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Instantaneous mass flowrate measurements through fuel injection nozzles

B Ünsal*, D Trimis, and F Durst

LSTM-Erlangen (Institute of Fluid Mechanics), Friedrich-Alexander Universität Erlangen-Nürnberg, Erlangen, Germany

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Abstract: A method of measuring the instantaneous mass flowrates of fuels passing through fast operating injection nozzles is described. The flowrate information is deduced from instantaneous pressure gradient measurements. Through a Fourier transformation, the Fourier coefficients of the pressure gradient are determined, which are used for the calculation of the transient mass flowrate of fuel through injection nozzles, utilizing a special reconstruction algorithm. The theoretical background of the measurement method is provided and the development work to yield a practically applicable measuring system is summarized. The latter is used for verification experiments, providing time-resolved flowrates for a magnetically driven injection valve operating at a pressure of approximately 6 bar. Online measurement of the injection mass flowrates can be recorded with this measuring system and information can be obtained for individual injection events. Cycle-to-cycle variations can be detected in this way.

Keywords: instantaneous mass flowrate measurements, fuels, injection nozzles

1 INTRODUCTION

There is currently intensive research and development work to advance automotive cars with respect to fuel consumption and emission of environmentally dangerous exhaust gases. Research efforts are concentrated on the advancement of car engines and here the fuel injection processes of internal combustion engines plays a vital role in obtaining high engine efficiencies and low emissions. Among other aspects of the combustion process, the control of the fuel injection and the formation of the fuel spray attracts great attention. Advances in this area can only be expected if a deeper understanding of the entire fuel injection process can be achieved. This requires more detailed information on the injection delay and on the instantaneous mass flowrate of fuel injected into each cylinder. This research is hampered since there is currently no technique available that can measure the instantaneous mass flowrate

of each individual injection event in a fuel supply line.

One of the oldest measurement methods for characterizing the performance of fuel injection systems, introduced by Bosch [1] in 1966, was a volumetric-flow-metering technique. It is based on the linear relationship between the pressure increase and the flowrate during fuel injection into a long tube connected to the exit of an injection nozzle. This technique is still frequently used for the quantification of injector characteristics but it is often pointed out that the exit of the spray is influenced in this way.

Several other injection-flow-metering techniques have been introduced by different researchers for the characterization of injector performance [2]. However, all available techniques are only able to measure the flowrate of the injected fuel after it has exited the nozzle. Mostly the momentum of the injected fuel is used for measurements and most techniques operate similarly to the above-mentioned Bosch method. Recently, a similar technique was presented by Schmidt *et al.* [3], based on the measurement of a piston displacement, with a displacement sensor placed at the outlet of an injection nozzle. With the associated electronic control and data-processing

*Corresponding author: LSTM-Erlangen (Institute of Fluid Mechanics), Friedrich-Alexander Universität Erlangen-Nürnberg, Cauerstrasse 4, D-91058 Erlangen, Germany. email: buensal@lstm.uni-erlangen.de

units, it was possible to calculate the injected fuel volume of each single injection and also the corresponding flowrate as functions of time.

In 1996, Durst *et al.* [4] presented an instantaneous volumetric-flowrate-metering technique, which was theoretically based on the exact analytical solution of the differential equation for a fully developed pulsating laminar pipe flow derived by Lambossy [5] in 1952. With this technique (see also references [6] and [7]) the instantaneous flowrate measurements were obtained through phase-resolved time series of the centre-line velocity of a fully developed pulsating pipe flow. The measurements were taken upstream of the injection nozzle, via laser Doppler anemometry (LDA), and the instantaneous injection flowrate was deduced by utilization of the analytical solution of Lambossy for the reconstruction of the phase-resolved flowrate. Since the LDA data rate was dependent on the available tracer particles in the fluid, only phase-averaged measurements, using LDA records over many cycles, could be performed, and this applied the assumption that cycle-to-cycle variations between various injection events were negligible. However, in the present authors' experience, the LDA-type transient flow meters require further studies to resolve existing problems. It may be that, during application to real engines, the optimum measuring volume size cannot be maintained during measurements. Further work is also needed to meet laser power requirements, depending on the working fluid and availability of tracer particles and mechanical stability. The effects of mechanical vibrations on the measuring volume size and measuring location need to be studied and methods developed to ease optical alignment. There is also a need for high-precision lasers and optical components at low cost. Additionally, the inherently stochastic nature of the LDA measurement events does not allow a single injection event to be resolved sufficiently under all operating conditions. Phase averaging over many periods has to be applied and cycle-to-cycle variations in the injection cannot be detected.

The objective of the present paper is to introduce a novel technique and a corresponding device for *instantaneous mass flowrate metering* of fuel injection systems for the entire range of injection pressures and opening and closing times of currently available fuel injectors. The proposed measurement technique and the associated instrument can be used in an engine environment and can be employed upstream of the injection nozzle so as not to disturb the outgoing spray. The theoretical background of the proposed measurement technique is based on the modified exact analytical solution of Lambossy

[5] and Uchida [8] for fully developed laminar pulsating pipe flows. The theory employed in the present paper was described by Ray *et al.* [9] and Ünsal *et al.* [10]. In the modified analytical solution, a new dimensionless approach was introduced, which brings certain advantages for the calculation of the instantaneous flowrate. The instantaneous *mass flowrate* is reconstructed on the basis of one measured instantaneous flow property, namely the pressure gradient in the present work. For calculations of the instantaneous mass flowrate, Fourier amplitudes from a lookup table are employed. Additionally, the kinematic viscosity of the fluid can be derived from the measurements.

The analytical solution and the calculation procedure for the flowrate from a measured time variation in the pressure gradient over one injection event are given in section 2. For verification of the measurement technique, an experimental test rig was set up, which enables pressure gradient-based and LDA centre-line velocity measurement-based flowrate reconstruction simultaneously at an injection pressure of 6 bar. Details of the test rig construction and experimental results together with comparisons of flowrates calculated from measured pressure gradients and center-line velocity time series are given in section 3. Finally, conclusions of this research and development effort are provided in section 4, together with a brief outlook for further studies.

2 THEORETICAL BACKGROUND OF MEASUREMENT TECHNIQUE

Today, there is an extensive theoretical background available on the measurement of pulsating flows by techniques that aim for instantaneous flowrate measurements. Looking at all information available, Lambossy [5] was the first to introduce an exact analytical solution for fully developed periodic pulsating laminar pipe flow driven by a time-varying prescribed pressure gradient. A more convenient and also a more general form of the solution was later presented by Uchida [8]. Recently the present authors [9, 10] presented a modified version of the solution which allows calculations of pulsating flow properties and shows that these properties are inter-related, permitting the pulsating mass flowrate to be computed from any given flow parameter. The solution [9, 10] depends on the superposition principle of the Fourier series. It results from the governing equation, which is a linear homogeneous ordinary differential equation that can be solved for the case

of a single sinusoidal pulsating flow. Hence, for arbitrary pulsations, which can be expressed by Fourier series, pulsations of arbitrary shapes can be solved as a summation of individual sinusoidal solutions. In the next section, this analytical solution [9, 10] will be summarized with the aim of the present work in mind, namely to develop a method for instantaneous injection mass flowrate measurements.

2.1 Arbitrary pulsating pipe flows

The governing momentum equation, in cylindrical coordinates, for fully developed axisymmetric pulsating laminar pipe flow can be expressed as

$$\rho \frac{\partial U}{\partial t} = -\frac{\partial P}{\partial x} + \frac{\mu}{r} \frac{\partial}{\partial r} \left(r \frac{\partial U}{\partial r} \right) \quad (1)$$

Any time-dependent axial pressure gradient for the attempted analytical solution is generally written in the form of a Fourier series

$$\begin{aligned} -\left(\frac{\partial P}{\partial x}\right) &= \hat{P}_{\text{ave}} + \sum_{n=1}^{\infty} \hat{P}_{\text{en}} \exp(2\pi n f t i) \\ &= \hat{P}_{\text{ave}} \left[1 + \sum_{n=1}^{\infty} \frac{\hat{P}_{\text{en}}^*}{\hat{P}_{\text{ave}}} \exp(2\pi n F t i) \right] \end{aligned} \quad (2)$$

where $\hat{P}_{\text{en}}^* = \hat{P}_{\text{en}} / \hat{P}_{\text{ave}}$ is the dimensionless amplitude of the pressure gradient pulsation and f is the overall frequency of the pulsation. $F (= R^2 f / \nu)$ and $\tau (= \nu t / R^2)$ are the dimensionless frequency and time respectively. $\hat{P}_{\text{ave}}$ is the time-averaged axial pressure gradient and $\hat{P}_{\text{en}}^*$ is the complex form of the pressure gradient amplitude, given in the form

$$\hat{P}_{\text{en}}^* = \hat{P}_{\text{cn}}^* - i \hat{P}_{\text{sn}}^* \quad (3)$$

The equality expressed in equation (2) for the last term of the equation holds only for the real part of the right-hand side. For a given pressure gradient in equation (2), the analytical solution for the velocity distribution as a function of radius and time can be obtained as [9]

$$\begin{aligned} U^* &= \frac{U}{U_{\text{m}}} = 2(1 - r^{*2}) - \sum_{n=1}^{\infty} \frac{4\hat{P}_{\text{en}}^*}{\pi n F} \left\{ 1 - \frac{J_0[(2\pi F)^{1/2} i^{3/2} r^*]}{J_0[(2\pi F)^{1/2} i^{3/2}]} \right\} \\ &\quad \times \exp(2\pi n F t i) \end{aligned} \quad (4)$$

where U^* is the dimensionless axial velocity, U_{m} is the average velocity of the fluid through the pipe, corresponding to the steady part of the pressure gradient, $U_{\text{m}} = \hat{P}_{\text{ave}} R^2 / 8\nu$, and $r^* (= r/R)$ is the dimensionless radius.

The mass flowrate through the cross-section of the pipe, as a function of time, can be obtained by integrating the velocity solution, which will give the mass

flowrate solution as

$$\begin{aligned} \dot{m}^* &= \frac{\dot{m}}{\dot{m}_{\text{m}}} = 1 - \sum_{n=1}^{\infty} \frac{4\hat{P}_{\text{en}}^* i}{\pi n F} \left\{ 1 + \frac{2i^{1/2} J_1[(2\pi n F)^{1/2} i^{3/2}]}{(2\pi n F)^{1/2} J_0[(2\pi n F)^{1/2} i^{3/2}]} \right\} \\ &\quad \times \exp(2\pi n F t i) \end{aligned} \quad (5)$$

where $\dot{m}^*$ is the dimensionless mass flowrate and $\dot{m}_{\text{m}}$ is the steady part (mean part) of the mass flowrate through the pipe due to U_{m} and can be expressed as $\dot{m}_{\text{m}} = \rho \pi R^2 U_{\text{m}}$. Equation (5) may also be written in the more convenient form

$$\dot{m}^* = 1 + \sum_{n=1}^{\infty} \dot{m}_{\text{os},n}^* \quad (6)$$

where, $\dot{m}_{\text{os},n}^*$ is the n th mode of the oscillating part of the mass flowrate pulsation and can be expressed as

$$\begin{aligned} \dot{m}_{\text{os},n}^* &= \frac{4\hat{P}_{\text{en}}^* i}{\pi n F} \left\{ 1 + \frac{2i^{1/2} J_1[(2\pi n F)^{1/2} i^{3/2}]}{(2\pi n F)^{1/2} J_0[(2\pi n F)^{1/2} i^{3/2}]} \right\} \\ &\quad \times \exp(2\pi n F t i) \end{aligned} \quad (7)$$

It may be noted that $\dot{m}_{\text{os},n}^*$ is directly proportional to the n th amplitude of complex pressure gradient amplitude $\hat{P}_{\text{en}}^*$. Hence each mode of mean oscillation has its oscillation amplitude and its corresponding pressure gradient. Therefore, the ratio $\dot{m}_{\text{os},n}^* / \hat{P}_{\text{en}}^*$ depends only on the dimensionless n th frequency of pulsation, nF . It may also be identified from the above equation that a complex variable ψ_n may be defined, which is solely a function of the dimensionless n th frequency nF . Therefore, ψ_n can be written as

$$\psi_n(nF) = -\frac{4i}{\pi n F} \left\{ 1 + \frac{2i^{1/2} J_1[(2\pi n F)^{1/2} i^{3/2}]}{(2\pi n F)^{1/2} J_0[(2\pi n F)^{1/2} i^{3/2}]} \right\} \quad (8)$$

Splitting the complex variable ψ_n into real and imaginary parts gives

$$\psi_n(nF) = \Re(\psi_n) + i \Im(\psi_n) \quad (9)$$

where $\Re(\psi_n)$ is the real part and $\Im(\psi_n)$ is the imaginary part of $\psi_n(nF)$. The complete expression for the mass flowrate through the pipe for a pressure driven flow may therefore be written from equations (8) and (9) as

$$\dot{m}^* = 1 + \sum_{n=1}^{\infty} \hat{P}_{\text{en}}^* |\psi_n| \exp \left(\left\{ 2\pi n F \tau - \tan^{-1} \left[\frac{\Re(\psi_n)}{\Im(\psi_n)} \right] \right\} i \right) \quad (10)$$

Equations (10) and (2) can be rewritten in the forms

$$\dot{m}^* = 1 + \sum_{n=1}^{\infty} \dot{m}_{A,n}^* \exp[(2\pi Nft + \theta_{\dot{P}_{en}}^* - \Delta\theta_{m,n})i] \quad (11)$$

$$\hat{P}^* = -\frac{dP}{dx} \frac{1}{\hat{P}_{ave}} = 1 + \sum_{n=1}^{\infty} \hat{P}_{A,n}^* \exp[(2\pi Nft + \theta_{\hat{P}_{en}}^*)i] \quad (12)$$

where $\dot{m}_{A,n}^*$ and $\hat{P}_{A,n}^*$ are the real normalized amplitudes, $\Delta\theta_{m,n}$ is the phase lag of the n th wave of the mass flowrate oscillation, and $\theta_{\hat{P}_{en}}^*$ is the phase of the complex amplitude $\hat{P}_{en}^*$; all are defined as

$$\dot{m}_{A,n}^* = \hat{P}_{A,n}^* |\psi_n| \quad (13)$$

$$\hat{P}_{A,n}^* = |\hat{P}_{e,n}^*| \quad (14)$$

$$\theta_{\hat{P}_{en}}^* = \tan^{-1} \left[\frac{\Re(\hat{P}_{en}^*)}{\Im(\hat{P}_{en}^*)} \right] \quad (15)$$

$$\Delta\theta_{m,n} = \tan^{-1} \left[\frac{\Re(\psi_n)}{\Im(\psi_n)} \right] \quad (16)$$

By comparing equation (11) with equation (12), it can be noted that both the amplitude ratio and the phase difference between mass flow and pressure gradient harmonics are functions only of the dimensionless n th frequency, i.e. depend only on nF . Since for a prescribed pressure gradient all the amplitudes $\hat{P}_{A,n}$ are known, equation (13) can be used to obtain the values of the mass flowrate amplitudes $\dot{m}_{A,n}^*$, since $|\psi_n|$ depends only on the respective dimensionless frequency nF (see equation (8) for further clarification). Similarly, the required phase difference between the mass flowrate and the pressure gradient oscillations for each harmonic can be obtained from equation (16). Hence the time-resolved normalized mass flowrate can be calculated if the corresponding pressure gradient is known.

In order to obtain the dimensional mass flowrate, the value of the steady part (mean) $\dot{m}_M$ of the mass flowrate would be required. This may be obtained from knowledge of the cycle-averaged pressure gradient $\hat{P}_{ave}$ given by

$$\dot{m}_M = \frac{\pi R^4 \hat{P}_{ave}}{8\nu} \quad (17)$$

Note that, in equation (2) the pressure gradient was normalized by the density. The units of $\hat{P}_{ave}$ in equation (2) are metres per squared second and in equation (17) are pascals per metre.

Figure 1 shows the variations in $\dot{m}_A^*/\hat{P}_A^* = |\psi|$ and $\Delta\theta$ with F . It can be seen that, for low pulsation frequencies, the amplitude ratio is unity and the phase difference is zero, corresponding to the steady state solution.

Fig. 1 Variation in $\dot{m}_A^*/\hat{P}_A^*$ and phase difference $\Delta\theta$ with pulsation frequency F

In summary it was shown that, when the known pressure gradient for an arbitrary, fully developed pulsating laminar pipe flow is expressed in terms of Fourier series, the amplitudes can be normalized in such a manner that the resulting mass flowrate can be calculated as a Fourier summation through defined dimensionless amplitude ratios and phase differences for each harmonic component. It is important to note that, in a similar way, the pressure gradient from a known mass flowrate variation can also be calculated. A similar analysis can also be made between any variable of the flow such as point velocities and the wall shear stress (see reference [10] for details). From the presented analysis, it becomes obvious that several individual time-resolved flow parameters will allow the entire transient flow field to be calculated in a very straightforward manner.

3 EXPERIMENTAL VERIFICATIONS

For the experimental verification of the presented mass flow-metering technique, an experimental test rig for pressure gradient and LDA centre-line velocity measurements was set up. The latter was employed in a manner similar to the experimental unit suggested by Durst *et al.* [4]. The pressure gradient test unit, also set up for the present experiments, is new and is the first of its kind for injection flow-rate measurements. For the present verification experiments, it was operated in series with the LDA instantaneous flowrate meter, yielding the same instantaneous flowrate through both measuring units. Hence both yielded the same flowrate as a function of time under the assumption of incompressibility of the applied liquids and pipe materials.

The experiments described in the following sections were conducted with diesel oil and water and both liquids produced excellent results. In the case of diesel oil, the LDA was not able to deliver sufficiently high data rates for all comparison purposes, owing to difficulties in introducing tracer particles into the oil medium. For this reason, reconstructed flowrates, measured by both methods, were only compared for the measurements with water, and drops of milk were added for the formation of tracer particles.

3.1 Test rig and measurement procedure

A photograph of the test rig is shown in Fig. 2. It includes a liquid container, pressurized by compressed air at 6 bar, connected to the measurement pipe sections. In the first part of the measurement pipe section (3 mm inner diameter), two pressure sensors were placed at a distance of 150 mm apart. The distance from the pipe inlet to the first sensor was 250 mm, providing a sufficient length for the flow to reach its fully developed laminar stage. The second measurement pipe section (also 3 mm inner diameter, but 300 mm long) was connected in series to the first and was made of glass in order to permit optical access for the LDA axial velocity measurements. At the exit of the second pipe section a low-pressure, magnetically driven gasoline injection valve was installed. The injected liquid was collected in a container for the purpose of gravimetric-based measurements of the mean flowrate. The gravimetrically obtained mean flowrates and corresponding mean velocities were used to assure the correct adjustment of the LDA optics. Comparisons were also made for mean flowrates measured by the pressure gradient instrument and LDA centre-line velocity-measuring unit.

For the pressure gradient measurements, two piezo-electric charge output-type pressure sensors from PCB were used, having a full scale of 250 bar and a natural resonance frequency of 250 kHz. The sensors were acceleration and temperature compensated and were specially selected for the present investigation. Both sensors were connected to laboratory-type charge amplifiers also supplied by PCB. The discharge time constant of the amplifier was set to its maximum value, which was significantly higher than the pulsation period. The pressure signal outputs from the charge amplifier were connected to a personal computer (PC) possessing a four-channel data acquisition (DAQ) card (National Instruments 4474) for transferring the data to a computer. The DAQ card had a 22 bit resolution and the maximum sampling rate for each channel was 100 kHz. In order to trigger the measurements for each individual injection period, a reset signal from the valve driver was acquired simultaneously with the pressure signals. The data acquisition and online and offline data-processing codes were written in LabView™ graphical programming language. The program created permitted the acquisition and processing of the data to yield the measurements described below.

The LDA system shown in Fig. 2 was driven by a 15 mW helium–neon laser. The laser beam passed through the collimator lenses of the sending optics for adjusting the beam diameter to obtain the appropriate control volume dimensions ($100\text{ }\mu\text{m} \times 300\text{ }\mu\text{m}$). After the collimator lenses, a standard TSI sending LDA optic was used, including a beam splitter, a Bragg cell, and a correcting prism. The Bragg cell driver was set to an operating shift frequency of 1 MHz. The parallel beams, with almost equal intensities, were focused on the pipe centre-

Fig. 2 Photograph of the experimental set-up

line with an 80 mm focusing lens installed on a two-dimensional traversing unit for easy focusing of both beams to the pipe centre-line. The scattered light from the measuring volume was collected in the forward direction and focused on to an avalanche photodiode (APD) with the help of a light-collecting lens. The signal output from the APD was amplified and fed into a TSI 1980 counter processor, which was connected to the PC with a Dostek interface card. For the phase-resolved LDA measurements, the trigger signal received from the valve driver gave a reset signal for the start of each measuring period. The valve driver also gave 1000 equidistant pulses per one measuring period to the interface card, in order to assign the phase of every LDA event. Phase averaging was applied over many periods in order to yield the LDA-measured flowrate as a function of phase. Hence, in order to have sufficient data to obtain the phase-averaged centre-line velocity in each of the 1000 phases of a period, 100 000 velocity data over several hundred periods were sampled. The phase-resolved volumetric flowrates were reconstructed from the stored and phase-averaged velocity data offline.

3.2 Results

An example result from the measurements with diesel oil is shown in Fig. 3 for a 0.5 s pressure record (Fig. 3(a)) with 5 Hz injection frequency and 10 ms injection duration. The figure also shows the pressure gradient obtained (Fig 3(b)) through the two sensors and the calculated mass flowrate (Fig. 3(c)), which was obtained by fast Fourier transformation of the pressure gradient according to equation (12) and calculation of the mass flowrate according to equation (11) together with equations (13) to (17).

It is important to note that the mass flowrate calculation procedure can be carried out for an arbitrary time series of pressure gradient if it is significantly longer than the actual periodic flow frequency, as in the case of Fig. 3, since the long arbitrary time interval considered can be assumed to be periodically repeating in time.

Accurate pressure gradient measurements, for the present purpose, require pressure sensors with a very large amplitude resolution. The pressure jumps observed during the valve opening and closing were of the order of 1 bar, as can be seen in Fig. 3(a). As shown in the pressure gradient waveform (Fig. 3(b)), the pressure gradient level during valve opening is of the order of 1000 Pa/m. However, after the valve opening and during fuel injection, the pressure gradient is of the order of 50 Pa/m, which cannot be

Fig. 3 An example pressure waveform, calculated pressure gradient, and mass flowrate with diesel oil for a 5 Hz injection frequency and 10 ms injection duration: (a) pressure waveforms of sensors 1 and 2; (b) obtained pressure gradient; and (c) calculated mass flowrate

recognized in the figure owing to the large-scale pressure jumps during valve opening and closing. Therefore, in order to be able to calculate the mass flow-rate accurately from the pressure gradient measurement, it is necessary to have a resolution of some pascals considering a maximum range for the pressure detection of nearly 1 bar. Hence a resolution of approximately 1 to 100 000 in the pressure amplitude is needed for the kind of measurement

described in this paper. Such a high resolution was only possible by employing the above-mentioned piezoelectric pressure sensors and the high-resolution DAQ card.

The measurements with diesel oil were carried out up to an injection frequency of 25 Hz and for various injection durations. The results were not compared with LDA owing to difficulties in introducing tracer particles into the oil medium. For this reason, average flowrates of the results were compared with gravimetric measurements. These comparisons yielded an exact match within 2 per cent but are not shown here.

The resulting mass flowrate plots for diesel oil are shown in Fig. 4 for injection durations of 10, 20, and 30 per cent of the injection period. The plots show only a single injection period for 5, 10, and 25 Hz injection frequencies. It can be seen from Fig. 4 that the time dependence of mass flowrate variations is similar for different injection frequencies. All measurements show the initial increase in the mass flowrate, resulting in an overshoot a short time after the valve opening. Thereafter, the flowrate reached a constant value if the valve stayed open for a long enough time interval. When the valve had closed, the flowrate suddenly decreased and it undershot the zero flowrate level. This undershoot of the flowrate is then followed by some oscillations. These oscillations are damped by some time constant given by the entire flow system. The valve opening time, which turned out to be independent of the injection frequency and injection duration, is about 3 ms, which can readily be seen from Fig. 5. The flowrate oscillations that are produced by the valve closing have a common time constant of about 4 ms, independent of the injection parameters. The oscillations actually depended on the whole injection system including the valve, pipe length, container volume, etc.

The particular features of the present flow-metering technique permit cycle-to-cycle variations in the injected mass flowrate to be detected and precisely quantified. This is illustrated in Fig. 6, showing four measured injection periods with one of the injection period providing artificially a higher mass flowrate. Hence the present instantaneous mass-flowrate-measuring system permits injection nozzle research to be based on reliable measurements.

For comparison of the transient flowrate measurements obtained with the pressure-gradient-based measuring system and the LDA-based device, the working medium was changed to water and a very small amount of milk was mixed into the water in order to generate tracer particles for the LDA measurements. Even if sufficient particles were present in

Fig. 4 Measured mass flowrates with diesel oil for 10, 20, and 30 per cent injection duration within an injection period. Plots show only a single injection period for injection frequencies of 5, 10, and 25 Hz

the water, the LDA-based measuring system required several hundred periods to complete one set of phase-averaged centre-line velocity measurements. Figure 7 shows an example velocity time record obtained during a complete LDA velocity measurement series. The vertical lines represent the time intervals where no velocity information was present. Depending on the data rate, a complete LDA velocity measurement series for one valve setting took from around several tens of seconds to several minutes.

Some of the results from the experiments with water are given in Fig. 8, showing the volumetric flowrate as a function of time for single injection

Fig. 5 Mass flowrate time variations for 5, 10, and 25 Hz injection frequencies: (a) the complete time record; (b) the first 20 ms of the record to show the dependence of the mass flowrate behaviour on the injection frequency when the valve opens

Fig. 6 Sequence of mass flowrate signals showing cycle-to-cycle variations of the injector-nozzle performance

periods of 2, 5, and 10 Hz and injection durations of 10 and 30 per cent of the injection period. Comparing the resultant flowrates, it can be concluded that the flowrate measurements obtained through

the measured pressure gradient and using the analytical solution presented in section 2 offer high accuracy and a significantly simplified measurement procedure with fast data processing, much faster than that shown for the LDA-based system.

4 CONCLUSION, FINAL REMARKS, AND OUTLOOK

The present investigations indicate that time-resolved instantaneous mass flowrates of fast-operating injection fuel nozzles can be accurately measured at an upstream position of an injection nozzle. A pressure-gradient-based measurement technique was theoretically derived and experimentally realized. This technique can be applied for laboratory investigations but is also applicable in automotive engine test stands for the detailed characterization of

Fig. 7 Example centre-line velocity time records, as measured by LDA, showing three measurement instances: (a) just after the start of acquisition; (b) acquisition not yet completed; (c) resulting velocity waveform after completed acquisition

Fig. 8 Comparison of volume flowrate time series of single injection periods obtained by the pressure-based technique and phase-averaged injection period with the LDA-based technique for injection frequencies of 2, 5, 5, and 10 Hz

instantaneous fuel injection events. There is at present no comparable measurement technique available for this purpose. The following conclusions can be drawn as a result of the present investigations.

1. The exact analytical solution for a fully developed laminar pulsating pipe flow with an arbitrary pulse shape can be utilized for instantaneous mass flowrate measurements. The solution provided by Ray *et al.* [9] allows normalized mass flowrate amplitudes to be calculated directly from the measured pulsation frequency and recorded pressure gradient, since normalized amplitude

ratios and phase differences of a single harmonic are a unique known function of the dimensionless frequency F

2. Time-resolved instantaneous mass flowrate measurements can be realized in practice by simple pressure gradient measurements over a pipe. Pressure data processing and mass flowrate reconstruction can be performed very rapidly by employing high-resolution DAQ devices and the reconstruction method based on the analytical solution summarized in this paper.
3. The technique allows non-intrusive time-resolved instantaneous mass flowrate measurements of fuel injection events. In contrast with other available measurement techniques, such as those summarized in reference [2], the present technique does not disturb the injection process.
4. The outstanding features of the presented technique were experimentally verified with diesel oil and water as working fluids. Comparisons with an LDA-based transient volume flowrate meter were performed for the case of water as the working fluid. The comparison of the phase-resolved volumetric flowrate obtained from the pressure-gradient-based mass flow meter and the LDA centre-line velocity-based volume flowrate meter demonstrated the excellent accuracy of the present method.
5. The present authors are already proceeding with further development work towards the application of the present method to high-pressure injection systems such as direct gasoline injection (about 250 bar) and diesel (up to 2000 bars) injection systems, employing piezo-driven fuel injectors with fast opening and closing times down to 0.1 ms.

The above conclusions readily suggest that the proposed mass flowrate measuring system is well suited to yielding valuable information on the performance of the injection process through nozzles.

REFERENCES

- 1 **Bosch, W.** The fuel rate indicator: a new measuring instrument for display of the characteristics of individual injection. SAE paper, 660749, 1996.
- 2 **Marčič, M.** Measuring method for diesel multihole injection nozzles. *Sensors Actuators*, 2003, **107**, 152–158.
- 3 **Schmidt, F., Eynard, P., Maurin, P., and Gauthier, C.** Device for instantaneous *ad hoc* analysis of an injection flow provided by an injection system used in a heat engine. US Pat. 6,755,076 B1, 2004.
- 4 **Durst, F., Ismailov, M., and Trimis, D.** Measurement

- of instantaneous flow rates in periodically operating injection systems. *Exp. in Fluids*, 1996, **20**, 178–188.
- 5 **Lambossy, P.** Oscillations forcees d'un liquide incompressible et visqueux dans un tube rigide et horizontal. Calcul de la force frottement. *Helv. Physica Acta*, 1952, **25**, 371–386.
- 6 **Bopp, S., Durst, F., Teufel, M., and Weber, H.** Volumetric flow rate measurements in oscillating pipe flows with a laser Doppler sensor. *Measmt Sci. Technol.*, 1990, **1**, 917–923.
- 7 **Teufel, M., Trimis, D., Lohmüller, A., Takeda, Y., and Durst, F.** Determination of velocity profiles in oscillating pipe-flows by using laser Doppler velocimetry and ultrasonic measuring devices. *Flow Measmt Instrum.*, 1992, **3**(2), 95–101.
- 8 **Uchida, S.** The pulsating viscous flow superposed on the steady laminar motion of incompressible fluid in a circular pipe. *J. Appl. Math. Physics (Z. Angew. Math. Physik)*, 1956, **7**, 403–422.
- 9 **Ray, S., Ünsal, B., Durst, F., Ertunc, Ö., and Saleh, O.** Mass flow rate controlled fully developed laminar pulsating pipe flows. *Trans. ASME, J. Fluids Engng*, 2005, **127**(3), 405–418.
- 10 **Ünsal, B., Ray, S., and Durst, F.** Pulsating laminar pipe flows with sinusoidal mass flux variations. *J. Fluid Dynamics Res.*, 2005, **35**, 317–333.

APPENDIX

Notation

f	frequency (Hz)
F	dimensionless frequency = $R^2 f / \nu$
$\dot{m}$	mass flowrate (kg/s)
$\dot{m}^*$	dimensionless mass flowrate = $\dot{m} / \dot{m}_M$

$\dot{m}_A^*$	dimensionless mass flowrate amplitude = $\dot{m}_A / \dot{m}_M$
n	integer number
P	pressure (Pa)
P_A^*	dimensionless pressure amplitude = $(dP/dx)_A / (dP/dx)_M = \hat{P}_A^*$
r	radial location (m)
r^*	dimensionless radial location = r/R
R	pipe radius (m)
t	time (s)
U	velocity (m/s)
U^*	dimensionless velocity
x	axial coordinate (m)
θ	phase (rad)
ν	kinematic viscosity
ρ	density (kg/m ³)
τ	dimensionless time = $\nu t / R^2$

Superscripts

*	dimensionless quantity
^	pressure gradient

Subscripts

ave	time-averaged mean
A	amplitude
c	cosine term of the Fourier series
M	time-averaged mean
n	n th term of the Fourier series
os	oscillating component
s	sine term of the Fourier series