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Measurement of volume velocity of a small sound source

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ABSTRACT

Two methods for measuring volume velocity of a back-enclosed driver, that make no assumptions about the shape or the vibration distribution of the driver's diaphragm, are investigated: a compression chamber and a blocked pipe. Both methods were implemented on an off-the-shelf driver using a microphone installed in the driver's cavity. The relationship between the pressure inside of the driver's cavity and the volume velocity of the driver's diaphragm was established by measurement. The two methods produced similar results.

1. Introduction

An engineering implementation of a simple source can be a small back-enclosed driver, provided that its driving surface is small compared to the wavelength and vibrates in phase [1]. In such a case the pressure response does not depend on the details of the vibrating surface and can be expressed by a point transfer impedance Z. Such an impedance relates volume velocity amplitude $\widehat{Q}$ of the driver located at a source point $\mathbf{s}$ to sound pressure amplitude $\hat{p}$ at a field point $\mathbf{f}$

$$Z(\mathbf{f}|\mathbf{s}) = \hat{p}(\mathbf{f})/\widehat{Q}(\mathbf{s}). \tag{1}$$

All the quantities in Eq. (1) are complex functions of frequency, and hat [^] is used to denote amplitude [2]. The driver becomes increasingly inefficient when the frequency decreases, whereas at higher frequencies it develops pronounced directivity and thereby ceases to be a simple source [1]. In the mid-frequency range the transfer impedance can be measured, independently of the choice of driver, provided that the volume velocity of the source is known.

The need for a volume velocity source has been motivated by the demand for quantifying radiation by vibroacoustic sources. The principle of such a characterisation is to replace the complex vibroacoustic source by a simpler substitute source for use in noise synthesis. It is presumed that the radiation can be modelled by superposition of simple sources set in a rigid closed baffle of similar volume and shape as the original source. Such a characterisation critically depends on the knowledge of the source volume velocities.

One way of measuring volume velocity of a driver is to measure the velocity of the voice-coil, and multiply it with the projected surface of the driver's diaphragm in the direction parallel to its motion. The velocity of the voice-coil may be deduced by knowledge of the blocked electrical impedance and the motional impedance of the driver [1]. The measurement of motional impedance can thus be used for an estimation of volume velocity assuming that the diaphragm moves as a rigid body. This requires prior knowledge of the blocked electrical impedance which can be found by e.g. casting the driver's moving parts into an epoxy resin [3]. Thus a second driver is needed for the measurement of motional impedance. This makes the method sensitive to differences between the two drivers [3]. Furthermore the motional impedance depends on the ambient space. Therefore the approach is rather cumbersome, and the supply voltage is not proportional to volume velocity.

Several designs for implementing volume velocity sources have been reported [4–6]. Common to all of these designs are that an additional transducer producing a signal proportional to either velocity, acceleration or volume displacement is used. The relationship to volume velocity can then be either deduced by theory or measured.

Three different designs of volume velocity sources have been reviewed by Salava [4]. The first design is based on use of a supplementary porous acoustic resistor. The advantage of the design is that it can be assembled quickly, but the disadvantage is that for accurate measurements careful calibration of the resistor is necessary and the resistor may not be linear in regards to the volume

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velocity [4]. The second design uses a microphone which provides a signal proportional to volume displacement. The disadvantage of the design is that in order to obtain a signal directly proportional to volume velocity a derivative circuit is required [4]. The third design is to equip a rigid piston, driven by an electrodynamical transducer, with a measuring voice-coil [4]. The advantage of the design is that the output signal is proportional to the velocity of the voice-coil, but the disadvantage is that it assumes mechanical rigidity of the moving parts which is not met in practice [4].

A more recent design described by Salava [5] uses two coupled drivers put together face-to-face: one acting as an exciter, and the other as a sensor. The exciting driver is in a rigid enclosure, and given that there is no supply voltage in the measuring and radiating driver's voice-coil the output voltage is proportional to its velocity. Another design is to equip the diaphragm with an accelerometer [5]. The relationship between the volume velocity and the transducer's signal is however not known, and experimental calibration is carried out in a free-space. The disadvantage of such a calibration is that it requires access to an anechoic room.

Anthony and Elliott [6] have investigated two designs of known volume velocity sources. One of the designs is based on the previously mentioned method by Salava, and uses two identical drivers put together face-to-face. The volume velocity can be estimated by summing up individual contributions of smaller patches each considered to be in rigid motion. The calibration is based on measurement of multiple point velocity - voice-coil voltage transfer functions using laser velocimetry. The volume velocity was then expressed in terms of an effective area. The disadvantage of such a calibration is that the effective area is not straightforward to measure, and the method requires access to a Laser Doppler Vibrometer. The second design uses an internal microphone installed in the driver's back cavity of precisely known volume which had to be designed and manufactured. Here the volume velocity was deduced from the internal pressure assuming a compliance law theoretically valid for small cavities of rigid walls. The two designs were compared using a single-point velocity as a reference measurement of volume velocity.

In this study the latter technique using an internal microphone was applied on a small off-the-shelf driver. In this case the calibration between pressure and volume velocity was not modelled, as done in [6], but instead had to be measured in dependence of frequency due to the presence of internal damping material and the effect of cavity resonances. Such a calibration is advantageous because the features of the driver's back cavity do not have to be known and may vary with frequency. The disadvantage is the need to independently measure the volume velocity which has turned out not to be a trivial task.

Four different calibration methods were compared in [7]. The first calibration was based on laser velocimetry, as done in [6]. The technique was found difficult to apply on curved surfaces such as dome shaped diaphragms. Therefore the volume velocity was deduced from a single point velocity measurement in the centre of diaphragm. This requires rigidity of the diaphragm which is not met in practice. The second calibration was performed in a free-space, as done in [5], but using a large flat baffle to avoid radiation from the driver's enclosure. The accuracy of the measured data was found to suffer from baffle diffraction.

The inconveniences have prompted the authors to find alternative ways to calibrate the source. Two novel methods were thus conceived: a compression chamber technique and a blocked pipe technique. The key advantage of the two methods, which will be described in detail in this paper, is that no assumptions are made regarding the shape or the vibration distribution of the driver's diaphragm.

2. Method

The principle of measuring transfer impedance based on an internal microphone is discussed in Section 2.1. The transfer impedance is split into two transfer functions: a source function and a space function. A driver is characterised by its source function. In order to estimate the source function volume velocity has to be measured. The two methods for measuring the driver's volume velocity are discussed further on in Section 2.2.

2.1. Internal pressure technique

If the volume of air inside the back cavity of the driver is tightly closed, and if its back enclosure is small and rigid, then sound pressure p inside of the cavity is effectively proportional to volume velocity Q of the diaphragm when it compresses and expands the interior air, $p \propto Q$. The assumption of a tightly closed cavity may not be fully true: drivers are often equipped with either a small vent or a porous diaphragm for the compensation of changing ambient pressure. Such a compensation is however practically ineffective where sound pressure is concerned and needs not be accounted for.

Using an internal microphone, the transfer impedance, Eq. (1), can be rewritten in a form suitable for experimental work. The transfer impedance will be split into two independent transfer functions: a source function Ψ which relates internal pressure $\hat{p}(\mathbf{i})$ to volume velocity $\widehat{Q}(\mathbf{s})$ and a space function Ω which relates external pressure $\hat{p}(\mathbf{f})$ to internal pressure $\hat{p}(\mathbf{i})$

$$Z = \Psi\Omega, \quad \Psi = \hat{p}(\mathbf{i})/\widehat{Q}(\mathbf{s}), \quad \Omega = \hat{p}(\mathbf{f})/\hat{p}(\mathbf{i}). \tag{2}$$

Here $\mathbf{i}$ denotes the position of the internal reference microphone. A driver's diaphragm is characterised by its source function which is theoretically governed by compliance-like behaviour of the air inside of its back cavity. This transfer function depends on ambient factors such as temperature and will be discussed in detail in Section 2.2.1. This characterisation procedure is therefore approximate as it depends on the ambiance.

2.1.1. Modelling the source function by a polynomial

A compliance-like behaviour implies that the source function is inversely proportional to frequency. A measured source function Ψ may in practice not have such an ideal behaviour and can be perturbed by noise. A remedy is to fit the source function to a polynomial

$$j\omega\widetilde{\Psi} = \psi_0 + j\omega\psi_1 - \omega^2\psi_2 + \ldots, \tag{3}$$

of order N using a least squares fit were tilde [~] denotes a fitted estimate [8]. The angular frequency is denoted by ω and the imaginary unit is denoted by j. A model of the source function is required in order to interpolate the data, as will be discussed in conjunction with the blocked pipe method in Section 2.2. If the order N of the model is chosen too high, the polynomial over-fits the acquired data. A concern was therefore how to choose the order of the polynomial.

2.1.2. Interpreting the source function as a filter

The small volume inside the driver's back cavity may be represented as a filter [1]. Since the distance between the internal microphone and the diaphragm is small compared to the wavelength, $\|\mathbf{i} - \mathbf{s}\| \ll \lambda$, and assuming that the pressure inside of the cavity is spatially uniform at low frequencies the sound pressure at the driver's surface is $p(\mathbf{s}) \approx p(\mathbf{i})$. As the cavity has its own resonance frequencies the compliance-type behaviour will be valid at frequencies well below the first cavity resonance. At these frequencies the source function can be represented as an impedance of a damped one-degree of freedom system

$$\widetilde{\Psi} \approx \hat{p}(\mathbf{s})/\widehat{Q}(\mathbf{s}) = \frac{1}{j\omega C} + R + j\omega M, \tag{4}$$

where the frequency independent and real valued constants are a compliance C, resistance R and inertance M [1]. The damping in the cavity, created by the incorporated absorbing layer, is accounted for in the resistance R. This impedance model, Eq. (4), suggests that the polynomial, Eq. (3), should be of second order.

2.2. Volume velocity estimation

The volume velocity, and thereby the source function, can be assessed in specific space conditions. In this investigation two spaces are considered: the first is a small compression chamber, and the second is interior of a closed pipe.

2.2.1. Compression chamber method

Let the driver's diaphragm be coupled to a small loss-less chamber of impenetrable surface such that the chamber's cavity is compressed and expanded by the vibrating diaphragm. Inside of the cavity density and pressure are related by

$$\frac{p}{p_0} = \left(\frac{\rho}{\rho_0}\right)^{\gamma}, \tag{5}$$

where the instantaneous pressure p is given by the sum of ambient pressure p_0 and sound pressure p_e [1]. The instantaneous density is denoted by ρ, the density at rest is denoted by ρ_0 and the ratio of specific heats is denoted by γ. The velocity is taken to be positive when the diaphragm moves into the volume V, in which case the internal air is compressed. The mass inside the chamber is conserved, and the volume velocity is related to the change of volume $dV = V_0 - V$ by $Q = \frac{\partial}{\partial t} dV$ given by

$$Q = \frac{\partial}{\partial t} V_0 \left[1 - \left(\frac{p_0}{p_0 + p_e}\right)^{\frac{1}{\gamma}}\right]. \tag{6}$$

The volume at rest, denoted V_0, should be known. The sound pressure is much smaller than the ambient pressure, and the equation can therefore be linearised using the following power series expansion [8]:

$$(1 + x)^{\alpha} = 1 + \alpha x + \frac{\alpha(\alpha - 1)}{2!} x^2 + \ldots. \tag{7}$$

If only the first two terms are taken into account the volume velocity, in the frequency domain, becomes

$$\widehat{Q} = j\omega \frac{V_0}{\gamma p_0} \hat{p}_e. \tag{8}$$

Thus in the compression chamber's cavity sound pressure is proportional to volume displacement. Recall that the speed of sound is $c^2 = \gamma p_0/\rho_0$ [2]. Substituting γp_0 with $\rho_0 c^2$ and changing sign, considering expansion as positive sign, yields the result of Anthony and Elliot [6]. Introducing a pressure reference $\hat{p}_{\text{ref}}$ in the driver's back cavity yields the source function

$$\Psi = \frac{1}{j\omega} \frac{\rho_0 c^2}{V_0} \frac{\hat{p}_{\text{ref}}}{\hat{p}_e}. \tag{9}$$

The frequency range of this technique is limited by cavity resonances: the limiting frequency should be well below of the first resonance frequency to justify the hypothesis of uniform pressure distribution.

2.2.2. Blocked pipe method

Sound propagating in a cylinder of air with a radius a and length l inside of a rigid pipe at frequencies below the first cut-on frequency can be idealised as forth and back travelling plane waves [1]. The pressure fluctuation $\hat{p}$ inside the pipe is excited by the driver at the termination $x = l$. The particle velocity amplitude $\hat{u}$ at a cross-section x $(0 \leqslant x \leqslant l)$ is

$$\hat{p} = \widehat{A}_+ e^{-jkx} + \widehat{A}_- e^{jkx}, \quad \hat{u} = \frac{1}{\rho_0 c}\left(\widehat{A}_+ e^{-jkx} - \widehat{A}_- e^{jkx}\right). \tag{10}$$

where $\widehat{A}_+$ and $\widehat{A}_-$ are the pressure amplitudes of two waves and k is the wavenumber. The subscript $+$ denotes forth going waves, in the direction of the x-axis and taken to be towards the driver, and $-$ back going waves, in the opposite direction. A non-planar centrally symmetric driver's diaphragm may excite many duct waves. At frequencies below the first cut-on frequency only plane waves will propagate and substantially contribute to volume velocity while other waves will decay. For a circular duct of radius a, the first cut-on frequency is given by $ka < 1.8$ [2].

Let the pipe termination at $x = 0$ be sealed by a rigid surface with a flush mounted microphone embedded in it. The overtones of a closed pipe are given by $f_n = nf_0 = nc/(2l)$ where n is a positive integer. At half-order frequencies equal to $f_{n-0.5} = (n - 0.5)f_0$ the relationship between volume velocity $\widehat{Q}$ at the driver and the blocked pressure $\hat{p}_b$ at the termination can be found from Eq. (10) to be

$$\widehat{Q} = j(-1)^{n-1} \frac{\pi a^2}{\rho_0 c} \hat{p}_b. \tag{11}$$

No assumption regarding the shape or the vibration of the driver's diaphragm has been made in the above. The advantage of this expression is its independence of the distance between the driver and the blocked end. Such a distance is difficult to define if the driver has not a flat diaphragm. The source function for a blocked pressure in a pipe at the half-order frequencies $f_{n-0.5}$ is then

$$\Psi = -j(-1)^{-n+1} \frac{\rho_0 c}{\pi a^2} \frac{\hat{p}_{\text{ref}}}{\hat{p}_b}. \tag{12}$$

3. Experiments

A Morel EM1308 driver was bought off-the-shelf, a small hole was drilled in its back enclosure, and a microphone was fixed inside its back cavity using silicone rubber, Fig. 1. This particular

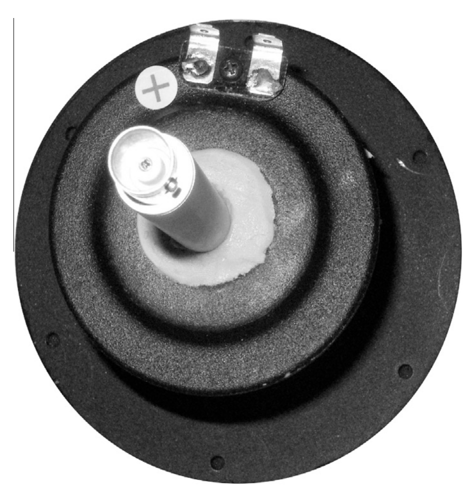

Fig. 1. View of the implemented driver's back-enclosure equipped with a fixated reference microphone.

driver has a convex dome-shaped diaphragm with a radius of 30 mm. The excitation signal was created by a noise generator, filtered through an analog filter, and then amplified. Thus in measurements of transfer functions the driver was fed with a band-pass limited white noise and a frequency estimate was obtained from the recorded signals by use of spectral densities [9]. A schematics of such a setup is shown in Fig. 2. In preceding experiments it was found that the limit of diaphragm displacement makes the driver inefficient below 100 Hz. The driver acts as a simple source up to about 1000 Hz. The frequency range of the analysis has thus been limited to the 100–1000 Hz band.

The volume velocity estimation methods are discussed in Section 3.1. The characterised driver is tested by measuring known transfer impedances in Section 3.2.

3.1. Volume velocity estimation

Both experimental setups were realised by a front-added closed volume of air as shown in Fig. 3. Their practical implementation were however different.

In the first measurement, the compression chamber was made small to allow the use of simple pressure – volume velocity laws. It was assembled by two rectangular aluminium plates of 16 mm thickness clamped together face-to-face. In the first plate there was a ∅1/4 inch hole, precisely adjusted to the microphone used to measure sound pressure in the chamber's cavity. In the second plate acting as a spacer there was a ∅80 mm cylindrical cut-out, an open air volume for fitting the driver. The centred driver was mounted with its diaphragm and rim entering the spacer cut-out. The cavity volume at rest was estimated using granular material of known density required to fill the cavity. Finally, the emptied cavity was sealed by the end plate equipped with a flush-mounted microphone. The assembly was done in such a way that the microphones were aligned to the axis of the driver.

In the second measurement, a blocked pipe was used as a cavity. It was made out of Plexiglas of internal radius 50 mm and internal length 990 mm. The pipe was sealed at one end by a circular Plexiglas plate in which there was a ∅1/4 inch hole, precisely adjusted to the flush-mounted microphone used to measure sound pressure at the blocked end of the pipe. At the other end the centred driver was mounted in such a way that its diaphragm and rim could move freely.

3.1.1. Estimating the source function

The measured source functions are shown in Fig. 4. A fair comparison between the measurement methods would require that the experiments were done in a controlled environment. However, the measurement using a blocked pipe Ψ_{bp} was done one year after the measurement using a compression chamber Ψ_{cc}. This means that the ambiance was not the same in the two measurements, and the time-lag might have changed the driver's mechanical performance. Despite mentioned inconveniences, it was found that the measured source functions are similar except in the real part at the first half-order frequency in the measured 100–1000 Hz range. In this case, it is observed that the real part is a few dB under the measured value in the compression chamber. Note that the measurements agree well in the imaginary part of the source function. The similarities between the measurements, in view of measurement uncertainties such as the estimation of volume at rest of the compression chamber and the time-lag, suggests that the estimation of volume velocity is robust.

The fitted source functions, shown in Fig. 5, tend to agree in the imaginary part. It is seen that the frequency behaviour of the imaginary part is dominated by a compliance law as expected. Nonetheless, the transformation to a lumped element filter is not straightforward. The real part, which should have theoretically been zero, is frequency dependent and shows large mismatch between the two measurements. Due to the lack of a sufficiently long pipe to cover low frequency measurements, the large discrepancy of the real part at low frequencies is due to a poor extrapolation outside the polynomial fit range.

The difference between the two measurements was assessed from the quotient Δ between fitted source functions,

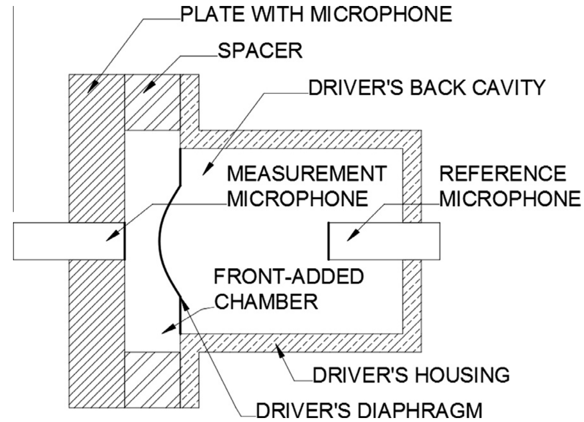

Fig. 3. Schematic setup of source function measurement using a compression chamber. This layout also applies to the experimental setup using a blocked pipe, with the spacer replaced by the pipe.

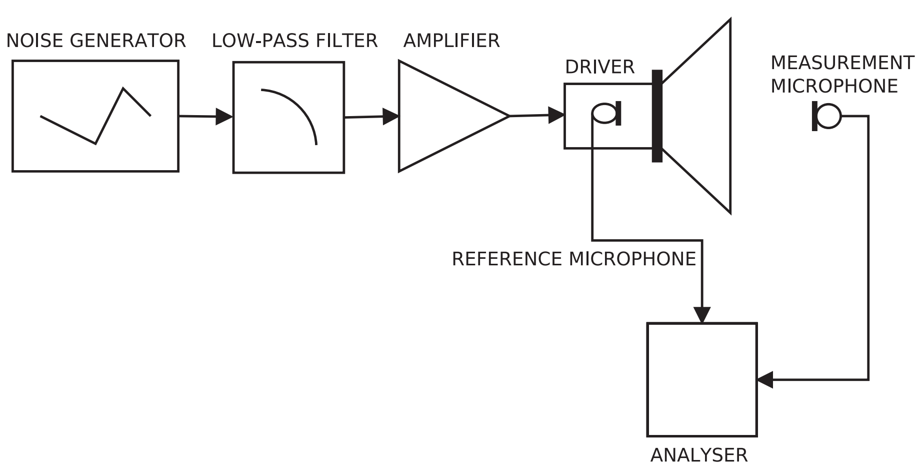

Fig. 2. Schematic setup of transfer function measurements. The driver is fed with band-passed white noise and the microphones signals are recorded by a data acquisition system and thereafter the pressure – pressure relationship can be computed off-line.

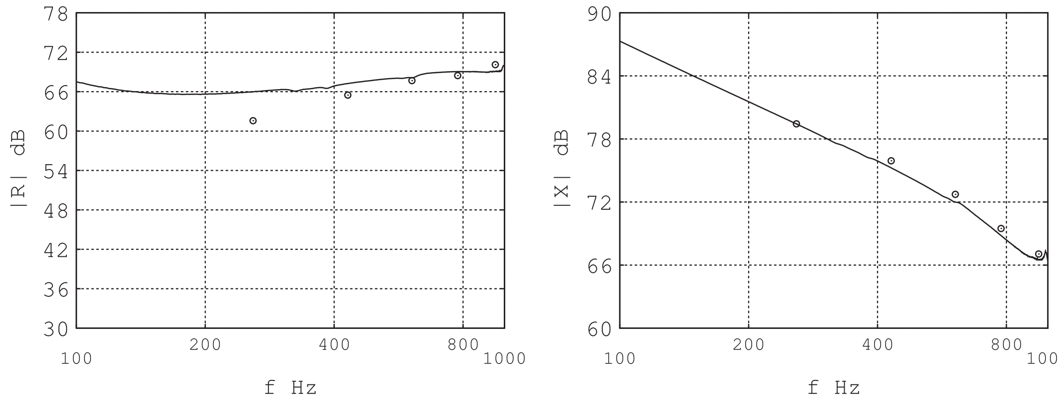

Fig. 4. Measured source function, ψ. Left: real part; right: imaginary part. Continuous line: compression chamber. ∘-marker: blocked pipe.

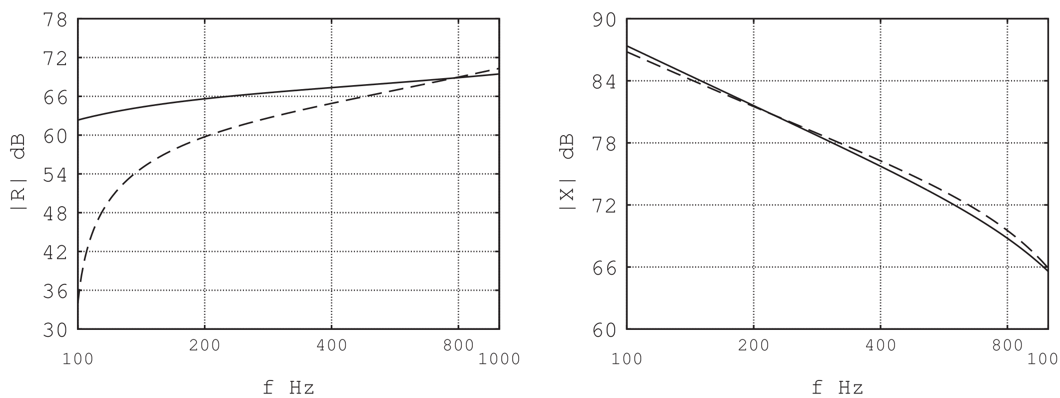

Fig. 5. Fitted source function, $\widetilde{\Psi}$. Left: real part; right: imaginary part. Continuous line: obtained from compression chamber measurement. Dashed line: obtained from blocked pipe measurement.

$\Delta = \widetilde{\Psi}_{bp}/\widetilde{\Psi}_{cc}$. The differences in level and in phase are shown in Fig. 6. The differences are small, less than 1 dB and 0.1 rad, and either of the two characterisation method looks well adapted to the measurement of transfer impedances. The level difference in the real part R at low frequencies in Fig. 5 does not influence the overall difference in level because the imaginary part X of the fitted source functions dominates as expected $R \ll X, |\Delta| \approx |X_{bp}|/|X_{cc}|$.

3.2. Measurement of transfer impedances

The characterisation should not depend on the external loading of the driver. The two characterisation methods satisfy this requirement in principle. One way of checking the accuracy of the obtained source function is to measure transfer impedances in rather dissimilar spaces. The sound pressure in two such spaces was investigated; namely, inside of a closed pipe and in an anechoic room.

3.2.1. Driver set in a closed rigid pipe

The pipe previously used for estimating the source function was also used for measuring transfer impedance. A schematics of the experimental setup is seen in Fig. 7. This setup was modelled as a rigid piston in a pipe assuming travelling plane waves in analogy to Section 2.2.2. The transfer impedance between the rigid piston

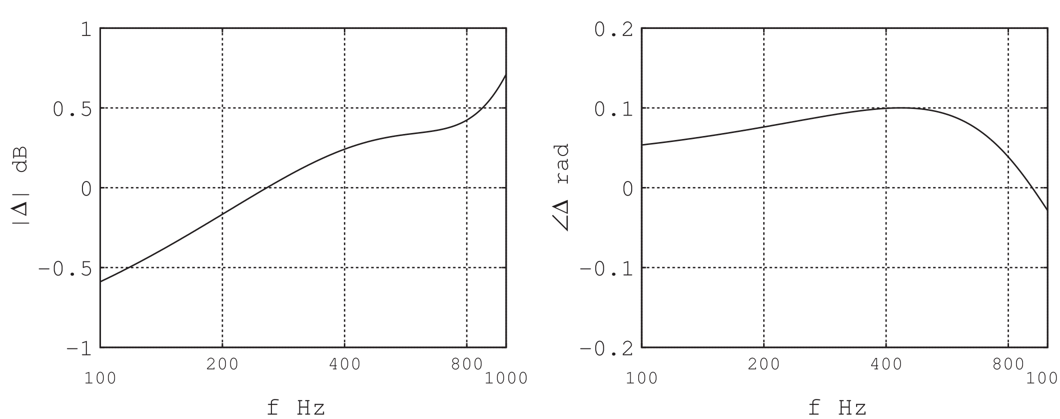

Fig. 6. The quotient between the obtained source function using the blocked pipe method and the compression chamber method. Left: level; right: phase.

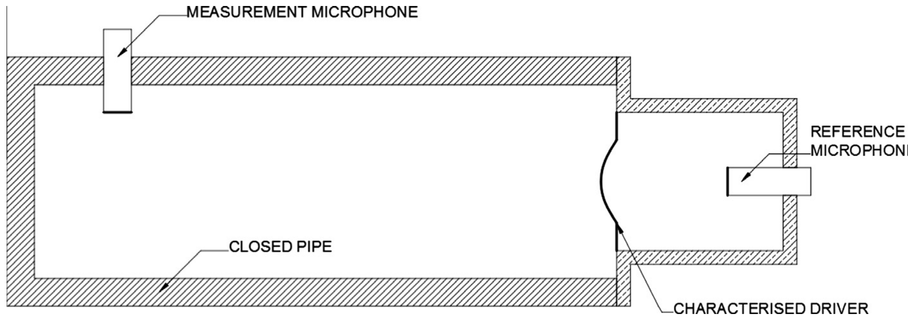

Fig. 7. Schematic setup of space function measurement in a closed pipe.

at $x = l$, taking positive velocity when the piston is moving into the pipe, and a pickup at x, due to forth and back travelling plane waves, is given by

$$Z = -j\frac{\rho_0 c}{\pi a^2}\frac{\cos kx}{\sin kl}. \tag{13}$$

This can be deduced from Eq. (10) by setting $u(x=0)=0$ and $u(x=l) = -1/(\pi a^2)$. The assumption of a flat circular disk is not realistic due to the convex dome on the real driver, therefore the plane of the rigid piston is not well defined in the model. Furthermore the equivalent length of the pipe may be frequency-dependent as a result of non-planar excitation. Damping is introduced in the model by a complex speed of sound $c' = c(1 + j\eta/2)$ where the loss-factor η is a frequency dependent parameter [10]. For simplicity, the measured transfer impedance is compared to a fictitious pipe of 980 mm length and 4% loss-factor. The pickup was at 330 mm. The speed of sound is taken to be 343 ms^{-1} and the density of air 1.2 kgm^{-3}.

The modelled transfer impedance and its measured counterpart are shown in Fig. 8. The agreement in magnitude, away from the resonances, and in phase is quite satisfactory considering the inconveniences in the modelling. The frequency shift of the fundamental tone may be due to the existence of non plane waves inside the pipe. The first few overtones are correctly positioned, but the magnitude of the modelled transfer impedance is not quite correct due to unknown loss-factor. The matching confirms that the characterisation is consistent with measurement and that the driver can be used to measure transfer impedances.

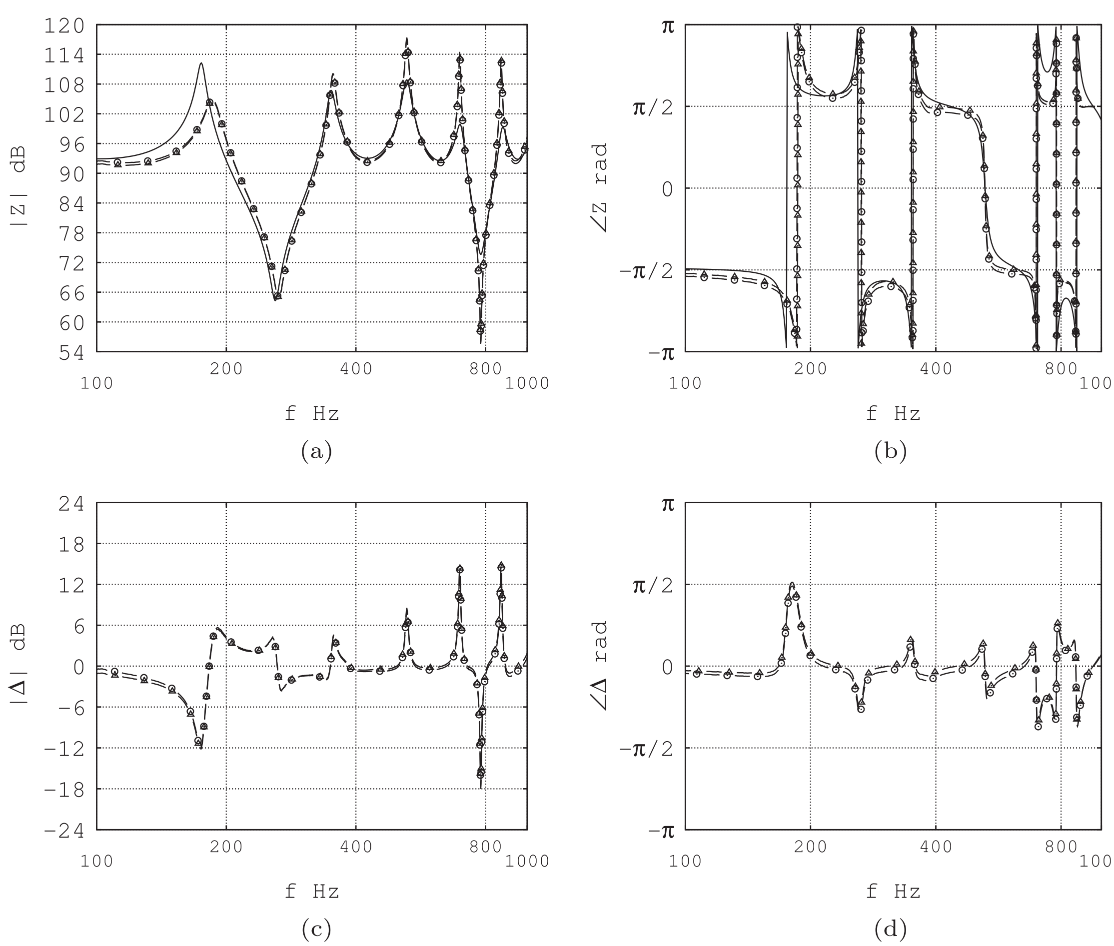

Fig. 8. Sample transfer impedance measured in a blocked pipe: (a) level and (b) phase. Modelled: continuous line; measured using a compression chamber, ○-marker, or a blocked pipe, △-marker. The quotient between measured and modelled estimates: (c) level and (d) phase.

3.2.2. Driver set in a rectangular flat baffle in an anechoic room

The driver was set in a large flat rectangular baffle 1350 × 1650 mm, made out of medium-density fibreboard and placed roughly in the centre of a large anechoic room. The driver was slightly offset with respect to the centre of the baffle. The experimental setup is seen in Fig. 9 and can be modelled at lower frequencies as a rigid piston in an infinite baffle. The transfer impedance on the axis of a rigid piston is [1]:

$$Z = -\frac{\rho_0 c}{\pi a^2}\left(e^{-jk\sqrt{z^2+a^2}} - e^{-jkz}\right), \qquad (14)$$

here a is the piston's radius and z is the distance from the piston's centre to the field point. This model is not fully realistic because it neglects edge effects of the finite baffle, which should be negligible only at wavelengths much smaller than the baffle size. Another issue with the simple model is that the convex dome-shaped diaphragm on the physical driver should not have the same near-field behaviour as a rigid piston, and the acoustic centre of the dome is moved towards the microphone pickup.

An example of a transfer impedance is shown in Fig. 10. The response was measured 50 mm from the baffle. The measured

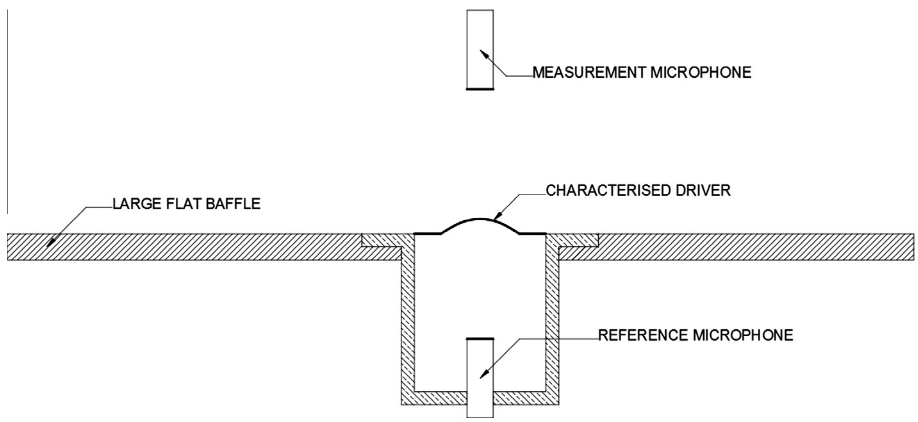

Fig. 9. Schematic setup of space function measurement in an anechoic room.

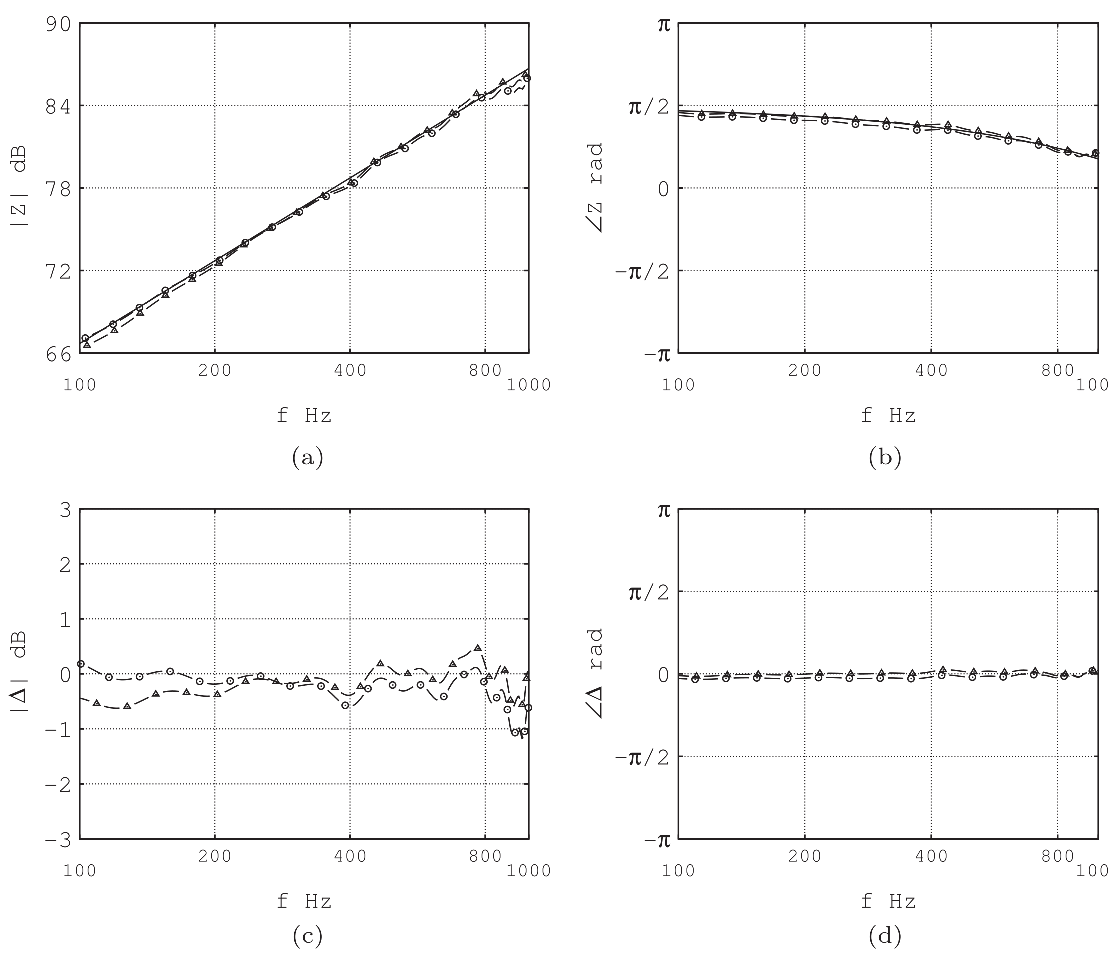

Fig. 10. Sample transfer impedance measured in an anechoic room: (a) level and (b) phase. Modelled: continuous line; measured using a compression chamber, ○-marker, or a blocked pipe, △-marker. The difference between measured and modelled estimates: (c) level and (d) phase.

transfer impedance follows the computed one of a rigid piston. The agreement in magnitude and phase between measured and modelled transfer impedances looks satisfactory. It shows that the driver can be used to measure transfer impedances, and that the characterisation is reasonably invariant to external loading. This also suggests that the two calibration methods are comparable with free-space calibration.

In this particular experiment the measured space function contained small amplitude fluctuations with frequency. The behaviour is perhaps due to scattering from alien objects or imperfect baffle diffraction. The measured space function has been smoothed by a polynomial fit prior to estimating the transfer impedance. This has reduced the fluctuations while still representing the correct tendencies in the raw data.

4. Discussion

The authors were looking for a practical measurement technique of transfer impedances using an ordinary driver. None out of several designs of known volume velocity sources found in the literature [3–5] has proved to be fully adequate for this task. The volume velocity of a driver can not be measured directly, and each design relies on a transducer producing an output which can be related to volume velocity. Such a transducer can be e.g. an additional measuring voice-coil or an accelerometer [4,5]. The disadvantage of such transducers is that they make the assumption of rigidity which is not met in practice. Another design is to equip a driver with an internal microphone [6]. The microphone should theoretically produce a signal proportional to volume displacement of the driver's diaphragm. This cannot be achieved in an ordinary back-cavity design because of the presence of cavity damping and resonance.

Unlike in [6], the methods developed in this work require the internal microphone to be calibrated experimentally. It has been proposed in [6] that calibration can be carried out by summing up individual contributions of surface patches. Apart from requiring specific equipment, the disadvantage of such calibration is that the surface patches are considered to be in rigid motion and are quantified by a single point velocity measurement. This calibration technique was found to be cumbersome in the case of a curved surface [7]. An alternative as done in [5] is free-space calibration. The advantage is that this technique is based on a simple law. The disadvantage of such an approach is that it requires access to an anechoic room. Furthermore it has been found that free-space calibration may suffer from radiation by the driver's back enclosure [7]. This can be shielded off by use of a baffle but risks to create unwanted diffraction phenomena.

The advantage of the two methods developed are that neither requires any specific equipment nor access to specific laboratory space. Perhaps the biggest advantage is that no assumption about the vibrating surface is needed, which is a property shared with the free-space calibration. The disadvantage of the compression chamber method is that in order to go up in frequency a very small calibration chamber is required. This makes the method sensitive to precise estimation of the calibration volume. The measurement using the blocked pipe method can be done only at discrete frequencies and thus requires interpolation. In order to reduce the frequency spacing and decrease the lower frequency limit of measurement a long pipe is necessary which may be a disadvantage. Thus, the two methods are complementary in the frequency domain.

Despite mentioned inconveniences the differences between the two calibration methods was found to be small. The identified source functions were applied in an anechoic room with satisfactory matching to modelled transfer impedances. This suggest that the two methods can be used as an alternative to free-space calibration.

5. Conclusions

Two calibration methods of a small volume velocity source have been developed. The source can be any small driver equipped with a sealed back cavity into which a microphone is installed. The calibration consists in finding the relationship between the sound pressure in the back cavity, measured by the microphone, and the volume velocity of the source produced by the driver's diaphragm. This relationship, which is theoretically given by a $1/f$ law for a small loss-less cavity, was found to deviate from it thus requiring an independent measurement of volume velocity. The latter is done in front of the diaphragm in an external cavity of rigid walls which allows obtaining the volume velocity from simple sound pressure measurements. Thus the entire calibration procedure consists in simultaneous measurement of two sound pressure signals. The external cavity used is either a small compression chamber of known volume or a long blocked tube. In both cases the relationship between the volume velocity and the measured sound pressure has been formulated in such a way to avoid dependence on either the geometry or the velocity distribution of the driver's diaphragm. Calibration by the two methods produced similar results. The volume velocity, once obtained by calibration, was then employed to compute transfer impedances in a closed pipe and in an anechoic room. The matching between the computed and measured transfer impedances was found satisfactory, suggesting that the two calibration methods are sufficiently accurate for use in engineering applications.

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