Diesel Injection-System Simulation at Part Loads under Steady-State and Transient Operations

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ABSTRACT

The NAIS code, which had been previously upgraded for numerical simulation of high-pressure fuel-injection-system dynamics under control, was further developed and applied to investigate the unsteady flow phenomena in an automotive diesel injection equipment with a distributor-type pump, at several loads and pump speeds, over a wide range of steady and transient operating conditions. In predicting the system behavior at the extremities of this range, it was necessary to solve some problems arising from the occurrence of critical conditions there. Particular attention was given to the intense cavitation and to the residual effects it might have on fluid properties, in the cycle-resolved analysis of transient control-phase operation at high pump speed.

A cycle-by-cycle experimental investigation was also performed on a test bench that is usually used by industry for assessing diesel injection equipment. The experimental and theoretical results were compared and discussed.

INTRODUCTION

The research in the automotive engine field is oriented towards a major reduction of fuel consumption and pollutant emission at part loads and under transient operation. High-pressure fuel injection is one of the primary diesel engine technology which will be enhanced in the near future to that end ([1-5]1). An accurate, efficient and flexible program for high-pressure injection-system simulation in both steady and unsteady operating conditions can provide a valuable means of investigating local flow phenomena in series-production systems and developing new concept design.

Many numerical models, usually based on the first-order accurate method of characteristics, have been applied to the analysis of diesel injection systems at full load under steady operations. An extensive literature review is reported in [6]. However, there is a lack of both experimental and theoretical studies on these systems at part loads under stationary and, more important, transient working conditions.

The recently upgraded NAIS computational code for diesel injection-system dynamics simulation ([7]), has further been developed and applied to analyze the unsteady flow phenomena in a DI automotive diesel injection system with a distributor-type pump, at several loads and pump speeds over a wide range of stationary and transient operations. Specific care was devoted to the prediction of critical conditions at the boundaries of this range, such as, for example, a relatively strong pipe-cavitation-flow, in the cycle-resolved analysis of transient control-phase operation at high pump speed.

The program, based on an efficient numerical algorithm with a second-order accuracy, is given a modular structure by a library containing a variety of component models, so that different pump-line-nozzle injection systems, with in-line or distributor pumps, several delivery-valve types, single- and dual-stage injectors, reduced-sac or sacless nozzles, as well as different control equipment, can be simulated. It takes the fuel-property dependence on pressure and temperature, in addition to the friction and minor losses, into account and is also capable of modeling the possible cavitation occurrence and propagation. Nonstationary test-derived flow coefficients were also employed for the injector-nozzle holes.

In order to assess the validity of the numerical results, cycle-resolved experimental investigations were also carried out on a test bench of industrial relevance.

The considered injection system was made up of a Bosch VER distributor pump, delivery valves of the reflux type and single-stage reduced-sac-nozzle injectors. It had a minimum-maximum speed governor with part- and full-load torque control. Although new electronically-controlled injectors are under development ([4]) at present, this pump-line-nozzle equipment was studied because it was the only automotive injection system available for experimentation and result divulgation. Nevertheless, it is still of interest for light-duty engines, as is shown by the continuous evolution it is being subjected to ([3]) and also by the fact that an electronically governed distributor pump is planned for the near future ([1]). It was also intended to show the potentialities of the code, which is of more general application, and to allow an insight into the details of the unsteady flow throughout the system at part loads and in transient conditions, regardless of the specific control device which may cause them.

1 Numbers in square brackets designate references at the end of paper.
INJECTION SYSTEM

The commercial fuel-injection system was equipped with the Bosch VER 286/1 distributor-pump for automotive diesel engines ([8]), combining several assemblies: high-pressure distributor pump, vane-type supply pump, hydraulic timing device and mechanical speed governor. The main pump body is drawn in Fig. 1, giving evidence to the plunger and to the delivery-valve assembly. The fuel is delivered from the single internal high-pressure chamber, which is located between the plunger and the cylinder head, to the injector through the delivery valve and pipe.

Figure 2 shows the reflux valve which was fitted to the pump delivery assembly. In order to avoid the injector-nozzle reopening, this valve presents a flat in the collar, forming a return-flow restriction with the seat, so as to attenuate the pressure-wave reflection at the valve head during the valve closure, consequent to the spill-port opening.

The single-spring injectors employed were characterized by a Bosch DLLA 160 P 171 four-hole nozzle with a reduced sac chamber (Fig. 3).

The fuel-injection rate was controlled by the mechanical speed governor on the basis of engine speed and control-lever position. The speed governor is shown by Fig. 4 in its basic configuration (the reference numbers are those reported in [8]). However, the present automotive diesel injection system includes minimum-maximum speed-governor assembly with part-load and negative full-load torque-control devices.

MODEL EQUATIONS

Pipe unsteady-flow

The equations governing the unsteady flow in pipes were developed and discussed elsewhere ([6,9,10]). Nevertheless, it is worthwhile to recapitulate the pipe-flow model and the related assumptions. The fluid flowing through the pipes is usually in the liquid state, but whenever cavitation occurs it becomes a mixture of liquid and gaseous, or vapor, phases.

Therefore, the pipe-flow equations are written, in general, for a homogeneous, bubbly, two-phase fluid, so that the pure-liquid flow equations can be directly derived as a particular case. Cavitation is described by a dependent variable termed void fraction that is defined as the volume of the vaporous component per unit volume of the mixture. Accordingly, the momentum interchange between vapor and liquid phases is ignored so that the vapor bubbles and the liquid possess the same velocity. A practically isothermal flow is considered, requiring only mass and momentum conservation equations for simulation. This assumption is consistent with the fact that an average fluid temperature could be considered along the delivery pipe and the injector drilled passages without any sensible accuracy loss ([11-13]). Besides, in the presence of cavitation, negligible thermal effects could be ascribed to the fluid phase-change, because of the generally very small amount of liquid involved.

An average cross-sectional representation of mixture velocity, pressure, component densities and void fraction, is employed. Therefore, following a control volume approach,
with reference to a pipe element of cross-sectional area \( A \) and length \( \delta x \), made of elastic material, the conservation of mass for the vapor and liquid phases and the mixture momentum balance, for a horizontal pipe, are respectively given by

\[
\frac{\partial}{\partial t}(\rho_v A \delta x) + \frac{\partial}{\partial x}(\rho_v u A \delta x) = \Gamma A \delta x
\tag{1}
\]

\[
\frac{\partial}{\partial t}[(1 - \alpha)\rho_l A \delta x] + \frac{\partial}{\partial x}[(1 - \alpha)\rho_l u A \delta x] = -\Gamma A \delta x
\tag{2}
\]

\[
\frac{\partial}{\partial t}(\rho u A \delta x) + \frac{\partial}{\partial x}(\rho u^2 A \delta x) = -\frac{\partial P}{\partial x} A \delta x - \frac{4\tau_0}{d} A \delta x
\tag{3}
\]

where: \( t \) is the time variable and \( x \) the axial variable along the pipe, \( \alpha \) is the void fraction, \( \rho_v \) is the vapor density and \( \rho_l \) the liquid density, \( u \) is the mean velocity of the mixture along the pipe, \( \Gamma \) is a source term, that is the gas/vapor production rate per unit volume, \( P \) is the mean pressure, \( d \) is the pipe diameter, \( \tau_0 \) is the wall shear stress, and \( \rho \) is the mixture density expressed by \( \rho = \alpha \rho_v + (1 - \alpha) \rho_l \).

Equations (1)-(3) imply an axisymmetric deformability of the pipe with a negligible axial straining-velocity in comparison to the flow speed. Taking the elastic properties of the vapor, liquid and pipe material into account and rearranging, these equations can be put in the following matrix form [\( [H] \)]:

\[
\frac{\partial \mathbf{w}}{\partial t} + [A] \frac{\partial \mathbf{w}}{\partial x} = \mathbf{H}
\tag{4}
\]

where

\[
[A] = \begin{bmatrix}
 u & \frac{1}{\rho} & 0 \\
 -\rho \frac{d\alpha}{dx} & u & 0 \\
 \alpha & 1 - \frac{d\alpha^2}{\rho, \rho v} & 0 & u
\end{bmatrix}
\]

\[
H = \frac{\Delta T}{\rho d^2}
\]

\[
\begin{bmatrix}
\frac{\Gamma}{\rho_v} & \frac{\rho}{\rho_v} \\
\frac{\Gamma}{\rho_l} & \frac{\rho}{\rho_l} \\
\end{bmatrix}
\]

\[
\alpha_v, \alpha_l \quad \text{are the isothermal wave propagation speeds of the vapor and liquid respectively} \quad \text{[16].}
\]

The variable \( \alpha \) is given by

\[
\frac{1}{\rho a^2} = \frac{\alpha}{\rho_v a^2_v} + \frac{1 - \alpha}{\rho_l a^2_l}
\tag{5}
\]

and expresses the wave propagation speed of the mixture, as can be easily verified by determining the eigenvalues \( \lambda \) of the matrix \( [A] \): \( \lambda_{1,2} = u \pm a \), \( \lambda_3 = u \).

**Pump and injector**

Pressure wave-propagation phenomena were simulated in the internal and outlet lines of the distributor pump as well as in the injector drilled passages, according to the foregoing pipe-flow model. A conventional lumped mass model was applied to write the continuity and compressibility equations that allow for flow rates through the pump and injector isobaric-volumes, in the presence or not of cavitation [\( [10] \)]. With reference to the chamber \( j \), one can write

\[
\sum_{k} Q_{j,k} + \left( V_{0j} \pm \sum_{i} S_{m_i} x_i \right) \xi_j = \pm \sum_{i} S_{m_i} \frac{dx_i}{dt}
\tag{6}
\]

\[
\xi_j = \begin{cases}
\frac{1}{E_i} \frac{dp_j}{dt} & p_j > p_v \\
\left( \frac{p_j}{p_v - p_i} + \alpha_j \right)^{-1} \frac{d\alpha_j}{dt} & p_j = p_v
\end{cases}
\]

where: \( \sum_{k} Q_{j,k} \) expresses the net admitted volume flow-rate, \( V_{0j} \pm \sum_{i} S_{m_i} x_i \) \( \xi_j \) is the accumulation term, \( \pm \sum_{i} S_{m_i} \frac{dx_i}{dt} \) designates the chamber volume reduction or increase due to the displacement \( x_i \) of the mobile-element surface \( S_{m_i} \), \( E_i \) is the liquid elasticity-modulus and \( p_v \) is the vapor pressure.

The dynamics of delivery valve and injector needle was simulated by a second-order linear-system model with one degree of freedom given by the ordinary differential equation

\[
m_m \frac{d^2 x_m}{dt^2} + \beta_m \frac{dx_m}{dt} + k_m x_m + F_0 = \pm \sum_{i} p_i S_{m_i}
\tag{7}
\]

where: the first term on the left side of Eq. (7) indicates the inertia contribution to the dynamic equilibrium of the mobile element with mass \( m_m \), the second left-hand term expresses the viscous action, \( k_m x_m + F_0 \) are due to the elastic forces, the right-hand term designates the fluid pressure action on the surface \( S_{m_i} \) of the mobile element. Therefore, a set of ordinary and partial differential equations resulted from the pump and injector models, as those reported in [\( [10,12] \)].

**Speed governor**

The dynamic system model of Fig. 5 was used for speed-governor simulation. This has two degrees of freedom, as can be deduced by considering the two independent variables \( \theta_r \) and \( \theta_l \) that express the angular positions of the starting and tensioning levers (71 and 72 in the figure), respectively.

The minimum-maximum speed-governor assembly was treated as an elastic element \( K \) with the nonlinear characteristic reported in [\( [7] \)]. This is given by the force \( F_r \), applied to the lever (72) in \( T \), as a function of the extension of the minimum-maximum speed-governor assembly.

In the same way, the effect of the negative full-load torque control was considered through the application of the equivalent moment \( M_{ef} \) to the starting and tensioning levers and through the introduction of the time dependent angle \( \gamma_s \). This is the angle between the lines which connect the pivot \( M_z \) of the levers (71,72) to the point \( R \) and to the ball pin \( S \), respectively. The ball pin \( S \) is located on the starting lever (71) where this joins the control collar. The distributions of
were determined through the application of the virtual-work principle to the control system. The stroke ends \((j = 1, 2, 3\) in Fig. 5), which limit the flyweight (67) and tensioning lever (72) movements, were regarded as perfectly elastic and with the same effective stiffness value \(k_w\) ([7]). The viscous-loss equivalent moments \(M_R\) and \(M_T\) are taken to be linearly dependent on the angular speeds \(\vartheta_R\) and \(\vartheta_T\). Further details on the speed-governor model equations are given in [7].

**NUMERICAL ALGORITHM**

The three-time level \(\text{TSBT} (\text{Trapezoidal Space, Backward Time})\) implicit finite-difference scheme of second-order accuracy, proposed in [11], was used to discretize the pipeline partial differential equations (4). In order to improve the algorithm efficiency, an automatic time-step size selection procedure was applied ([14]). Then, for a computational grid with variable adaptive time steps, the TSBT scheme yields

\[
\begin{align*}
1 + \frac{2\xi}{1 + \xi} [I] &+ \frac{\Delta t}{\Delta x} \left[(A)^{n+1}_j + (A)^{n+1}_{j-1}\right] w^{n+1}_j = \\
\frac{1 + 2\xi}{1 + \xi} [I] &- \frac{\Delta t}{\Delta x} \left[(A)^{n+1}_j + (A)^{n+1}_{j+1}\right] w^{n+1}_{j+1} = \\
(1 + \xi) \left(w^{n}_j + w^{n-1}_j\right) - \frac{\xi^2}{1 + \xi} \left(w^{n-1}_j + w^{n-1}_{j+1}\right) +
\end{align*}
\]

where the time-stepping coordinate in the grid is indicated by the superscript and the spatial coordinate by the subscript, \(\Delta t = t^{n+1} - t^n\) and \(\Delta x = x_j - x_{j-1}\) being the current time- and space-interval of the grid, respectively. \(\xi\) is the ratio between the current time step \(\Delta t\) and the preceding one \(\Delta t^{n+1} = t^n - t^{n-1}\).

To discretize the ordinary differential equations which model the speed-governor, pump and injector dynamics, BDF (Backward Differentiation Formulas) implicit multistep schemes of the second-order accuracy were selected, being suitable for problems of the stiff type ([6,9]). In particular, a second-order BDF approximation to the first-order time derivative of any time-function \(y\), at the instant \(n+1\), in a variable time-step computational grid is given by

\[
\frac{dy^{n+1}}{dt} = \frac{1}{\Delta t} \left[1 + \frac{2\xi}{1 + \xi} y^{n+1} - \left(1 + \frac{\xi^2}{1 + \xi}\right) y^n\right]
\]

The difference approximation to the partial differential equations of the pipe flow and to the ordinary differential equations of the speed governor, pump and injector led to a nonlinear algebraic equation-system, which can be expressed in vector notation as

\[
F(x) = 0
\]

\(^2\) Sometimes, it is also referred to as a two-step scheme, with reference to the time-stepping coordinate ([12]).
\( x \) being the vector of unknown variables. This system was solved using the Newton-Raphson method, which transforms it into the following linear form

\[
\begin{bmatrix}
\Delta x^{(i)}
\end{bmatrix}
= - F\left( x^{(i)} \right)
\]

where \( x^{(i)} \) is an iterative approximation of the exact solution, \( F(x^{(i)}) \) is the system Jacobian matrix evaluated at \( x^{(i)} \) and \( \Delta x^{(i)} \) is the iterative correction so that \( x^{(i+1)} = x^{(i)} + \Delta x^{(i)} \).

The linear equation system (13) was resolved using a very efficient direct method, taking advantage of the multidiagonal structure of the system coefficient matrix \([9, 10]\). Such a method can be considered to be an extension of the Thomas algorithm to the multidiagonal form of the linear system (13).

**EXPERIMENTAL SYSTEM**

A Hartridge 2500 test-bench, supplied with standard oil ISO 4113 as working fluid, was equipped and instrumented at Fiat Research Center for investigating the injection system behavior at different loads and pump speeds under stationary operating conditions \([10, 12]\). The test bench was also set up and applied for experimental simulations of transient system operation under control. These were carried out by giving a programmed rapid movement to the pump control lever from its idle position to the full-load position, through an electric actuator, at the constant pump angular-speeds of 1000 rpm and 2000 rpm, respectively. During the steady and transient operations, pressure time-histories were acquired cycle-by-cycle in the pumping chamber and at the extremities of the delivery-pipe. Injector needle-lift and injection-rate distributions versus the pump-shaft crank angle were also measured in the stationary operating conditions.

Pressures were measured with piezoresistive transducers, the needle lift data were acquired by means of an inductive transducer and the injection rate was determined using a modified Bosch indicator, as reported in [15]. A schematic of the experimental system layout with the measured quantities and locations, is shown in Fig. 6.

Based on previous experience \([12]\) and on the average deviations of repeated measurement sets, by cumulating the effects of the different uncertainty sources, it was prudential to assign a maximum overall uncertainty of ±12% to the experimental data, in general. However, this did not influence the result involvements and conclusions.

**RESULTS**

Figure 7 shows the static-characteristic response of the pump speed governor under stationary operating conditions. This reports the effective pump-plunger stroke \( s_u \) versus the pump speed \( n \) at the loads indicated by percent numbers, 100% corresponding to the full load. The test-derived results are reported with circular symbols, while numerical results are plotted by lines. The dash-dot line shows the minimum effective pump-plunger stroke that is necessary to open the injector. The blackened symbols represent the steady-state part- and full-load conditions which were numerically and experimentally analyzed. During these operation tests, the time-histories of the following quantities were simultaneously recorded by means of a high-frequency automatic data-acquisition system: the pressures in the pumping chamber \( (p_c) \) and at the extremity of the delivery-pipe close to the pump delivery-valve assembly \( (p_d) \), the injector needle-valve lift \( (l_e) \) and the injected flow-rate \( (Q) \). The ensemble average of these data over 50 injection cycles was taken.

Figure 8 compares numerical and experimental results obtained for 100% (full load), 60% and 40% loads, at the pump speeds of 1500 rpm (Fig. 8a) and 750 rpm (Fig. 8b) in steady-state conditions. The line curves show the numerical results and the symbols represent the experimental data. The figure plots the pressures \( p_c \) and \( p_d \), the needle lift \( l_e \) and the injection rate \( Q \) versus the pump-shaft crank angle \( \theta - \delta_0 \), \( \delta_0 \) being a reference crank angle. The fundamentally good agreement shown by the computed and measured quantities substantiates the validity of the NAJS simulation code at the engine part-loads. Fig. 9 reports the distributions of \( p_c, p_d, l_e \) and \( Q \) obtained for the pump control-lever positions corresponding to the full load and to the idling engine operation with minimum injection of fuel, that is 0% load, at the very low pump speed of 250 rpm. As expected, at this speed the \( p_d \) patterns were closer to the \( p_c \) crank-angle histories than those at higher speeds. Besides, a more gradual opening of the injector nozzle can be inferred from the \( l_e \) distributions. The consistency of theoretical and experimental results in

**Fig. 6 Measurement system layout.**

**Fig. 7 Speed-governor static-characteristic response.**
Fig. 8 Numerical versus experimental results at part and full loads under steady-state operations.
and 2000 rpm, respectively, on the test-bench. At each pump speed, the actuator moved the pump control lever from its idle position to the full-load position, which were at the extremities of the lever swivel range, so as to automatically reproduce a rapid and realistic control phase. Fig. 10 plots the computed time histories of the effective pump-plunger stroke (solid lines) set by the speed-governor in response to the actual engine-load variations (dashed lines). The gray bands show the pump crank-angle intervals ($\Theta - \Theta_0 = 20$-50 deg at $n = 1000$ rpm and $\Theta - \Theta_0 = 10$-60 deg at $n = 2000$ rpm) which the numerical and experimental results of Fig. 11 relate to, for the indicated cycles. Such intervals correspond to those cycle portions where the fuel delivery occurred through the specific pipeline and injector which were instrumented with pressure transducers. During the transient control-phase tests the following data records were simultaneously acquired by the four channels of the available acquisition system: the pump control-lever position, the pressure in the pumping chamber ($p_c$) and the pressures at the extremities of the distribution pipe, that is, close to the pump delivery-valve assembly ($p_{pd}$) and at the injector inlet ($p_{in}$).

Fig. 9 Numerical versus experimental results at idling and full-load engine steady-state conditions.

this case shows the good predictive capability of the code even at the boundary of the pump operating speed-range. The computed patterns of $I_n$ and $Q$ versus the experimental ones indicates the care that should be devoted to the unsteadiness of the nozzle-hole discharge flow-coefficients ([13,15]).

The simulation program of the whole injection system, including the speed governor, was also applied to predict the system response to the induced transient loads at 1000 rpm and 2000 rpm. The predicted speed-governor response (solid line) to the experimentally simulated (dashed line) engine-load variation from idle to full load.

Fig. 10 Predicted speed-governor response (solid line) to the experimentally simulated (dashed line) engine-load variation from idle to full load.
Fig. 11 Computed and measured cycle-resolved pressure distributions during the transient-load tests.

Figure 11 compares the theoretical and experimental distributions of $p_c$, $p_{pd}$ and $p_{pi}$ in the specified cycles, with reference to the dynamic-load conditions of Fig. 10. These distributions basically show a very good agreement between the numerical results and measured data. Therefore, the validity of the injection-system computational model in accurately predicting transitory operations was confirmed for the more critical conditions of the present system dynamics investigation under control, completing the previous study [7]. In this latter, an almost nil cavitation inducing delivery valve of the constant-pressure type was fitted to the pump. Lower pump speeds (500 rpm and 1500 rpm) were considered and the
lever movement was actuated without any lasting, though
gradual, load increase in its final stage. A rather intensely
cavitating pipe-flow can be inferred from the \( P_{m} \) patterns of
Fig. 11b, mainly for low loads, after the injector-nozzle
closure. Particular care was exercised in predicting the cycle-
by-cycle temporal evolution of such flow anomaly that is
quite likely to take place in high-pressure injection systems.
Nevertheless, a virtually negligible residual void-fraction
effect on the fluid elasticity modulus was predicted and
experimentally observed in subsequent cycles during the
pump compression stroke.

For \( n = 1000 \) rpm (Fig. 11a), the pressure time-histories
at each location during the first full-load cycle are practically
the same as those in the tenth full-load cycle, meaning that
the system adapts to the steady-state operating conditions in
a few cycles, soon after the effective pump-plunger stroke has
come into its full-load position. A similar deduction was also
drawn for \( n = 2000 \) rpm (Fig. 11b). The main reasons of this
system behavior base on the small sensitivity of the delivery-
pipe residual pressure to significant load variations allowed
by the reflux-type valve, and on almost nil lasting cavitation
effects during the pump compression stroke.

No fuel injection was predicted or measured at the idle
position of the pump control lever, because the pipe-pressure
increase (Fig. 11) was not sufficient to open the injector.

CONCLUSION

Unsteady flow phenomena were numerically simulated and
experimentally investigated in an automotive diesel
injection equipment with a distributor-type pump, at several
loads and pump speeds, over a wide range of steady and
transient operating conditions.

The recently upgraded NAIS computational code for
simulating diesel injection-system dynamics under control
was further enhanced and applied for theoretical analysis.
The program, with a modular structure given by a library
containing a variety of system component models, is based on
a robust and efficient second-order accurate numerical
algorithm of the implicit type. This resolves all of the system
component equations at the same time, ensuring an essential
fluid-dynamic coupling between them. The code takes the
dependence of fuel properties on pressure and temperature,
friction and minor losses, as well as the nozzle-hole flow-
coefficient unsteadiness, into account. Besides, it is capable
of simulating cavitation occurrence and propagation.

A Hartridge 2500 test-bench, supplied with standard oil
ISO 4113, was used for experimental analysis of part-load
stationary operations at three pump speeds, one of which
being very low (250 rpm) and also for simulation and
analysis of transient control-phase working conditions. These
were actuated by giving a programmed rapid movement to
the pump control lever from its idle position to the full-load
position, at two constant pump angular speeds (1000 rpm
and 2000 rpm).

Pressure time-histories were measured and computed in
the pumping chamber and at the extremities of the delivery
pipe. Injector needle-lift and fuel injection-rate distributions
versus the pump-shaft crank angle were also analyzed in the
stationary operating conditions.

The consistency of theoretical and experimental results
substantiated the validity of the NAIS program at the engine
part-loads. The code presented good predictive capabilities
even for idling engine operation at the lowest pump speed,
where the nozzle-hole discharge flow-coefficient was shown
to be critical for the fuel injection-rate simulation.

The numerical system-model application to predict the
test actuated transient-load phases confirmed the NAIS code
capability of accurately predicting more critical dynamics
conditions requiring a cycle-by-cycle simulation of intensely
cavitating pipe-flow, mainly at low loads, after the nozzle
closure. A virtually negligible residual void-fraction effect
was apparent in subsequent cycles during the pump compres-
ssion stroke. This and the small sensitivity of the pipe residual
pressure to significant load variations, allowed by the reflux
valve, were the main reason for the injection system to adapt
to steady operation in a few cycles, soon after the effective
pump-plunger stroke had come into its full-load position.

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