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Modified multi-load method for nonlinear source characterisation

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Abstract

Linear frequency domain prediction codes are useful for calculation of low-frequency sound transmission in duct and pipe systems. To calculate insertion loss of mufflers or the level of radiated sound information about the acoustic source is needed. The source model used in the low-frequency plane wave range is the linear time invariant one-port model. The acoustic source data is usually obtained from experimental tests where multi-load methods and especially the two-load method are most commonly used. The exhaust pulsations of for example an IC-engine are of high level, and the engine is not a perfectly linear and time invariant source. It is therefore of interest to develop source models and experimental techniques that try to take this nonlinearity into account. In this paper a modified version of the two-load method to improve the characterisation of nonlinear acoustic one-port sources has been developed and tested. Simulation results as well as experimental data from various source configurations for a modified compressor and experimental data from 6-cylinder turbocharged truck diesel engine were used to validate the method. The influence of parameters controlling the linearity of the system was investigated. The time-variance of the source model was varied and the accuracy of source characterisation results using the two-load method and the modified two-load method was evaluated.

1. Introduction

Noise and vibration are usually most efficiently reduced at the source. It is therefore important to be able to characterise noise sources by their source strength and also to know how they interact with their surroundings. The goal of source characterisation is to provide a complete and independent source model, which fully describes how the source interacts with the receiving system and does not depend on the properties of the receiver. An acoustic source model can be used for calculation of the acoustic field generated in duct systems coupled to fluid machines: e.g., pumps, fans, internal combustion engines, for source modifications, for evaluation of noise sources and for appropriate source and receiver design.

The types of source models can first be divided into linear and nonlinear models. The linear models can further be subdivided into time-invariant models and time-varying models which means that the parameters (or boundary conditions) in the governing equations are either independent of or dependent on time.

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The methods used for experimental source characterisation can be classified as direct methods (with external sound source) and indirect methods or multi-load methods (without external sound source).

An overview on previous studies and the measurement methods available for determination of the source data can be found in Refs. [1,2].

Regarding linear and time-invariant one- and two-port sources, the measurement techniques for obtaining source data are quite well developed but regarding time-varying and nonlinear source models there is still a lack of complete models, e.g., for the intake and exhaust noise of IC-engines [3]. In the conclusions of Ref. [3] it is stated that one of the areas where considerable research input is needed is the frequency domain characterisation of the engine exhaust source. Due to the simplicity of existing characterisation models the fluid machines connected to a duct system are quite commonly modelled as linear and time-invariant sources. In Ref. [4] a method to characterise linear time-variant sources was proposed and tested by Bodén. Jang and Ih [5] have proposed a method for including nonlinearity in direct source characterisation techniques without showing any experimental results.

Many fluid machines such as compressors and IC-engines are high-level acoustic sources where clearly nonlinear processes take place, for example, combustion, critical flow through the valve openings and large temperature fluctuations. As these processes, generated by the source, may violate the linearity assumption, nonlinear effects included in the models could improve the source characterisation results.

In this paper the source data from the experimental and simulated data is determined using indirect characterisation methods and the focus will be on characterisation of fluid borne sources with nonlinear behaviour, using a refined version of the well-known two-load method in which nonlinear effects are included. The new method is developed and used to characterise the source data of various acoustic sources. The source characterisation results are then evaluated and the results are compared to the ‘‘classical’’ two-load method.

Knowledge about systems linearity and time-variance is often needed to be able to make decisions when choosing a method for source modelling and characterisation. Using a one-cylinder ‘‘cold’’ engine simulation model [6] the linearity and time-variance of this simplified IC-engine as acoustic one-port source, are investigated in this paper in order to study the influence of geometric and dynamic parameters on the system behaviour. The developed methods have also been tested on the experimental data.

2. Source models

In developing source models the aim is to produce the most simple model which is able to provide acceptable results. The models can be classified as linear time-invariant, linear time-varying, hybrid and nonlinear models, in order of increasing complexity.

The source is described by the physical quantities via which it interacts with the outside world. In the simplest case where an infinite space around the source can be assumed, the source can be characterised by its radiated acoustic power. For low frequencies in-duct more complex descriptions such as multi-port models are often needed [1].

2.1. Linear time-invariant source model

If only plane waves are considered in the duct system the simplest model that can be used to describe the source is the linear time-invariant frequency domain one-port model. In the frequency domain an acoustic one-port can be completely described by two complex parameters: the source strength (P_+^s) and the source reflection coefficient (R_s) (or alternatively the source impedance). The behaviour of the one-port (see Fig. 1) can in the frequency domain, be described by [1]:

$$P_+ = R_s P_- + P_+^S, \tag{1}$$

where (P_-, P_+) are travelling acoustic pressure amplitudes, (R_S) is the source reflection coefficient at cross section, where $x = 0$ (see Fig. 1), and (P_+^S) is the source strength. The source strength (P_+^S) can be interpreted as the pressure generated by the source-side when the system is reflection free.

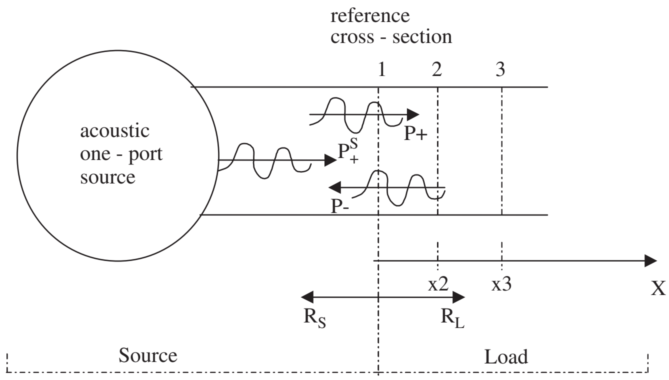

Fig. 1. An in-duct source modelled as an acoustic one-port.

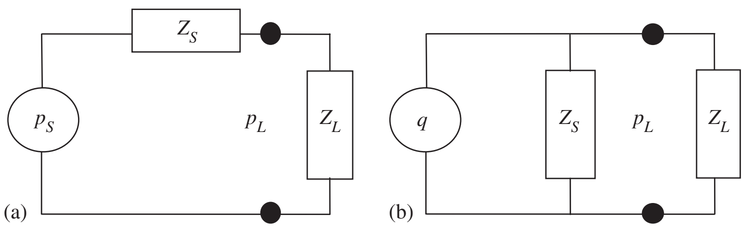

Fig. 2. Equivalent acoustic circuits for linear time invariant source. (a) Pressure source, (b) volume velocity source.

In the literature the source model for one-ports is often expressed in terms of source strength (P_S) and normalised source impedance (Z_S),

$$P = P_S - Z_0 Z_S Q, \tag{2}$$

where (P_S) is the source pressure, (P) and (Q) are acoustic pressure and volume velocity, respectively, and (Z_0) is the characteristic impedance of the fluid. The source impedance (Z_S) represents the acoustic impedance seen from the reference cross section towards the source.

Fig. 2 shows the equivalent acoustic circuit for a linear and time invariant source. In this figure (P) and (Z) denote the acoustic load data (the load pressure and the load impedance), while (P_S), (Q_S) and (Z_S) denote the source data respectively. Theoretically the two representations of the source shown in Fig. 2 are equivalent and it is possible to go from one representation to the other by using the relationship $Q_S = P_S/Z_S$. If there are errors in the experimental data or deviations from system linearity it can be expected that the error propagation is different for the two representations leading to different results when source data is extracted using over-determination. Other techniques for testing system linearity are discussed in Section 3.3. Extracting source data using both formulations and comparing the resulting source impedance is a possibility to see if experimental data are in agreement with linear time-invariant source model. It can also be expected that if the source is more a velocity source than a pressure source this model will give smaller errors than if a pressure source model is applied and vice versa.

Linear time-invariant source models are strictly only applicable in situations where the pressure-fluctuations are small. Several authors have however found the linear time-invariant model to give reasonable results for modelling the systems with relatively large pressure fluctuations, i.e., weakly nonlinear systems [7,8].

2.2. Nonlinear source model

To improve the described source characterisation methods, especially for applications where nonlinear effects are expected, such as for IC-engines, a technique was suggested by Jang and Ih [5]. The idea was to include nonlinear effects in the direct methods for source impedance determination. This method was suggested without showing any experimental results and without making it clear how the time domain volume velocity ($q(t)$) would be obtained from experiments.

In this paper the method has been modified for applications with indirect or multi-load methods. The time domain representation of the source model with nonlinear term is described by

$$\int z_S(\tau)q(t-\tau)d\tau + \int h_S(\tau)b(t-\tau)d\tau = p_S(t) - p(t), \tag{3}$$

where ($p(t)$) and ($q(t)$) denote the pressure and volume velocity at the source cross section, ($z_S(t)$) is the time domain representation of the source impedance, ($p_S(t)$) is the source strength, ($b(t)$) is the nonlinear input and ($h_S(t)$) is the source data coefficient for the nonlinear part. When applying this technique in Section 5 and 6 it has been assumed that $b(t) = q^3(t)$ which is the first higher order series expansion term obtained for the pressure drop over an orifice. It can be expected that the main type of nonlinearity for many applications will be caused by the flow through a constriction characterised by the pressure difference over the constriction being equal to $\Delta p(t) = (\rho_0/2S^2)q(t)|q(t)|$. It is shown in Ref. [9] that, under the assumption that ($q(t)$) follows a zero mean Gaussian distribution, the third-order polynomial least-squares approximation to $(\rho_0/2S^2)q(t)|q(t)|$ is

$$p(t) = \frac{\rho_0}{2S^2}\left[\left(\sigma_q\sqrt{2/\pi}\right)q(t) + \left(\frac{\sqrt{2/\pi}}{3\sigma_q}\right)q^3(t)\right], \tag{4}$$

where σ_q is the standard deviation of ($q(t)$). Taking the Fourier transform of Eq. (4) gives

$$P(f) = \frac{\rho_0}{2S^2}\left[\left(\sigma_q\sqrt{2/\pi}\right)Q(f) + \left(\frac{\sqrt{2/\pi}}{3\sigma_q}\right)Q_3(f)\right], \tag{5}$$

where ($P(f)$), ($Q(f)$) and ($Q_3(f)$) are the Fourier transforms of ($p(t)$), ($q(t)$) and ($q^3(t)$). The original "square-law system with sign" $(\rho_0/2S^2)q(t)|q(t)|$ can therefore to the third order be replaced by a linear system in parallel with a cubic system.

In the frequency domain Equation (3) can be formulated as

$$P_S Z - PZ_S - H_S ZB = PZ, \tag{6}$$

where (H_S) and (B) are the Fourier transforms of ($h_S(t)$) and ($b(t)$). This equation has compared to Eq. (2) a third complex unknown (H_S), which means that now at least three acoustic loads will have to be used in order to solve the equation and to obtain the source data.

2.3. Prediction of sound pressure using source data

The purpose of determining acoustic source data is to be able to predict the sound pressure generated by the source for acoustic loads not used as part of the source characterisation. For a linear time-invariant source model the sound pressure (P) at the source cross section can be calculated if the impedance of the acoustic load is known. This follows directly from Eq. (2) if the relationship $Q = P/(Z_0 Z)$ is used,

$$P = P_S\frac{Z}{Z+Z_S}. \tag{7}$$

The corresponding expression for the nonlinear source model discussed in section 2.2 follows from Eq. (6),

$$P = P_S\frac{Z}{Z+Z_S} - H_S B\frac{Z}{Z+Z_S}. \tag{8}$$

The extra term in Eq. (8) compared to Eq. (7) does however depend on P since B is the Fourier transform of $q^3(t)$. One possible way to solve for P is by iteration. Using an initial guess for $q^3(t)$ P can be calculated from Eq. (8). A new estimate of $q^3(t)$ is then calculated using $Q = P/(Z_0 Z)$ and inverse Fourier transformation. These steps are then repeated until convergence is reached. It should be pointed out that this technique has not been used in the prediction results presented in Sections 5–7. In all of these cases both P and Z are available from simulations or measurements making the iteration procedure unnecessary.

3. Experimental source characterisation

A number of different methods exist for determining acoustic source data from experiments. An overview of the state of the art of experimental methods for determining the one-port source data for in-duct fluid-borne sound sources was described in the review papers [1,2]. Here we will concentrate on so-called indirect or multi-load methods.

3.1. The two-load method

If the source is time-invariant, the one-port source characteristics can be determined by using only two external loads. This method is known as the two-load method.

$$P_S Z - P Z_S = P Z. \tag{9}$$

Eq. (9) has got two complex unknowns, which means that it can be solved if we have at least two complex equations. If we use n acoustic loads we get

$$\begin{bmatrix} Z_1 & P_1 \\ Z_2 & P_2 \\ \vdots & \vdots \\ Z_n & P_n \end{bmatrix} \begin{pmatrix} P_S \\ Z_S \end{pmatrix} = \begin{pmatrix} P_1 \cdot Z_1 \\ P_2 \cdot Z_2 \\ \vdots \\ P_n \cdot Z_n \end{pmatrix}, \tag{10}$$

where we have included more acoustic loads than we need, in order to get an over-determined system, which can be useful for improving the measurement results [4,12], and for checking if the source behaves as a linear system [10,11].

Alternatively the one-port model for the volume velocity source (see Fig. 2) can be expressed as:

$$Q_S - P \cdot \frac{1}{Z_0 Z_S} = \frac{P}{Z_0 Z}, \tag{11}$$

or with n loads:

$$\begin{bmatrix} 1 & -P_1 \\ 1 & -P_2 \\ \vdots & \vdots \\ 1 & -P_n \end{bmatrix} \begin{pmatrix} Q_S \\ 1/(Z_0 Z_S) \end{pmatrix} = \begin{pmatrix} P_1/(Z_0 Z_1) \\ P_2/(Z_0 Z_2) \\ \vdots \\ P_n/(Z_0 Z_n) \end{pmatrix}, \tag{12}$$

where we now solve for (Q_S) and ($1/Z_S$). In order to determine the normalised impedances of the acoustic loads (Z_i), used for experiments, a number of pressure transducers are usually mounted in the exhaust pipe. In the plane wave range we can use this information to perform wave decomposition and to determine the reflection coefficient looking into the acoustic load, which in turn gives the normalised load impedance.

To determine the complex load pressures (P_i) we need a reference signal to ensure that the pressure time histories for the different acoustic loads start at the same point in the engine cycle. The pressure time histories are then Fourier transformed and used to calculate the load pressures and load impedances and subsequently the source data.

For a linear source it can be expected that the source impedance results from both the source formulations (see Fig. 2) should converge towards the same results when a large degree of over-determination is used. Any discrepancy between the results can therefore be an indication of a nonlinear source behaviour.

As described above the two-load method requires complex pressure measurements and a reference signal unaffected by acoustic load variations, which is related to the sound generating mechanism of the source. For fluid machines with periodic operation cycle the normal solution is to try to obtain a trigger signal for instance giving one pulse per revolution [4]. This procedure can catch harmonic part of the spectrum generated by machine but not the broadband part. It can also be noted that a trig signal can also be used to reduce flow noise disturbances from measured pressure signals.

Although the two-load method is strictly valid only for a linear time-invariant source model it has been reported to give useful results also in situations that are not exactly time-invariant or linear [1,2,4]. By using a number of extra loads, a solution which is the best fit in least squares sense, can be obtained.

3.2. The nonlinear multiple-load method

The procedure for obtaining Eq. (6) is that $(p(t))$ and $(q(t))$ are first determined from measurements or simulations. The nonlinear function $(b(t))$ is then calculated followed by the Fourier transform of these quantities. It should be noted that an anti aliasing filter should be applied on $(b(t))$ to avoid aliasing problems caused by the presence of frequency components higher than half the sampling frequency. The load impedance (Z) is obtained from the ratio of the Fourier transform of $(p(t))$ and $(q(t))$ $(Z = P/Q)$. By using n acoustic loads (6) gives

$$\begin{bmatrix} Z_1 & -P_1 & -Z_1B_1 \\ Z_2 & -P_2 & -Z_2B_2 \\ \vdots & \vdots & \vdots \\ Z_n & -P_n & -Z_nB_n \end{bmatrix} \cdot \begin{pmatrix} P_S \\ Z_S \\ H_S \end{pmatrix} = \begin{pmatrix} P_1 \cdot Z_1 \\ P_2 \cdot Z_2 \\ \vdots \\ P_n \cdot Z_n \end{pmatrix}. \tag{13}$$

A minimum of three loads is required to solve for the source data while over-determination is used to reduce effects of measurement errors and deviations from the model just as for the two-load method. It should be noted that one difference between the method proposed here and the method of Jang and Ih [5] is that in the method proposed here the nonlinear term is expressed using boundary conditions seen from the source side towards the load while situation is reverted for method of Jang and Ih [5]. This makes the method of Jang and Ih [5] potentially more relevant for characterisation of nonlinear source behaviour. In their method it is however difficult to see how the particle volume velocity $(q(t))$ at the source cross section can be determined from experiments. In the method suggested here this is easier. It is also possible that the extra nonlinear term introduced will anyway give an improved result compared to the linear time-invariant model for nonlinear sources. This will be tested in the subsequent sections.

3.3. Linearity tests

To treat the problem of source characterisation in an efficient way an appropriate source model should be chosen. It is therefore important to apply linearity tests to assess whether a linear model is sufficient. If the linearity test indicates nonlinear behaviour, the use of nonlinear or hybrid methods is a natural step.

For the one-port sources a linearity test for direct methods has been proposed by Lavrentjev et al. [10]. Further linearity tests for in-direct or multi-load methods were proposed by Bodén and Albertsson [11]. The idea behind the tests is to verify that the source data (P_S, Z_S) are unchanged under acoustic load variations.

If we assume that we have a problem with m complex unknowns and make n measurements the over-determined equation system for determining the unknowns $\mathbf{x}$ can be written in the following way:

$$\mathbf{Ax} = \mathbf{b}, \tag{14}$$

where $\mathbf{A}$ is a $(n \times m)$ matrix, $\mathbf{x}$ is a $(m \times 1)$ vector and $\mathbf{b}$ is a $(n \times 1)$ vector. The idea is now to formulate tests, i.e. linearity coefficients, which can tell us if the measured data in $\mathbf{A}$ and $\mathbf{b}$ are consistent with the linear relationship (14). The number of measurements, n in Eq. (14), has to be larger than m for the linearity coefficients to be meaningful. If n equals m the linearity coefficients will always indicate a linear relationship. A linearity coefficient, which is similar to the coherence function, can following Ref. [11] be defined as

$$\gamma = \mathbf{x}^{-1}\mathbf{x} = \mathbf{b}^{-1}\mathbf{A}\mathbf{A}^{-1}\mathbf{b}, \tag{15}$$

where $\mathbf{x}^{-1}$ is interpreted as the pseudo-inverse of $\mathbf{x}$. This linearity coefficient will have a value in the interval $0 \leqslant \gamma \leqslant 1$, where the upper limit represents a perfect linear relationship. A drawback with such a test is that it is also sensitive to random errors. In Ref. [11] techniques to determine if a value lower than 1 is caused by nonlinearities or random noise are discussed.

To increase the sensitivity of the linearity test, the right hand side in the equation $\mathbf{Ax} = \mathbf{b}$ can be normalised to a unity vector, which means that every row in the equation system $\mathbf{Ax} = \mathbf{b}$ is divided by the corresponding right hand side.

4. Test cases

A theoretical model of a simple piston-restriction system, according to Fig. 3, was developed in Refs. [4,6] and analysed with forced oscillations. The solutions were obtained using the Harmonic balance method [6]. The duct system connected to the one-cylinder engine was modelled in the frequency domain, assuming that linear acoustic theory holds, while for the source part—a nonlinear time-domain model was used. The purpose of using this model is to be able to vary the degree of time-variation and nonlinearity. As the first part of the present study the influence of geometric parameters on the linearity of the source was studied. It was initially assumed that the flow velocity through the constriction would be the main factor determining the linearity. In order to investigate if this was true, linearity tests as described in Section 3.3, were first applied to various system configurations. Since it was easy to perform a parametric study using the computational H.B.M. code, a relatively large number of different parametric settings of the system were studied.

The results for a number of geometrically different configurations giving approximately the same velocity variation were compared. Fig. 4 shows an example of the flow velocity obtained for the three different configurations giving approximately the same flow velocity in the constriction. The three parameters varied (see Fig. 3) were: the piston amplitude (A), the piston diameter (d_1), and the diameter of the constriction (d_2). Only one of the parameters was varied at a time. Another parameter that was varied was the geometric compression ratio (ε) of the cold engine model, the distance (L_V) between TDC (top dead centre) of the piston and the constriction was decreased compared to the original value in the simulations.

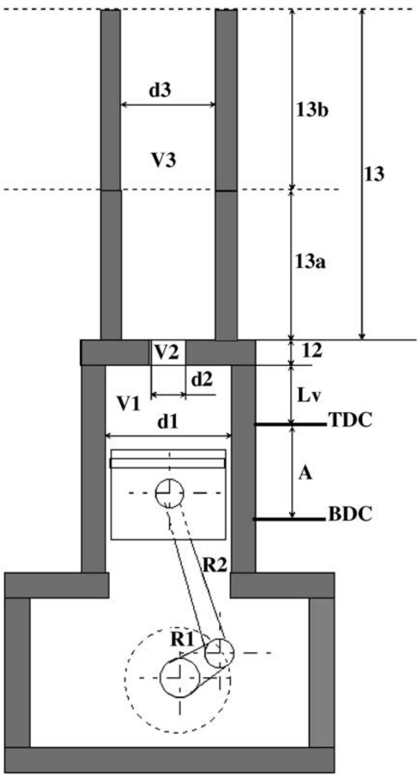

Fig. 3. Piston-constriction system studied.

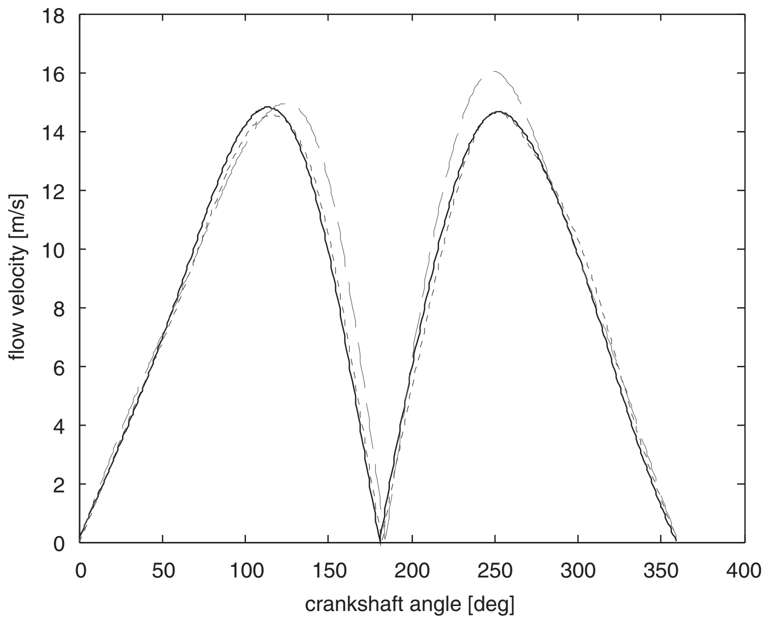

Fig. 4. Flow velocity in the constriction for three geometric configurations: $A = 0.160\,\mathrm{m}$, $d_2 = 0.030\,\mathrm{m}$, $d_1 = 0.070\,\mathrm{m}$ (full line); $A = 0.080\,\mathrm{m}$, $d_2 = 0.021\,\mathrm{m}$, $d_1 = 0.070\,\mathrm{m}$ (dotted line); $A = 0.080\,\mathrm{m}$, $d_2 = 0.030\,\mathrm{m}$, $d_1 = 0.099\,\mathrm{m}$ (dashed line).

In the second part of the study the new nonlinear multi-load source characterisation method was implemented and tested for various source configurations. In order to evaluate the new technique and to compare the results with the conventional two-load technique simulated system configurations with varied linearity and time-variance were used.

In order to test the new source characterisation technique on experimental data the results from one-cylinder valve-less compressor measurements previously used in Ref. [13] and a 6-cylinder turbocharged truck diesel engine exhaust system were used. During the truck diesel engine measurements 12 different acoustic loads (10 side-branches with lengths from 0 to 3130 mm and one muffler with and without particulate trap) were used as acoustic loads. Measurements were made for various engine operating points: 4 different engine speeds (1200, 1400, 1600, and 1800 rev/min) and 3 engine loads (25%, 50%, and 100% of full load).

5. Results from numerical simulations

5.1. Source linearity study

The acoustic impedance of the loads and the pressure, for the piston-constriction system shown in Fig. 3, at a cross section just outside the constriction were extracted from the simulations. These data were then analysed using the linearity tests described in Section 3.3. It was concluded that the linearity of the studied system was not completely controlled by the flow velocity through the constriction, although this parameter was important. Fig. 5 shows the linearity coefficient (γ) according to (15) for three different geometric configurations giving approximately the same constriction velocity. The difference between the results for the three geometric configurations, for the six first harmonics studied, was less than 0.1. The normalised linearity coefficient (γ_n), shown in Fig. 6, shows a similar difference in the results for different geometric configurations. The normalised linearity coefficient value decreases for higher harmonics indicating an influence of nonlinear effects. The velocity profile in the constriction was reasonably comparable for the different parametric configurations with only slight differences, see Fig. 4. The results of these studies of the dependence of system linearity on geometrical parameters were used in the next section to see if there is a correlation between linearity according to the linearity tests and improvement in results obtained using the new nonlinear source model.

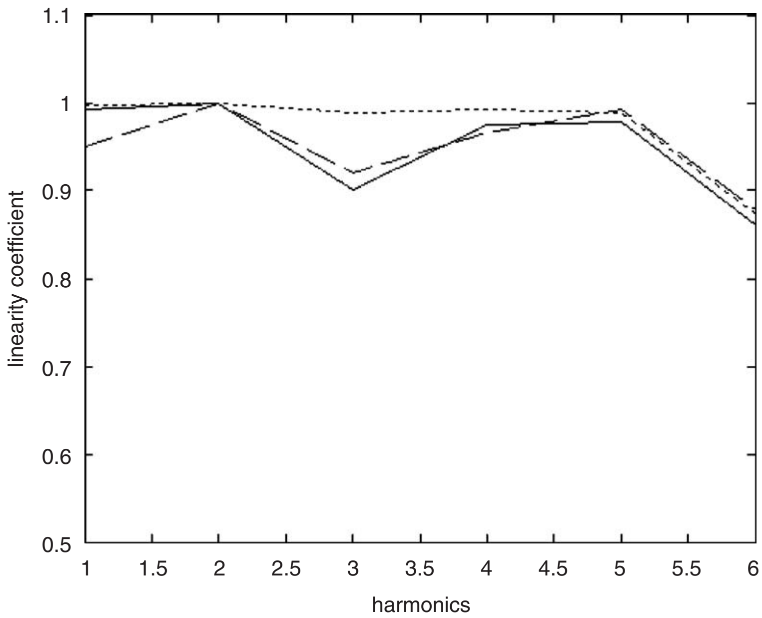

Fig. 5. Linearity coefficient γ; $A = 0.160$ m, $d_2 = 0.030$ m, $d_1 = 0.070$ m (full line); $A = 0.080$ m, $d_2 = 0.021$ m, $d_1 = 0.070$ m (dotted line); $A = 0.080$ m, $d_2 = 0.030$ m, $d_1 = 0.099$ m (dashed line).

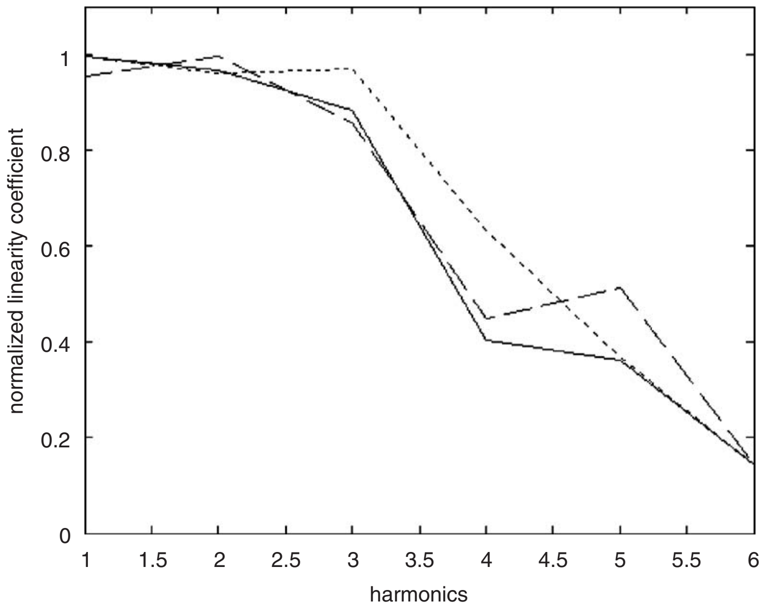

Fig. 6. Normalised linearity coefficient γ_n; $A = 0.160$ m, $d_2 = 0.030$ m, $d_1 = 0.070$ m (full line); $A = 0.080$ m, $d_2 = 0.021$ m, $d_1 = 0.070$ m (dotted line); $A = 0.080$ m, $d_2 = 0.030$ m, $d_1 = 0.099$ m (dashed line).

5.2. Results from source data extraction

In the simulations, 10 load ducts with a stepwise increased length (l_3) from 0.3 to 2.1 m with a step of 0.2 m, were used as the acoustic loads connected to the source. Based on acoustic load impedances and pressures obtained from the simulations the source data was extracted using the two-load technique and the new nonlinear multi-load method. In order to have a sufficient over-determination, the data from eight acoustic loads out of the 10 were included in the calculations of the source data following the procedures given in Section 3. The remaining two data sets were used for checking the result by predicting the pressure in the load duct using the extracted source data. Different combinations of loads were used to investigate the influence of the choice of loads on the results. The procedure was repeated until the pressure predictions were performed for all the described loads and for all the 16 system configurations, altogether for 160 different cases. The result of the prediction was evaluated by comparing the results obtained using the new nonlinear multi-load method and the conventional two-load method with directly simulated results. The prediction results were classified into five categories and an attempt was made to statistically analyse the influence of linearity and time-invariance.

The nonlinear multi-load method gave better prediction results for 71 different system configurations, while the two-load method was slightly better in 43 cases. In 46 investigated cases the prediction results were considered to be equally good for both the tested methods. A better agreement between predicted and directly simulated data was noticeable when the linearity coefficients (γ) and (γ_n) indicated more linear system behaviour.

Some examples of the source data extraction results are presented in Figs. 7–11. Fig. 7 shows an example where good prediction results for the sound pressure level in the load duct were obtained using both techniques. In this case the piston amplitude had been decreased to an extremely low level which should give a linear result. Fig. 8 presents an example of typical prediction of sound pressure level in a system with high compression ratio, were the new nonlinear multi-load method normally gave better results. In Figs. 9–11 results from the configurations already discussed in connection with Figs. 4–6, where the flow velocity in the constriction was kept almost constant, are shown. The agreement between predicted and simulated results differs substantially and it can be related to the varying system linearity, see Figs. 5 and 6.

The evaluation of the influence of time-variance on the source characterisation results obtained from the simulations of the system configurations with varied piston amplitude (A) showed that both the two-load and the nonlinear multi-load method gave better prediction results when the system was more time-invariant. It was also noticed that the conventional two-load method showed a higher sensitivity to the time-variance than the nonlinear multi-load method. An example of the results, obtained for model configuration with high

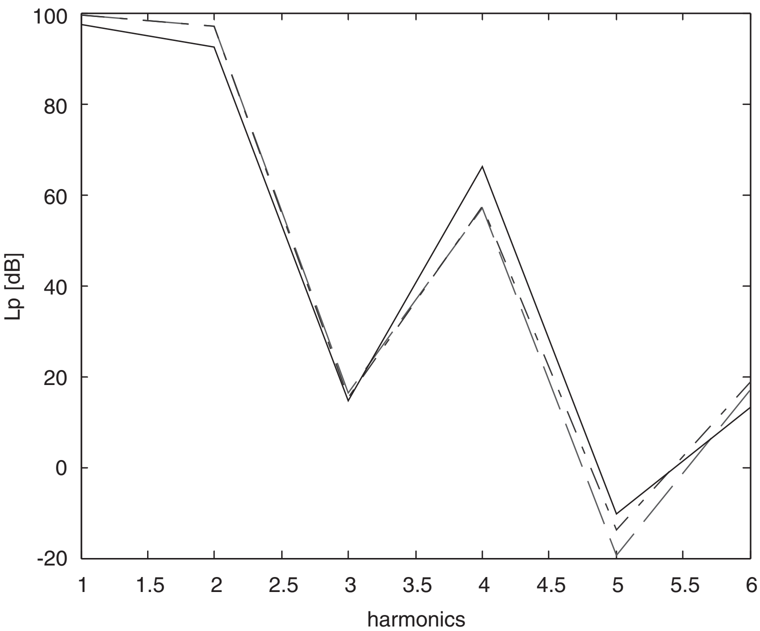

Fig. 7. Sound pressure level for a source configuration with low time variation, $A = 0.0008$ m, $l_3 = 1.3$ m; two-load technique (dashed line), nonlinear multi-load technique (dash–dotted line), direct simulation (full line).

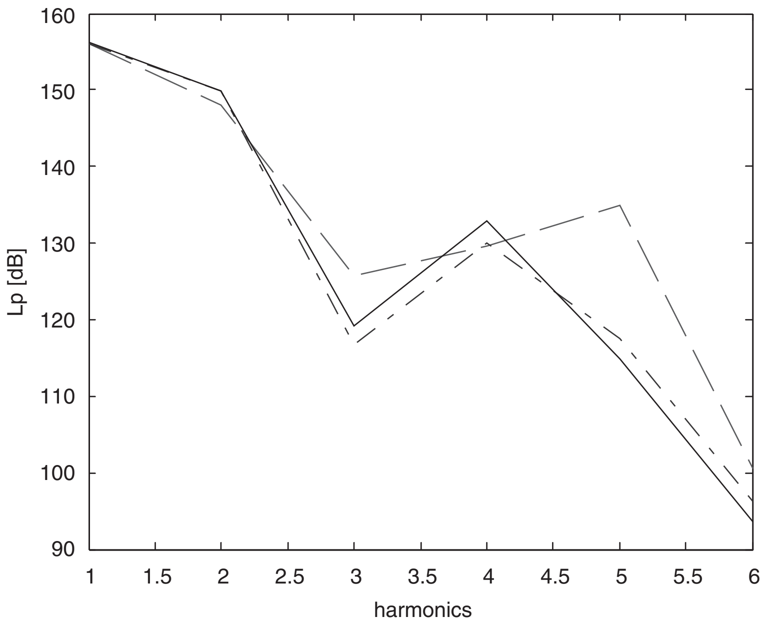

Fig. 8. Sound pressure level for a source configuration with high compression ratio, $L_V = 0.12\,\mathrm{m}$, $l_3 = 0.5\,\mathrm{m}$; two-load technique (dashed line), nonlinear multi-load technique (dash–dotted line), direct simulation (full line).

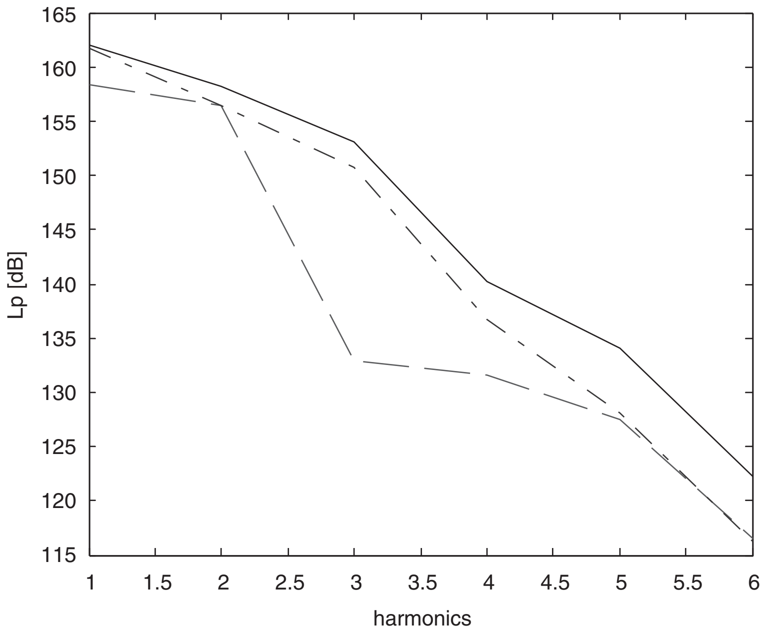

Fig. 9. Sound pressure level, $A = 0.080\,\mathrm{m}$, $d_2 = 0.030\,\mathrm{m}$, $d_1 = 0.099\,\mathrm{m}$, $l_3 = 0.5$; two-load technique (dashed line), nonlinear multi-load technique (dash–dotted line), direct simulation (full line).

time-variance, is shown in Fig. 12. These curves should be compared to the results shown in Fig. 7 for a case with low time variation.

In Figs. 13–15 the new method has been tested using a highly nonlinear model configuration with geometrical parameters similar to those used in diesel powered IC-engines. The linearity coefficient for this

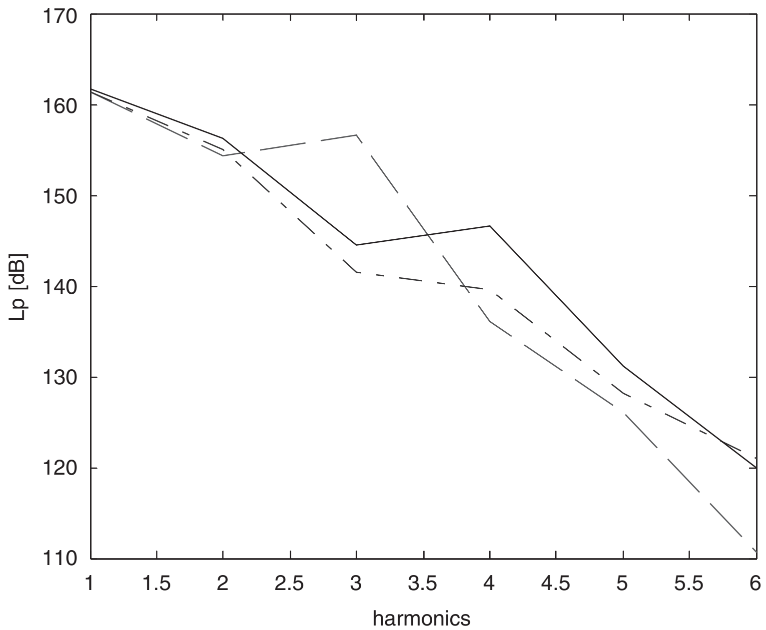

Fig. 10. Sound pressure level, $A = 0.16\,\mathrm{m}$, $d_2 = 0.030\,\mathrm{m}$, $d_1 = 0.070\,\mathrm{m}$, $l_3 = 0.3\,\mathrm{m}$; two-load technique (dashed line), nonlinear multi-load technique (dash–dotted line), direct simulation (full line).

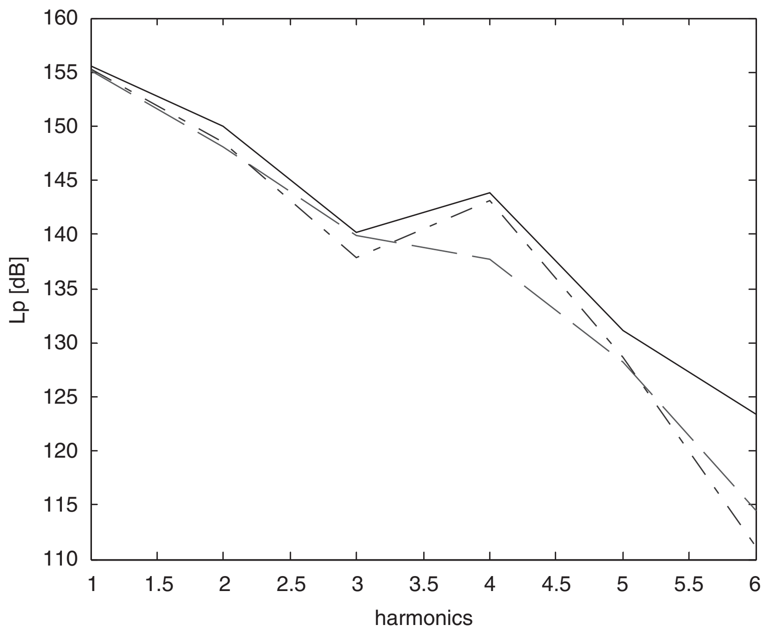

Fig. 11. Sound pressure level, $A = 0.080\,\mathrm{m}$, $d_2 = 0.0212\,\mathrm{m}$, $d_1 = 0.070\,\mathrm{m}$, $l_3 = 0.3\,\mathrm{m}$; two-load technique (dashed line), nonlinear multi-load technique (dash–dotted line), direct simulation (full line).

system configuration with high compression ratio is shown in Fig. 16. It can be seen from Fig. 13 that the nonlinear multi-load technique gives a better result for this case. In Figs. 14 and 15 the resulting source impedance when a pressure source and a volume velocity source model have been used in the two-load method is compared. Rather big differences can be seen especially at frequencies where the linearity coefficient indicates nonlinearity.

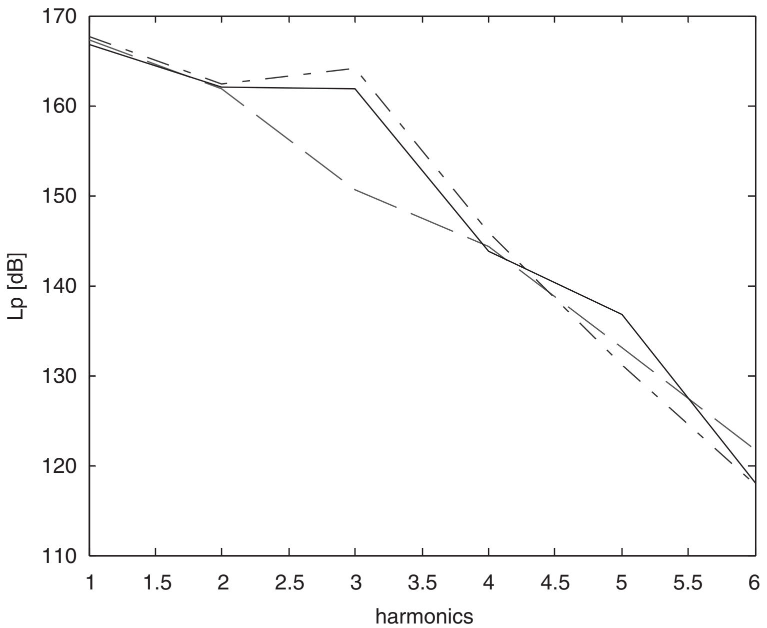

Fig. 12. Sound pressure level for a source configuration with high time variance, $A = 0.24\,\mathrm{m}$, $l_3 = 0.7\,\mathrm{m}$; two-load technique (dashed line), nonlinear multi-load technique (dash–dotted line), direct simulation (full line).

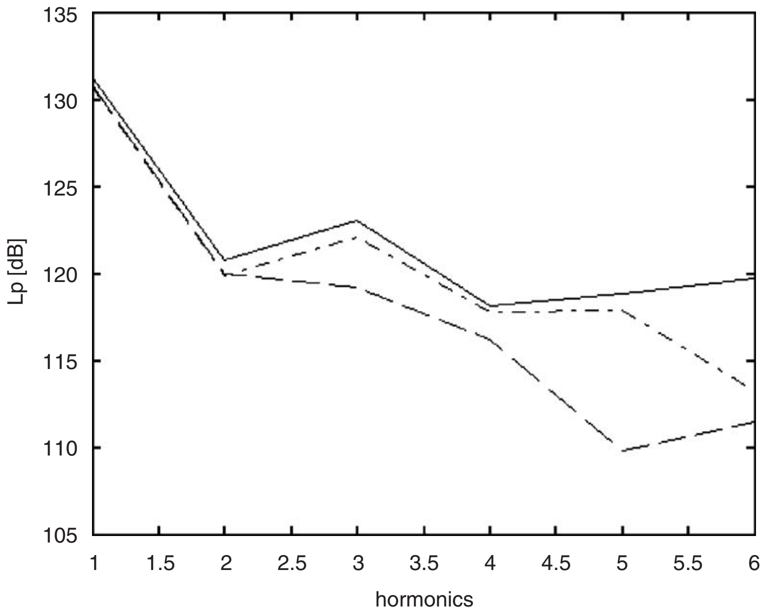

Fig. 13. Sound pressure level for a highly nonlinear model configuration, $d_2 = 0.002\,\mathrm{m}$, $L_V = 0.1\,\mathrm{m}$, $A = 2.0$, $l_3 = 0.3\,\mathrm{m}$; two-load technique (dashed line), nonlinear multi-load technique (dash–dotted line), direct simulation (full line).

The source data obtained from the simulations using the conventional two-load method with 10 loads included in calculations were finally compared to the source strength and source impedance based on the measurements performed by Bodén [4,13], see Fig. 17. Fig. 17 shows the imaginary part of the source impedance and it can be seen that there is a similarity in frequency dependence except for the first harmonic

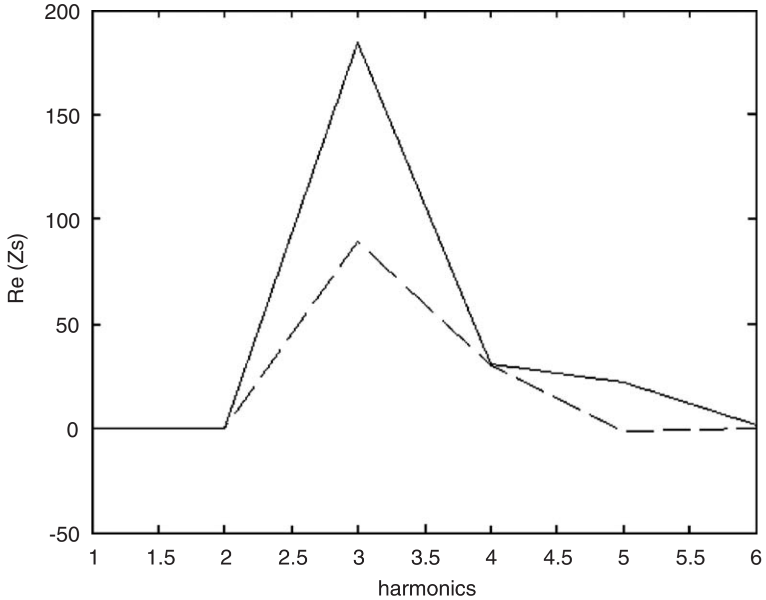

Fig. 14. Real part of normalised source impedance for a highly nonlinear model configuration, $d_2 = 0.002\,\mathrm{m}$, $L_V = 0.1\,\mathrm{m}$, $A = 2.0$; constant pressure source model (full line), constant volume velocity source model (dashed line).

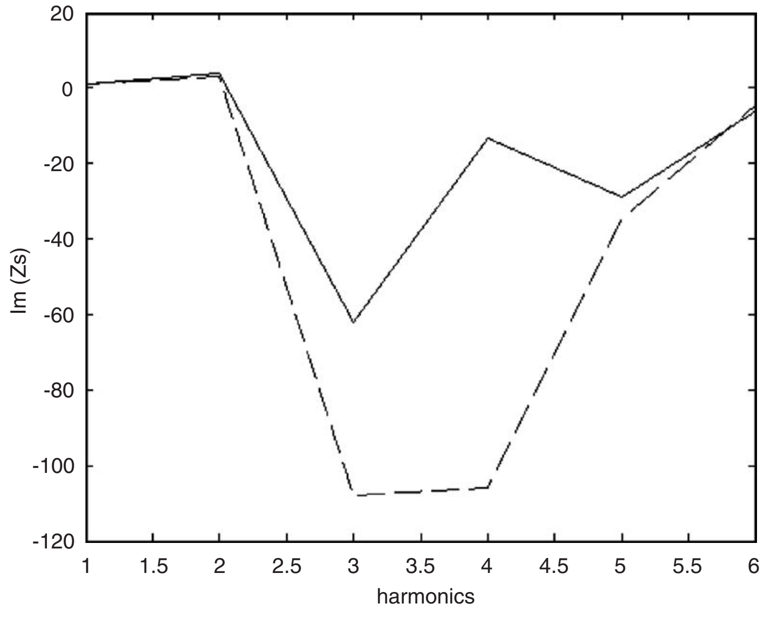

Fig. 15. Imaginary part of normalised source impedance for a highly nonlinear model configuration, $d_2 = 0.002\,\mathrm{m}$, $L_V = 0.1\,\mathrm{m}$, $A = 2.0$; constant pressure source model (full line), constant volume velocity source model (dashed line).

where the simulation, obtained using two-load technique gives a result not consistent with the Helmholtz resonator like result expected for low frequencies. However the source impedance extracted from the simulated data using the new multi-load technique gives considerably better agreement with the measured results compared to the conventional two-load technique. The latter could indicate that the unexpected behaviour of the model at the first harmonic is caused by the system nonlinearity. The agreement between simulated and measured real part of impedance and source strength are also far from perfect. It can also be

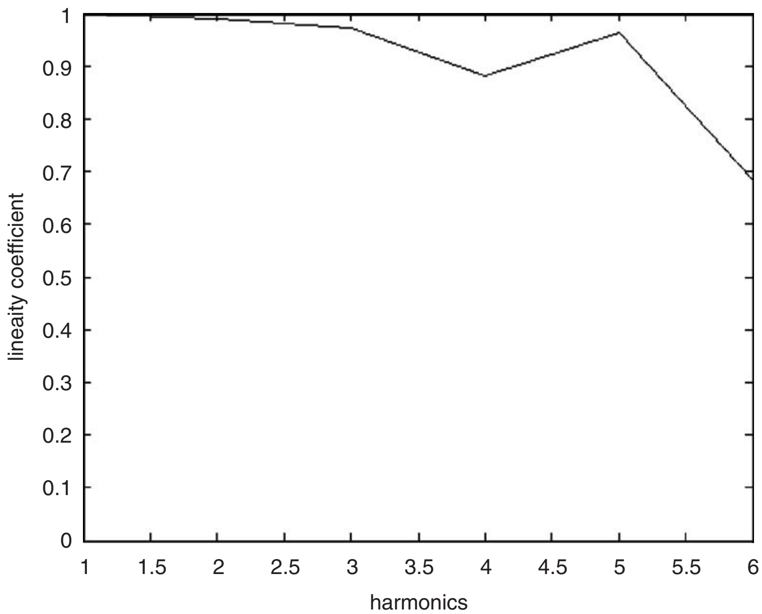

Fig. 16. Linearity coefficient γ for a highly nonlinear model configuration, $d_2 = 0.002\,\mathrm{m}$, $L_V = 0.1\,\mathrm{m}$, $A = 2.0$.

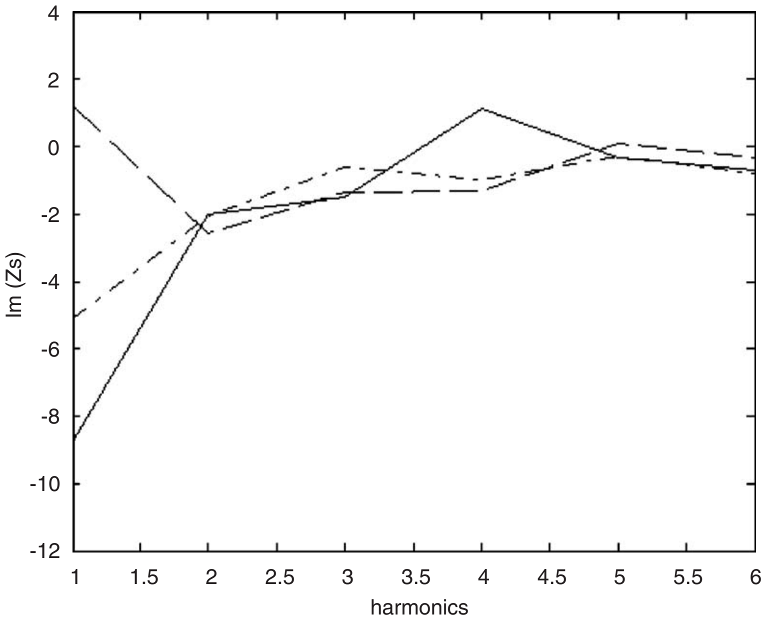

Fig. 17. Imaginary part of normalised source impedance; obtained from measurements using two-load technique (full line), obtained from simulations using two-load technique (dashed line), obtained from simulations using nonlinear multi-load technique (dash–dotted line).

noted that the measurements give unphysical negative resistance values at the same frequencies where the imaginary part gives an unexpected result. Negative resistance values in the source impedance have been reported in a number of studies [1,2,4,14] where it has been suggested that the cause could be nonlinearity or time variance. The negative resistance values could therefore be caused by nonlinear or time-variation effects not fully accounted for in the simulation model.

6. Results from experimental tests on valve-less one-cylinder cold engine

An example of sound pressure prediction in a load duct obtained from the source data from the one-cylinder compressor measurements [4,13] is presented in Fig. 18. The agreement between predicted and measured sound pressure levels was reasonable for both the two-load method and the nonlinear multi-load

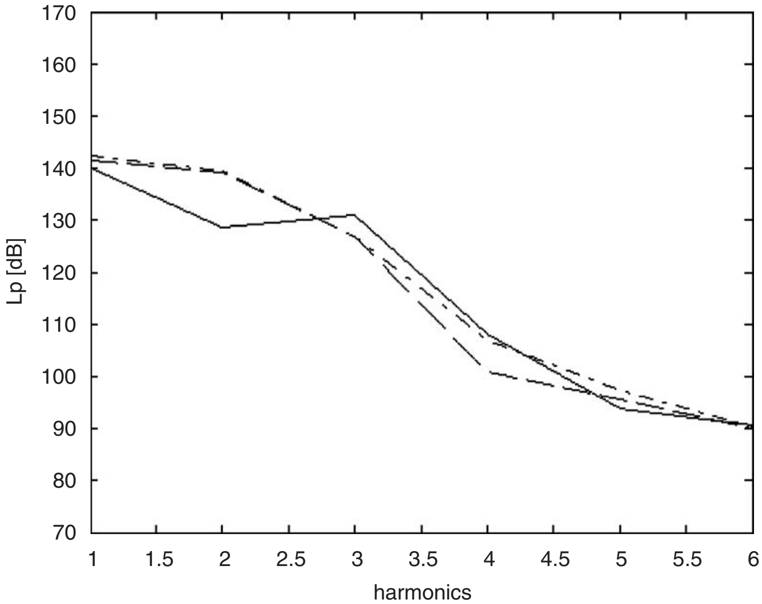

Fig. 18. Sound pressure level in a load duct with length 1.5 m; obtained from measurements using two-load technique (full line), obtained from simulations using two-load technique (dashed line), obtained from simulations using nonlinear multi-load technique (dash–dotted line).

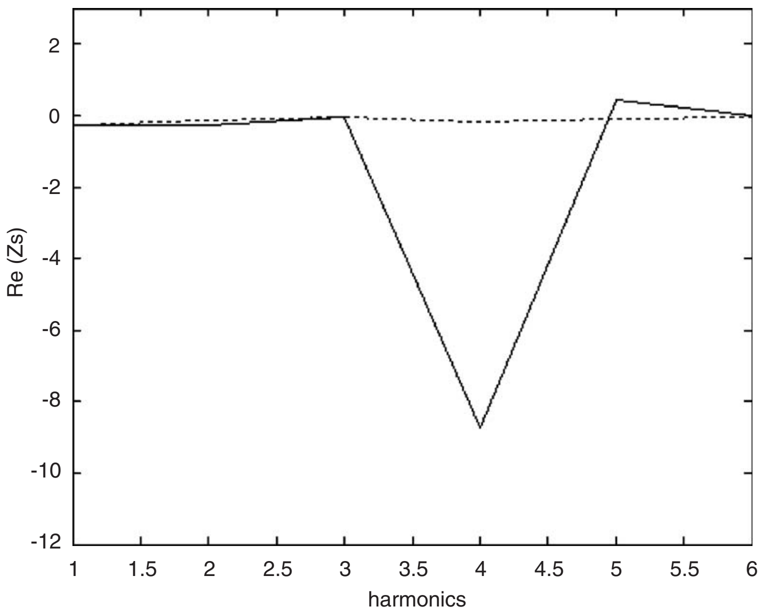

Fig. 19. Real part of measured normalised source impedance; constant pressure source model (full line), constant volume velocity source model (dotted line).

method for most of the tested crankshaft rotation speeds and system configurations. From Figs. 19 and 20, where the real and imaginary part of the source impedance is calculated using the constant pressure source model and the volume velocity based source model, it can be noticed that except at 1 or 2 harmonics the source exhibited a relatively linear behaviour. The linearity coefficient of the cold engine is presented in Fig. 21.

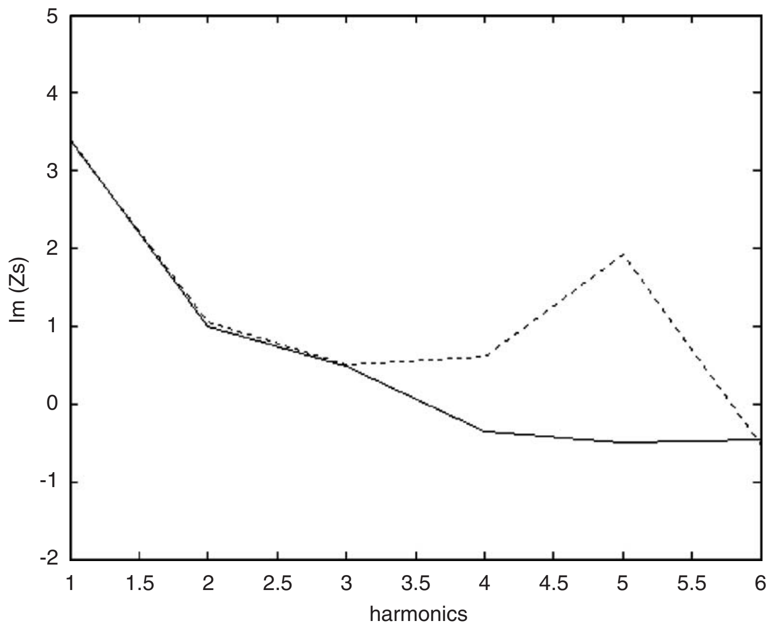

Fig. 20. Imaginary part of measured normalised source impedance; constant pressure source model (full line), constant volume velocity source model (dotted line).

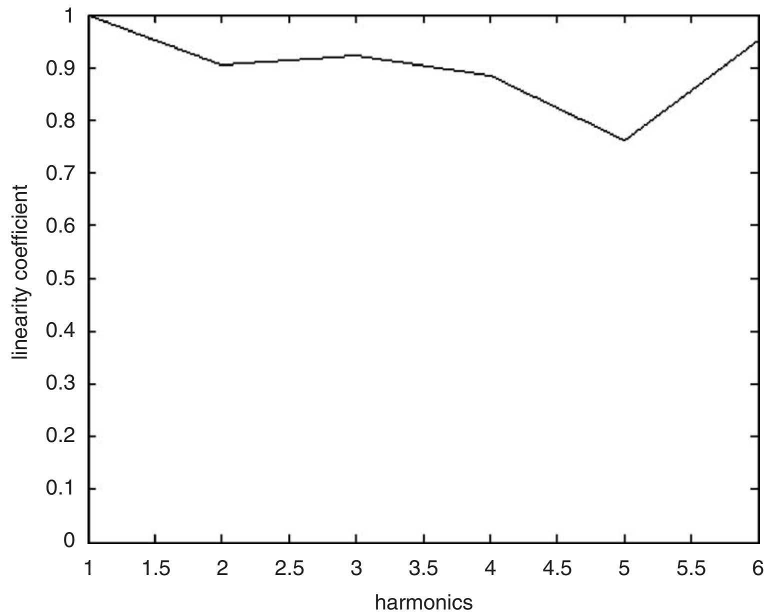

Fig. 21. Linearity coefficient γ for one-cylinder valve-less cold engine.

It should be noticed that at most of the harmonics where the source behaved linearly the difference between the results obtained from modified multi-load method calculation and the two-load method calculation was marginal. The advantage of the new method can be seen at the harmonics where the nonlinearity of the system occurred.

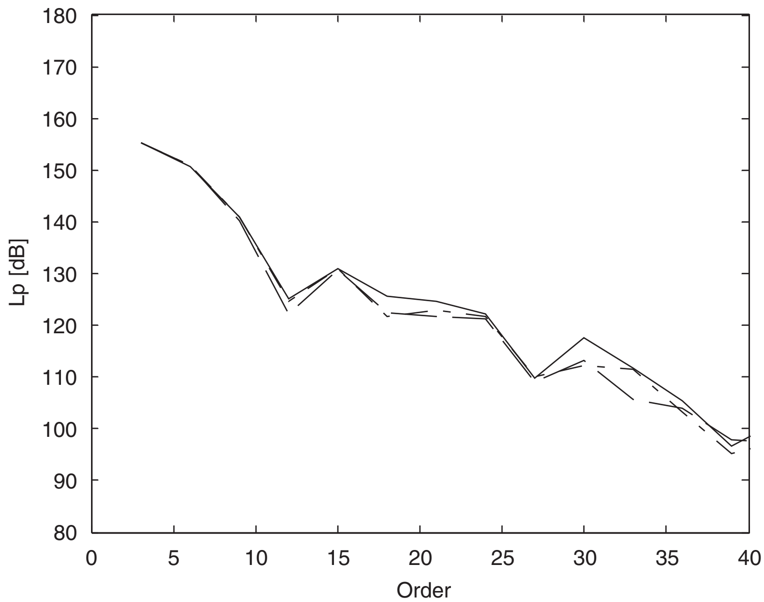

Fig. 22. Sound pressure level in exhaust system for 50% engine load and 1600 rev/min; two-load technique (dashed line), nonlinear multi-load technique (dash–dotted line), direct measurements (full line).

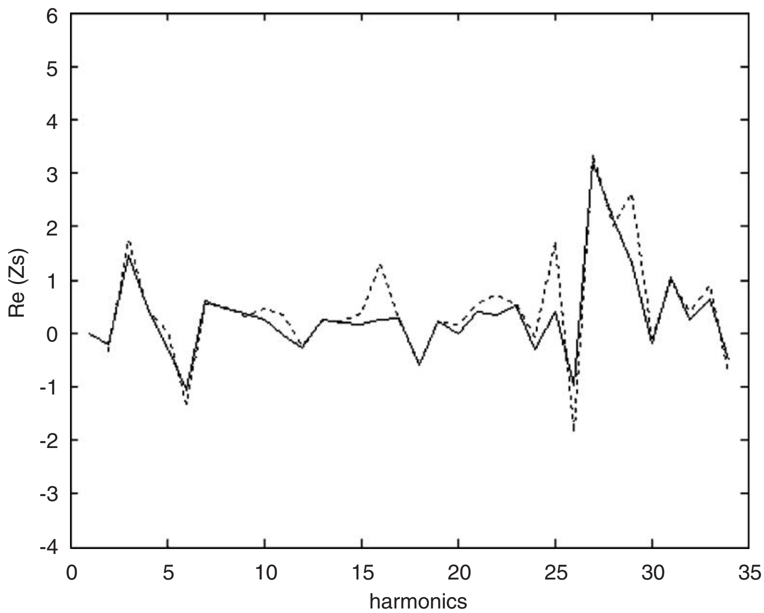

Fig. 23. Real part of normalised source impedance; 50% engine load and 1600 rev/min; constant pressure source (full line), constant volume velocity source (dotted-line).

7. Results from experimental tests on IC-engines

An example of source data extraction results for a truck diesel engine is presented in Fig. 22. The agreement between predicted and measured sound pressure levels was good for both the two-load method and the nonlinear multi-load method for most of the tested engine loads and speeds. This engine exhibited a relatively linear behaviour. The linear behaviour of the engine can also be seen from the Figs. 23 and 24, where the real and imaginary part of the source impedance calculated using the constant pressure source model are compared

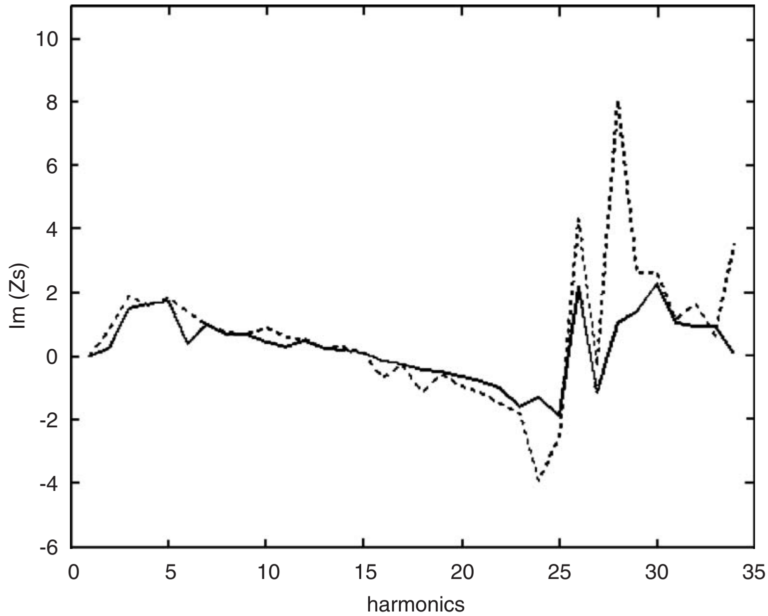

Fig. 24. Imaginary part of the source impedance; 50% engine load and 1600 rev/min; constant pressure source (full line), constant volume velocity source (dotted-line).

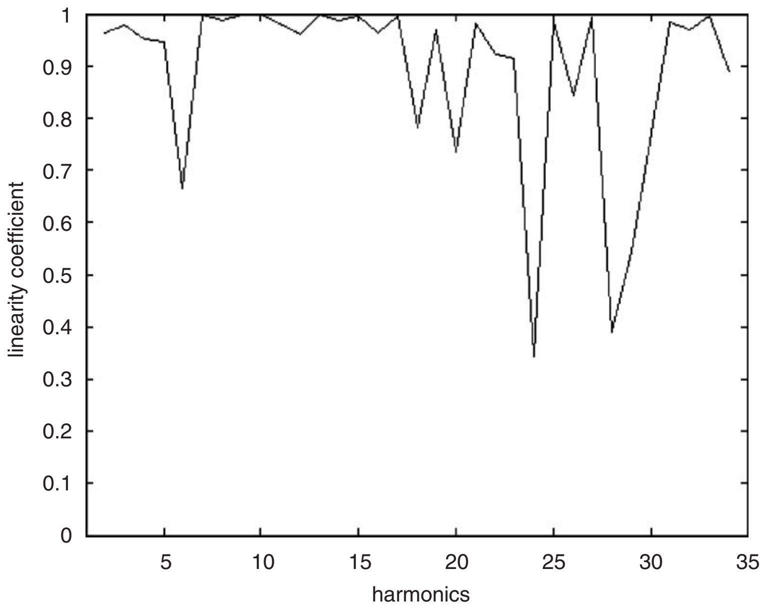

Fig. 25. Linearity coefficient γ for 6-cylinder turbocharged truck diesel engine, 50% engine load and 1600 rev/min.

to the source data obtained using the constant volume velocity based source model. The linearity coefficient of the engine is presented in Fig. 25.

8. Conclusions

A new nonlinear source model and a multi-load technique for extracting the source data have been presented. The new method has been tested using numerical simulations for a cold one-cylinder valve-less engine. A study was made on a number of geometrical configurations that were nonlinear and time-varying. This information was used to try to correlate the outcome of the new method with the nonlinearity and time variance.

Better source characterisation results were obtained when the linearity coefficients indicated more linear system behaviour.

The nonlinear multi-load technique gave better results for more nonlinear sources, compared to the conventional two-load technique.

Both the source data extraction techniques gave better source characterisation results when the system was more time-invariant. The nonlinear multi-load method gave better results compared to the two-load technique when the source was more time-varying, even though the model used is time-invariant.

The method has also been tested on experimental data from a one-cylinder valve-less compressor and from 6-cylinder turbocharged truck diesel engine. Better source characterisation results were obtained, using the new nonlinear multi-load method compared to the two-load method, when the source under test exhibited nonlinear behaviour.

The new nonlinear indirect source characterisation technique proposed requires one additional acoustic load compared to the two-load technique. Since over-determination is anyway used in many cases the additional data would often be available. It has been shown that the new technique gives improved results compared to the two-load technique if the source is nonlinear or time-varying. For cases when the source is linear and time-invariant the new technique gives the same result as the two-load technique. This means that there is no risk for increased errors when the source is linear and time-invariant which can happen if a linear time-varying source model is used [4]. It can therefore be recommended that the new technique proposed is used whenever sufficient data is available.

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