\partial x} + \frac{fv|v|}{2d} = 0 \tag{4}$$

Having in mind the conical part between needle and needle seat (Detail A, Figure 1), for conical pipe the following equations were used:

$$W_1 = \frac{\partial p}{\partial t} + a^2 \rho \frac{\partial v}{\partial x} \pm \frac{2a^2 \rho v f}{d} = 0, \tag{5}$$

$$W_2 = \frac{\partial v}{\partial t} + v\frac{\partial v}{\partial x} + \frac{1}{\rho}\frac{\partial p}{\partial x} + \frac{fv|v|}{2d} = 0. \tag{6}$$

Equations (3 and 4) as well as (5 and 6) are solved by the method of characteristics. All hydraulic losses of local and line character were calculated from the literature [Pikula, 2007].

Starting from integral form of continuity equation and using expression for fluid compressibility the final change of pressure (p_Z) in the volume (V) can be described by the following expression:

$$\frac{dp_Z}{dt}\frac{V}{E} = Q_i - Q_{i+1} \tag{7}$$

VERIFICATION OF MATHEMATICAL MODEL

With the aim of verifying mathematical model, measurements of characteristic systems for the fuel injection has been carried out comprising of a linear six-cylindrical high pressure pump of BOSCH manufacturer of PES 6A 95D 410 LS 2542 type with piston diameter of 9.5 *mm* and lift of 8 *mm*. The high pressure pump is connected via the high pressure pipe (of outer diameter of 6 *mm* and inner diameter of 1.8 *mm* and the total length of 1024 *mm)* with injector holder of *KDAL 80S20/129* type *and* injector body of *DLL 25S834* type. The injector holder is characterised by only one nozzle with a diameter of 0.68 *mm* and with the maximum needle lift of 0.3 *mm*. For the purpose of mathematical model verification, an experimental set up was installed and shown in the Figure 2.

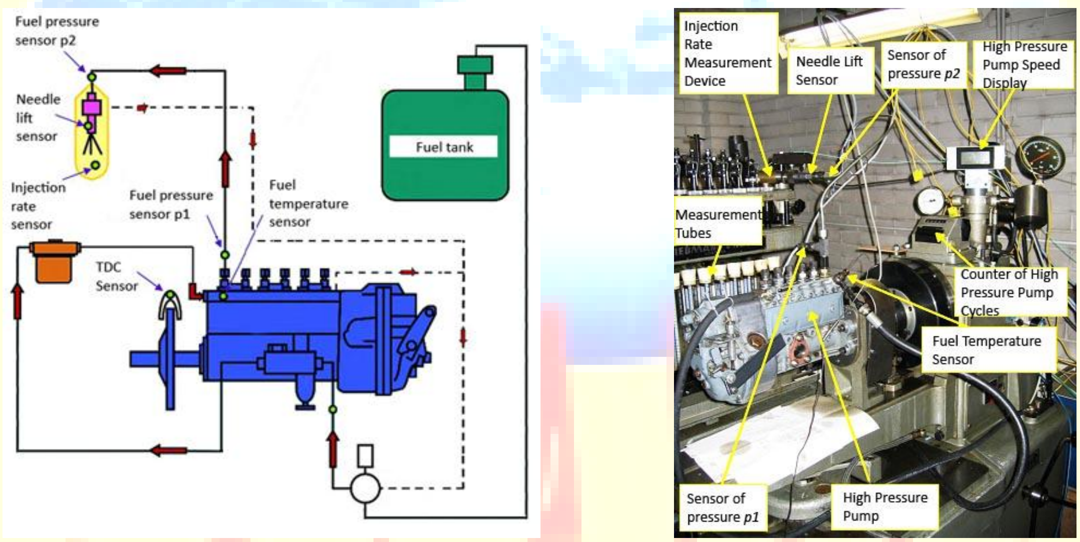


Figure 2. Schematic view of the fuel injection system with marked locations of measured points

Fuel temperature in the low pressure collector of the high pressure pump has been measured using a thermo-element of *OMEGA* type *K* manufacturer, (diameter of 1.6 *mm* and length of 250 *mm*), with measurement range -200 *ºC* up to +600 *ºC*, accuracy ± 1 *ºC* and sensitivity 0,039 *mV/ºC*. For recording of pressure p_1 at the high pressure pipe inlet, located on 35 mm from the outlet of high pressure pump, strain gauge device of *AVL* manufacturer of *31 DP 1200 E-1.8* type has been used. A measurement range of this pressure sensor is 0-1200 *bar* with accuracy ± 1.5 %. On the other side, a pressure at the outlet of the high pressure pipe immediately before the injector, located on 134.5 mm before injector, has been recorded by piezoelectric sensor of *KISTLER* manufacturer of *6227* type, with measurement range 0-2000 bar in wide temperature range -50 *ºC* up to 200 *ºC* and sensitivity 2.5 pC/bar. Needle lift was measured by inductive sensor *AP 5,5/1,0*. (external diameter 5.4 *mm*, core diameter 1.1 *mm* and length 20 *mm*) with 2x17.4 *Ω* resistance. The measurement tubes and counter of high pressure pump cycles are presented in the Figure 5. Knowing a total injected quantity of fuel and number of high pressure pump cycles, the exact fuel quantity per cycle and cylinder can be defined.

A special focus has been paid to measurement of the injection characteristics using piezoelectric sensor during defining experimental setup. For this purpose a special housing has been designed, explained in [Pikula, 2007], which enables connection of all elements in one unit. On the left hand side of the housing, there is an injector with needle lift sensor, whilst the right hand side has been prepared setting piezoelectric sensor of *KISTLER* manufacturer, of *7031* type, (diameter 9.5 *mm* and length 12 *mm*), measurement range 0÷250 *bar* in wide temperature range -196 *ºC* up to 200 *ºC* and sensitivity 55 *pC/bar*. The housing has an opening from the bottom side through which fuel flows without delay towards measurement tubes. Piezoelectric sensor at the distance of only 11*mm* from the injector nozzle makes possible to fuel jet a directly hit on the surface area of piezoelectric sensor at an angle of 30 DEG and it then bounces towards the opening of measurement tube. It can be concluded that after placing all elements inside a special housing, there is enough volume which does not allow collection of fuel and it prevents possibility of an error.

Physical characteristics of fuel such as: density, bulk modulus of elasticity, viscosity and sound velocity through a fuel, depend on a pressure and a temperature for a given fuel. Based on conducted experiments, physical characteristics of fuel according are defined in [Pikula, 2007] and [Pogorevc, 2006].

Working fuel temperature in the fuel injection system has been in the range of 35÷38 *ºC*. Therefore, it can be assumed that an average fuel temperature is of a constant value. A dominant effect on the fuel physical characteristics change has a change of a fuel pressure which for a used injection system is in the range from 1 to 500 *bar*.

A value for the pressure at the end of the high pressure pipe p_2 obtained by the experiment and as shown in the Figure 3 (dashed line), has represented an input data i.e. an inlet boundary condition for the simulation model. On the other hand, an outlet boundary condition at the end of a nozzle has been defined by a surrounding pressure to where a fuel is injected through injector nozzle. It should be noted that the initial conditions for simulation in the whole model have been defined by an initial pressure i.e. “remaining pressure” (static pressure) between the two injection processes which are also obtained based on experimental results. The boundary conditions for simulation of the whole model, which refer to a flow, have been defined with an assumption that an initial flow velocity equals to zero (i.e. no flow).

The results, presented in the Figure 3, have been obtained using the computer programme for simulation developed on the basis of the previous mathematical model. These results enable to define values for pressures and fuel flows as well as the integral characteristics of the injection system across the high pressure pipe, i.e. from a location of placing of piezoelectric sensor of fuel pressure p_2 and through inside the injector at each point in time. The most important characteristics of the fuel injection system like injected quantity (fuel supply) and the duration of injection for different load and speed regimes, have been obtained by the experiment and simulation and presented in [Pikula, 2007].

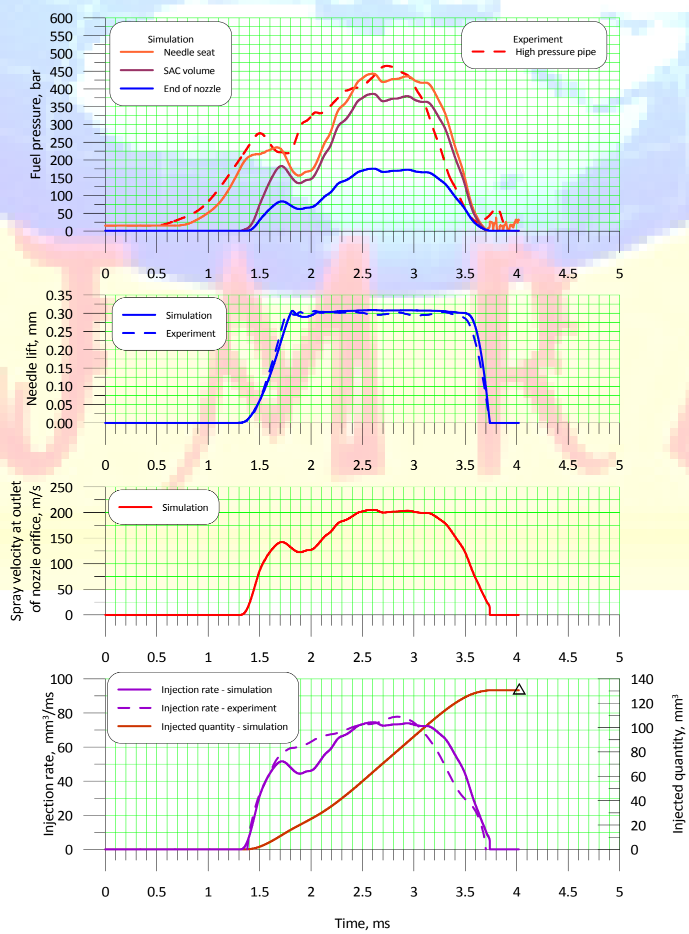


Figure 3. Characteristic parameters during the fuel injection process D2 at 1100 min^{-1} of high pressure pump and cycle injected quantity of 132 mm^3

TRANSFORMATION OF FUEL ENERGY ALONG INJECTOR

A numerical program developed on basis of the mathematical model and used for simulation can calculate the fuel flows inside the injector, the needle movement and the definition of energy potential in order to show transformation of the fuel energy along the injector. Having in mind the Bernoulli equation in following form:

$$p+\rho\frac{v^2}{2}=const. \tag{8}$$

it is easy to define energy potential (level) in every single node of the 1D computation grid inside the injector in every moment of time. Since energy level of diesel fuel inside the injector can be presented in form of the potential energy, that is pressure of fuel, as well as in form of the kinetic energy due to velocity of fuel flows, it is interesting to show values of the potential and the kinetic energy along inside the injector. The energy potential (level) along the injector in the moment of achieving the maximum of the fuel pressure at the end of high pressure pipe, in case of the maximum of the high pressure pump speed, is presented in the Figure 4.

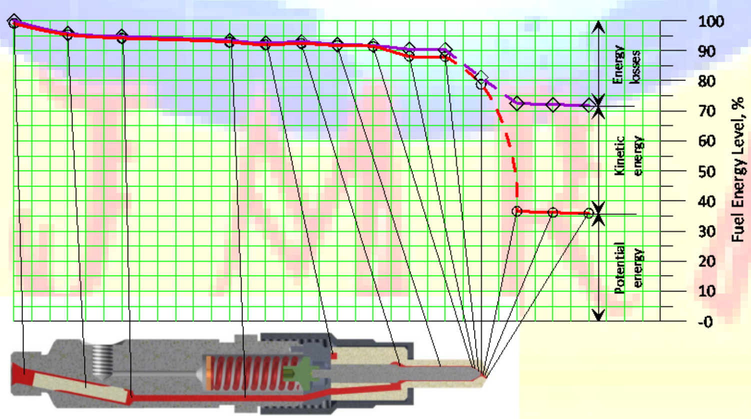


Figure 4. Transformation of fuel energy along the injector at maximum value of fuel pressure at the end of high pressure pipe and at the maximum of high pressure pump speed of 1100 rpm

Having in mind the results of the energy potential of fuel along the injector, presented in the Figure 4, one can notice a decreasing of the energy potential along the injector due to friction and local losses, generally losses during fuel flows through the injector. From the beginning of the injector and up to location of the needle seat, diesel fuel has mostly the potential energy. During fuel flows through the conical orifice between the needle and the needle seat as well as through the SAC volume, a significant transformation of fuel energy from the potential to the kinetic energy was obtained. Also, it is very important to notice that the greatest transformation of fuel energy and the greatest losses in the same time are registered just in this area. Similar conclusions could be derived for other speed regimes of the high pressure pump. Furthermore, an analysis of the results presented in the Figure 4, the difference between total fuel energy at the inlet of the injector and sum of the potential and the kinetic energy at the outlet of the injector could be noticed. The mentioned difference is called "losses" that can be express through absolute values, as a pressure, or through relative values, as ratio between temporally value of fuel pressure along the injector and total fuel energy at the inlet of the injector.

CONCLUSION

This paper presents a universal one dimensional mathematical model for solving fuel flows across the whole inside the injector. This has been carried out by taking into consideration one dimensional models of injectors based on solving fuel flow through an effective flow cross section defined with a critical geometrical flow cross section and an equivalent coefficient of fuel flow across the whole injector. In addition, consideration has been taken of modern 3D models of injectors which are still based on solving fuel flow in final part of injector or more exactly to structure of needle seat – sac volume – nozzle.

One dimensional mathematical model presented in this paper enables solving fuel flows through inside the injector based on simple elements such as pipes, blends and pipe-volume junctions. Fuel flow solution at each point of inside the injector has been obtained by a method of characteristics which defines energy losses during fuel flow across the injector. In case of one conventional diesel fuel injection system that is still in use at heavy duty commercial vehicles, the value of energy losses in the injector is up to 30 %. The greatest transformation of the diesel fuel energy, from the potential to the kinetic, and the greatest energy losses are located at the inlet of the nozzle orifice. On this way, characteristic places inside the injector where special care has to be paid during design of new geometric shapes are shown.

Very good initial conditions for calculating the characteristics of dispersed fuel can be obtained with this model, which up until now was not the case.

As a result of very complex form of inside the injector and universality of the model, the identical computer programme can be used for calculating the parameters of the complete injection system.

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Nomenclature

a Speed of sound

C	Stiffness
d	Pipe diameter
D	Diameter
d_H	Hydraulic diameter
E	Bulk modulus of elasticity of fuel
F	Force
f	Darcy-Weisbach friction coefficient
g	Acceleration due to gravity
h	Needle lift
k	Damping coefficient
L	Pipe length
m	Mass
p	Pressure
Q	Flow rate
S	Surface
t	Time
V	Volume
v	Speed
x	Longitudinal coordinate of model and injector

Greek letters

μ	Flow coefficient
ρ	Density

Subscripts

N	Needle
OIB	Around injector needle
OB	Injector spring
PD	Moving parts of injector
SAC	Sac volume between needle and nozzle
SD	Lower needle seat
SG	Upper needle seat
Z	Related to volume