Reduction of bass-reflex port nonlinearities by optimizing the port geometry

N.B. Roozen
Philips Centre for Manufacturing Technology
P.O.Box 218, 5600MD Eindhoven, the Netherlands
e-mail: n.b.roozen@philips.com

Bass-reflex ports are used to enhance the bass reproduction of loudspeakers. However, at high output levels blowing sounds and acoustic losses can occur, especially in the case of small loudspeaker cabinets which necessarily use relatively narrow bass-reflex ports.
The blowing sounds and acoustic losses are caused by an unsteady separation of the acoustic flow, resulting in the generation of vortices in the port. These vortices have been visualized by means of a Schlieren method.
It was found that the blowing noises are dominated by discrete frequencies corresponding to acoustic standing waves in the longitudinal direction of the port. These port resonances are impulsively excited by the vortices which are created due to the periodic separation of the acoustic flow.
The vortices, and thus also the blowing noises and the associated acoustic losses, can be reduced significantly by using a port contour geometry that slowly diverges from the center towards both port ends and is rounded with curvature radii that are not too large at both port terminations.
Another way to reduce the unwanted blowing noises is to dampen down the acoustic standing waves in the longitudinal direction of the port. This can be achieved by perforating the port at the positions where the acoustic standing waves originally have a maximum amplitude and wrapping foam around the port at the positions which are perforated.

0 Introduction

In the case of so-called bass-reflex boxes or vented loudspeaker boxes [1],[2], a port is applied. The port is an open pipe connecting the volume of air enclosed within the box with its surrounding environment. The port contributes to the sound reproduction in the lowest part of the frequency range around the Helmholtz frequency of the port-box system. Without a port, the box volume would have to be approximately a factor of two larger to achieve the same low frequency acoustical performance. The typical improvement that can be obtained with a port is shown in figure 1.

However, a bass-reflex port can also cause blowing sounds to be generated. The blowing sounds are generated by aero-dynamic phenomena which occur at the port terminations. Especially small loudspeaker cabinets suffer from these unwanted blowing noises because of the relatively narrow bass-reflex ports which are usually applied in those loudspeaker boxes.

In the literature a number of studies have been reported on this topic. It is generally recognized that a larger port cross-sectional area is better than a smaller cross-section having the same Helmholtz frequency, as the acoustic flow amplitude is smaller. It is also generally recognized that edges should be smooth and discontinuities in the port contour should be avoided.

Backmann performed an extensive study on a number of ports, such as ports with different types of flanges and bends [3]. He concluded that both the inner and outer port ends should be designed to reduce the blowing sounds. Rounding only one port end reduces these blowing sounds slightly; therefore both port ends need to be rounded for maximum reduction in blowing sound.

The approach which was chosen to reduce the blowing sounds by the authors of this paper was to increase the cross-sectional area of the port towards the port ends, thereby reducing the level of vorticity, while maintaining a certain acoustic mass m_a and the same physical port length L. These conflicting requirements can be partly fulfilled by making use of tapered or converging-diverging ports [4], [5], [6]. It is essential to widen the port towards both port ends, as the acoustic flow oscillates from inwards to outwards and vice versa.

Numerical simulations and experimental investigations are presented. It is shown that the blowing sounds can be attributed to the unsteady separation of the acoustic flow at the port end. It was found that this separation results in an impulsive excitation of the $\lambda/2$-mode in the longitudinal direction of the port and its harmonics, where λ denotes the acoustic wavelength. This opens the road to quite a different way to reduce the unwanted blowing noises, i.e. to dampen down the acoustic standing waves in the longitudinal direction of the port to occur by perforating the port and wrapping foam around the port at the positions which are perforated [7].

The remainder of this paper is organized as follows. In section 1 it is made clear that small loudspeaker cabinets require narrow bass-reflex ports. The dimensionless numbers pertinent to flow separation and boundary layer turbulence are discussed in section 2. In section 3 the visualization of the vortices at the end of a loudspeaker port are discussed. The acoustic port resonances due to the vortex shedding are discussed in section 4. The effect of the port geometry on the generation of blowing sounds are discussed in section 5, while in section 6 the effect of port perforations on the acoustic port resonance will be discussed. Finally, the conclusions are drawn in section 7.

1 Bass-reflex port applied to small boxes

Application of a bass-reflex port to small loudspeaker boxes implies that the cross-sectional port radius has to be relatively small in order to achieve a sufficiently low Helmholtz frequency.

This can easily be seen from the expression for the Helmholtz frequency (see [1] or [2]):

$$f_{Helmholtz} = \frac{c}{2\pi}\sqrt{\frac{S_0}{L_{eff}V}} = \frac{c}{2\pi}\sqrt{\frac{\rho_0}{m_a V}} , \tag{1}$$

where c is the speed of sound, ρ_0 is the mean density of air, S_0 is the reference cross-sectional area of the port, V is the volume of the box, L_{eff} is the effective port length defined as

$$L_{eff} = \int_0^L \frac{S_0}{S(x)} dx , \tag{2}$$

and m_a is the acoustic mass of the port defined as

$$m_a = \int_0^L \frac{\rho_0}{S(x)} dx . \tag{3}$$

In these equations L is the physical length of the port, x is the axial coordinate along the port axis and $S(x)$ is the cross-sectional area of the port at axial coordinate x. In the calculation of the effective length L_{eff} and the acoustic mass m_a of the port, end corrections should be *additionally* accounted for. Appropriate end corrections can be found in the literature [1], [2], [8].

A smaller box volume V forces the port cross-sectional area S_0 to become smaller for a given physical port length L and a given Helmholtz frequency $f_{Helmholtz}$. This means that for small boxes, with a relatively small cross-sectional area S_0, the acoustic flow velocity in the port is relatively high. The acoustic flow velocity amplitudes can easily amount 10 m/s or more (see table 1). As a result of these high air velocities, bass-reflex ports applied to small loudspeaker boxes often lead to blowing sounds due to airflow nonlinearities.

Table 1: Required acoustic flow velocity for a cylindrical port with specifications as given in the table, fluid density of air taken to be 1.2 kg/m^3.

Cross-sectional radius	1 cm,
Helmholtz frequency	50 Hz,
sound pressure level in free field at 1 m distance	70 dB.
Required volume flow (RMS)	2.3 l/s,
required acoustic flow velocity amplitude	10 m/s.

2 Dimensionless parameters

Two mechanisms can be hold responsible for the production of the blowing sounds: boundary layer turbulence and unsteady separation of the acoustic flow at the port termination.

The dimensionless parameters which determine the occurrence of these mechanisms are the Reynolds number Re_δ, based on the boundary layer thickness δ, and the Strouhal number St.

The Reynolds number Re_δ based on the boundary layer thickness δ is defined as (see [9], [10] or [11])

$$Re_\delta = \frac{u\delta}{\nu} \ , \tag{4}$$

where the boundary layer thickness $\delta = \sqrt{(2\nu/\omega)}$. Here ν is the kinematic viscosity, ω is the angular frequency and u is the acoustic flow velocity.

The critical Reynolds number Re_δ for a boundary layer on a flat plate equals $Re_{\delta\_crit} = 520$ [9]. For Reynolds numbers larger than 520, the boundary layer will become turbulent. Merkli [10] gives critical Reynolds numbers Re_δ which vary between 500 and 1000. In table 2 the critical velocity of the acoustic flow in the port and the critical sound pressure level at a distance of 1 m in free field for occurrence of boundary layer turbulence is given for a specific situation.

Table 2: Critical velocity and sound pressure level for a bass-reflex port with a cross-sectional port radius a=1 cm that is tuned to a Helmholtz frequency of 45 Hz, the kinematic viscosity of air taken to be $\nu = 15 \cdot 10^{-6}$ m^2/s.

	Critical velocity of the acoustic flow	Critical SPL at 1 m in free field
Boundary layer turbulence	25 m/s	78 dB
Unsteady flow separation	3 m/s	60 dB

The Strouhal number is a measure for the ratio of the acceleration caused by the unsteadiness of the flow and the convective acceleration caused by the non-uniformity of the flow at the port termination. It is defined as the ratio of the cross-sectional port radius a and the particle displacement u/ω:

$$St = \omega a/u. \tag{5}$$

If the particle displacement u/ω approaches the order of the port radius a, i.e. if St is of the order 1 or smaller, then there will be an unsteady separation of the acoustic flow at the port end, vortices associated with the separation of the flow will form, and jets will form (see [12]).

In figure 4, the physical phenomena occurring during inflow and outflow are sketched for low Strouhal numbers. Jet formation occurs at outflow, while a sink-like flow takes place at inflow (for ports with rounded ends).

In table 2 the critical velocity of the acoustic flow in the port and the critical sound pressure level at a distance of 1 m in free field for occurrence of unsteady flow separation of the acoustic flow at the port end is given for a specific situation.

For small bass-reflex ports (port cross-sectional radius a=1 cm) it can be concluded that unsteady separation of the acoustic flow and the associated free jet formation will be rather common. Broadband jet turbulence or boundary layer turbulence is expected to become relevant only at very high pressure levels.

3 Visualization by means of the Schlieren method

The unsteady separation of the acoustic air flow was visualized at high air speeds by means of the Schlieren method [13]. The port used for this experiment was a cylindrical port with rounded edges. Carbon-dioxide gas was injected through porous walls made from sinter bronze around the port exit. With the aid of a stroboscope it was possible to take pictures at fixed moments of the oscillating period. A sketch of the test set-up is shown in figure 2. The results are presented in figure 3.

As can be seen in figure 3, the vortex shedding takes place at the transitional area of the air flowing out of the port and the stagnant surrounding air. The mushroom-shaped vortices are produced each time the air flows out of the port. When the loudspeaker is driven at a frequency of for instance 45 Hz, a vortex will be generated 45 times per second at one end of the port and 45 times per second at the other end of the port. It can be seen in figure 3 that a vortex is generated during outflow and that this vortex moves away from the port. These mushroom-shaped vortices give rise to blowing sounds.

4 Port resonances

The blowing sounds consist of a number of noise types. Besides broadband noise generation by the vortices, it was found that the periodic vortex shedding gives rise to a periodic excitation of acoustic port resonances. The acoustic port resonances are excited impulsively each time a vortex is being generated.

The first acoustic resonance corresponds to the $\frac{1}{2}\lambda$ acoustic wave in the longitudinal direction of the port. The resonance frequency f_1 of this fundamental acoustic mode and its harmonics ($f_2, f_3, ...$) can easily be calculated with the formula

$$f_n \approx \frac{nc}{2L_{eff}} \ . \tag{6}$$

where n is an integer (n=1,2,3,...). Measurements on a cylindrical bass-reflex port with a physical length L=0.13 m and a cross-sectional radius a=0.01 m are shown in figure 5. The resonance frequency f_1 for this port appears to be about 1100 Hz, which agrees well with equation 6, with due allowance for the port's end effects. The harmonics of the $\lambda/2$ resonance were also excited, but less strongly than the fundamental bass-reflex port mode at 1100 Hz.

Although the loudspeaker was driven by an electrical signal with a frequency of 45 Hz only, harmonics of 45 Hz were also present. These harmonics, which are all at least 20 dB below the 45 Hz signal, are caused by harmonic distortion of the loudspeaker itself. This was confirmed by separate measurements of the loudspeaker response.

The resonance peak at 1100 Hz is rather wide, which indicates that the resonance is strongly damped. At low velocity amplitudes the damping of the 1100 Hz resonance is mainly due to acoustic radiation into open space. Damping increases at higher amplitudes because acoustic energy is dissipated by convective effects [5].

To illustrate the periodic nature of the $\lambda/2$ port resonance excitation, the blowing sounds of a bass-reflex system driven at 40 Hz were measured in the time domain. The sound pressure signal was filtered using a high-pass filter. Below 500 Hz the filter reduces the level of the signal by more than 30 dB. The sound pressure was measured at a distance of 1 m from the port in a free-field condition (in an anechoic room) for both a port with sharp edges and a gradually converging-diverging port. No time-averaging was employed. The results are shown in figure 9.

The high-frequency components of the sound radiated from the port with sharp ends as shown in figure 9 clearly demonstrates the pulsatile behavior of the port. Each time the acoustic resonance is excited, the high-frequency (1 kHz) component increases sharply. The frequency with which the acoustic resonance is excited equals twice the driving frequency of 45 Hz. After this excitation an exponential decay is observed. Note that a pulsatile excitation takes place both during flow of air out of the box and during flow of air into the box. Thus, it can be concluded that the high-frequency blowing sounds have a deterministic character rather than a stochastic character.

5 Port contour geometry

Experimental investigations were performed for the ports as shown in figure 6. Additionally, a cylindrical port with sharp edges was examined. This port will be denoted by "port 0". For a fair comparison between the ports, both the physical length of the ports and the acoustic mass m_a of the ports are kept the same. The latter means that the Helmholtz resonances of the box-port combinations are the same. In order to obtain ports with the same acoustic mass m_a for a given contour shape and a fixed port length, the smallest diameter of the port is varied, the acoustic mass of the port is calculated with a dedicated software program, and the required smallest diameter of the port is calculated iteratively. Therefore a port which has a rather large outer diameter (such as the parabolic shaped port, Port D, see figure 6) will have a rather small diameter at its center in order to get the same acoustic mass.

In a reverberant room of 227 m^3 the blowing sounds which are produced by the ports were measured. A bandpass loudspeaker system was used for this purpose, as shown in figure 7. With this measurement set-up, the sounds which are produced by the port only can be measured well. The loudspeaker cannot easily radiate sound directly into the measurement room.

The amount of blowing sound was measured in the frequency range between 100 Hz and 10 kHz for different driving frequencies and different levels. The sound power levels correspond to 85, 90 and 95 dB

The A-weighted sound power of the blowing sounds (including the harmonic distortion) for driving levels corresponding to a sound power level at the driving frequency (45 Hz) of 85, 90 and 95 dB are shown in figure 8.

The amount of blowing sounds which are produced by the ports appears to be strongly dependent upon the level of excitation. Obviously, the amount of blowing sounds increases at higher levels. More subtle, however, is that at low levels a cylindrical port with a large curvature radius produces less blowing sound than a cylindrical port with a small curvature radius, while at higher levels it is the other way around.

Let us consider the cylindrical ports first. Port 0 is a cylindrical port with a sharp end and ports A, B, and C are cylindrical ports with a curvature radius of R=5 mm, R=10 mm and R=20 mm, respectively. From figure 8 it can be seen that at low levels (L_W = 85 dB at 45 Hz) a port with a large curvature radius (port C, for instance) produces less blowing sound, while at high levels (L_W = 95 dB at 45 Hz) a port with a small curvature radius (port A or even the port with a sharp edge, port 0) produces less blowing sound. At other driving frequencies the same tendency was found.

At low levels (L_W = 85 dB at 45 Hz) a port with a larger curvature radius produces less blowing sound because the flow separates at a larger port radius, resulting in a less intense vortex. In order to understand what is happening at high velocities, it must be understood that the production of blowing sounds depends not only upon the vortex intensity but also upon the quality factor of the port, i.e. the amount of damping of the port resonance. For low velocity amplitudes the quality factor of the port is determined by the amount of acoustic radiation (which in turn is determined by the acoustic impedance of the port). At low velocity amplitudes, the quality factor is approximately the same for all ports. At higher levels energy is transfered from the acoustic oscillation of the port (at about 1 kHz) to the free jet formed during the vortex shedding at the driving frequency, which reduces the quality factor significantly. This transfer of energy is strongest when the acoustic oscillation can interact with the free jet. In first order approximation the energy losses are proportional to the flow velocity on the jet [12].

It appears that this type of interaction is strongly reduced for ports with large curvature radii. In some cases one can even find a net energy production (whistling) rather than absorption [12]. At high air velocities, the quality factor of a port with a small curvature radius or even a port with sharp ends is significantly smaller than the quality factor of a port with a large curvature radius. Additional measurements confirmed a reduction of the quality factor at higher driving levels. Apparently, at high air velocities a cylindrical port with a large curvature radius produces more blowing sounds than a cylindrical port with a small curvature radius, even if the source of sound is weaker. This indicates that the increase in quality factor with curvature radius compensates the reduction of sound source and radiation efficiency at high amplitudes.

Vanderkooy [15] also found that a cylindrical port with a large curvature radius (curvature radius R about twice the port cross-sectional radius a) can produce more blowing sounds than a cylindrical port with sharp port ends. He explained this behavior by reasoning that there are less losses at the edges for a port with a large curvature radius. A better explanation, however, is that the damping of the $\lambda/2$ port resonance is strongly increased for a port with sharp edges as a result of a dissipation of acoustic energy by convective effects.

Now that the effect of different curvature radii for cylindrical ports has been discussed, the effect of a non-cylindrical port geometry, such as that of port D and port E, will be considered. It can be seen from figure 8 that port D, which is strongly parabolic, produces high levels of blowing sounds. This is because for such a port contour the vortices are created *inside* the port, as was concluded from numerical simulations of the acoustic flow in the port [14]. This is highly undesirable because in such a situation the vortices will accumulate inside the port, resulting in strong broadband turbulent sound production.

By slowly converging and diverging the port geometry, like the geometry of port E, the flow is able to adhere to the port wall. This prevents separation *inside* the port, while at the same time the cross-sectional area of the port towards the port ends is increased compared to a cylindrical port with the same acoustic mass m_a. The increased cross-sectional area reduces the local air velocity and therefore also reduces the vortex intensity and the associated production of blowing sounds. Furthermore, also at high acoustic levels port E produces less blowing sound compared to the other ports.

As can be seen from figure 8, the blowing sounds of port E are the lowest of all ports for all levels tested. Compared to a cylindrical port with the same curvature radius (port A), port E produces 1 dB, 4 dB and 5.5 dB less blowing sound for a sound power level of 85, 90 and 95 dB, respectively.

6 Port perforations

In section 4 it was shown that the unsteady separation of the acoustic flow at the port end results in an impulsive excitation of the $\lambda/2$-mode in the longitudinal direction of the port and its harmonics. From figure 5 it can be seen that the blowing noises are dominated by the acoustic port resonances. So, a significant reduction of the perceived blowing sounds can be expected if these resonances are dampened down. Dampening down the acoustic resonances can be realized effectively by perforating the port at positions corresponding to the maxima of the acoustic standing waves and wrapping foam around the port. The purpose of the foam around the port, which is in contact with the inside of the port via the perforations, is to absorb the acoustic energy of the port resonances. The fundamental acoustic resonance (a $\lambda/2$-resonance) can effectively be dampened down by perforating the port at $x = L/2$, where x denotes the longitudinal direction of the port, L denotes the physical length of the port, and the inlet and outlet sides the port are located at $x = 0$ and at $x = L$, respectively. The first harmonic (a λ-resonance) can be dampened down effectively by perforating the port at $x = L/4$ and/or at $x = 3L/4$.

Measurements have been performed on a cylindrical port with sharp ends ("port 0") with and without perforations. The port was wrapped with foam, covering the port perforations on the outer side of the port, as shown in figure 10. The measurement results are shown in figure 11 for a cylindrical sharp ended port without perforations, with perforations only at $x = L/2$, and with perforations at $x = L/2$ and at $x = 3L/4$. Perforating the port at $x = L/2$ and wrapping foam around the port reduces the acoustic port resonance at about 1100 Hz with 20 dB ! The acoustic port resonance at about 2200 Hz is also somewhat reduced, but not very much because the pressure variations of this acoustic resonance vanish at $x = L/2$, which makes the perforation ineffective. The overall reduction of the blowing noises is 15 dB.

Perforating the port at $x = L/2$ and at $x = 3L/4$ reduces the resonance at 1100 Hz with 15 dB and the resonance at 2200 Hz also with 15 dB [7]. The overall reduction of the blowing noises is 16 dB.

From additional measurements it was found that the port perforations did not have an adverse effect on the sound reproduction at low frequencies (i.e. at the Helmholtz frequency of the port).

Perforating the port and wrapping foam around the port can very well be combined with the application of a more optimal port geometry as discussed in the previous section. In fact, the disadvantage of some ports in that the quality factor is relatively high compared to a sharp ended port at high driving levels can be elevated by perforating the port. Thus, the big advantage of a converging-diverging port geometry at low driving levels can be enhanced at higher driving levels by perforating the port and wrapping foam around the port.

7 Conclusions

It was found that there are two kinds of blowing sounds which can be attributed to the unsteady separation of the acoustic flow at the port end, i.e. broadband noise and acoustic resonances due to standing waves in the longitudinal direction of the port. These acoustic resonances are impulsively excited by the periodic separation of the acoustic flow.

The blowing sounds can be reduced by designing the port geometry such that the strength of the vortices caused by flow separation and jet forming at the end of the port is minimal. Reducing the vortex strengths reduces the production of blowing sounds, causes less acoustic energy to be dissipated by the vortices and therefore also improves the reproduction of the primary noises.

From experimental investigations it was found that not only the vortex strength and vortex location are important, but also the sound radiation efficiency and the quality factor of the $\lambda/2$ port resonance (and its harmonics). It appears that for a sharp edged port as well as for a port with a small curvature radius the quality factor of the $\lambda/2$ port resonance drops significantly with increasing driving levels. The quality factor of the port becomes smaller as a result of the transfer of energy from the acoustic oscillation of the port to the free jet formed during the vortex shedding at the driving frequency. For this reason, both a sharp edged cylindrical port and a cylindrical port with a small curvature radius produce less blowing sound at higher driving levels than a cylindrical port with a large curvature radius. The curvature radius should therefore not be made too large.

A port contour which combines a reduction of the vortex strength with a low quality factor at high driving levels is a port contour which slowly diverges towards both port ends with a maximum angle of 6 degrees (measured from port contour to port axis) and which is rounded with relatively small curvature radii at both port ends (see figure 6, port E). Compared to a cylindrical port with the same curvature radius, this "optimal" port produces 1 dB, 4 dB and 5.5 dB less blowing sound for a sound power level of 85, 90 and 95 dB, respectively.

Another way to reduce the blowing noises is to dampen down the acoustic resonances due to standing waves in the longitudinal direction of the port. Dampening down the acoustic resonances can be realised effectively by perforating the port at positions corresponding to the maxima of the acoustic standing waves. The blowing noises are reduced by 15 dB by perforating the port in the middle and wrapping foam around the port. However, perforating the port in the middle reduces the 1100 Hz resonance only. Perforating the port in the middle and at three quarter of its length affects both the acoustic resonance at 1100 Hz and at 2200 Hz, and reduces the overall acoustic power of the blowing noises by 16 dB.

Perforating the port and wrapping foam around the port can very well be combined with the application of a more optimal port geometry. In fact, the disadvantage of some ports in that the quality factor is relatively high compared to a sharp ended port at high driving levels can be elevated by perforating the port. Thus, the big advantage of a converging-diverging port geometry at low driving levels can be enhanced at higher driving levels by perforating the port and wrapping foam around the port.

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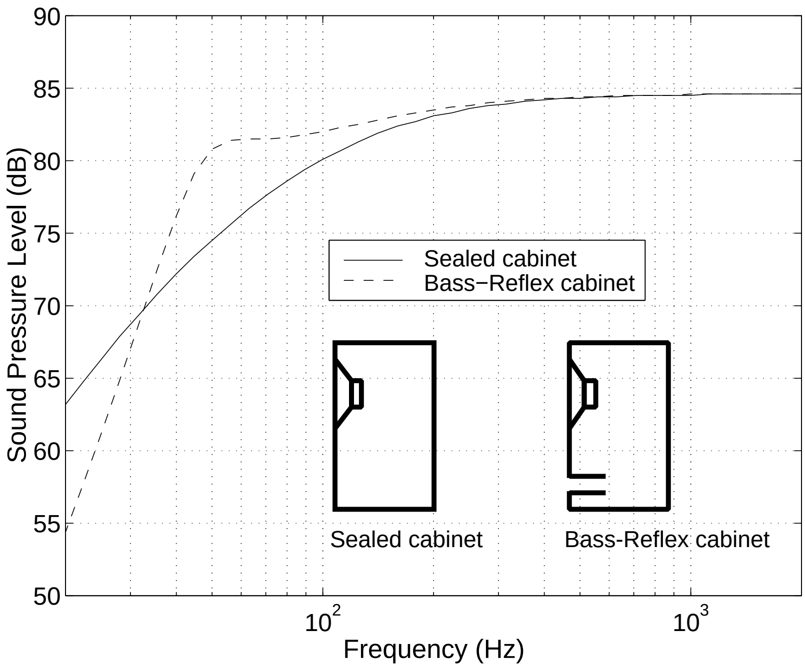


Figure 1: *A sealed cabinet versus a bass-reflex cabinet.*

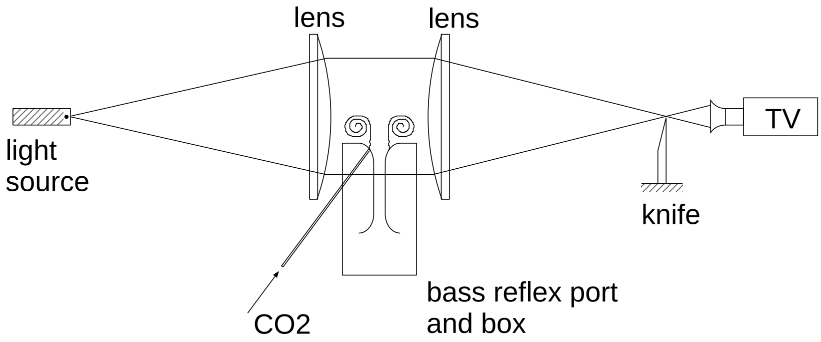
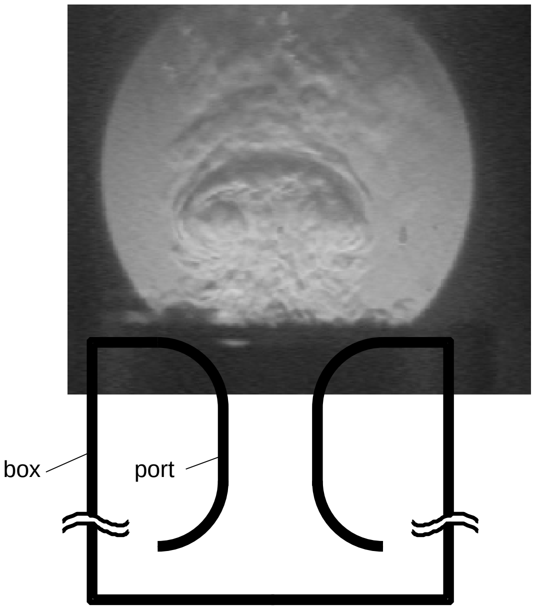
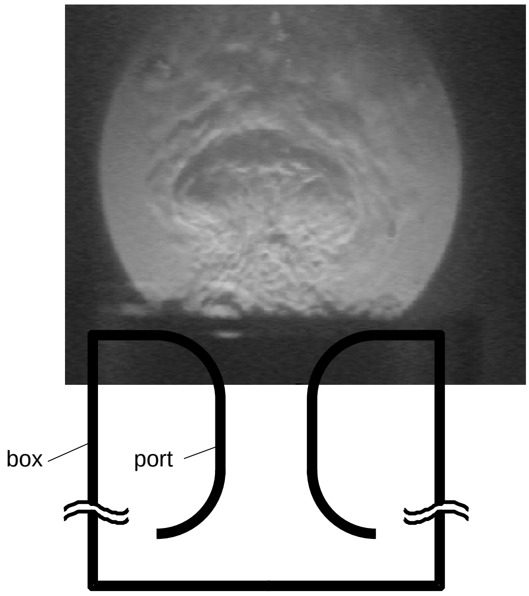


Figure 2: *Visualization test set-up.*



(a)



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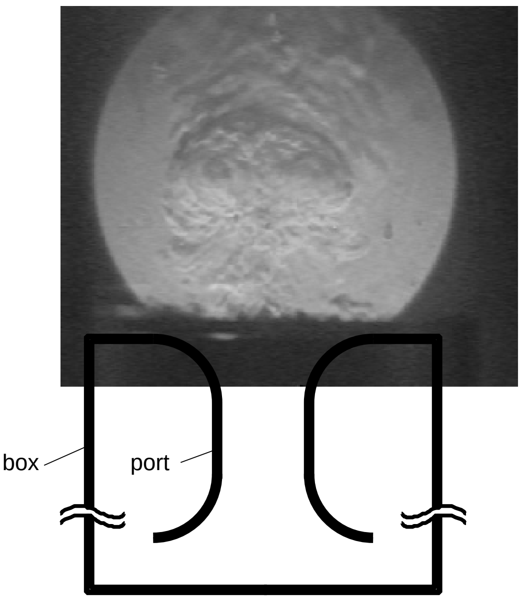

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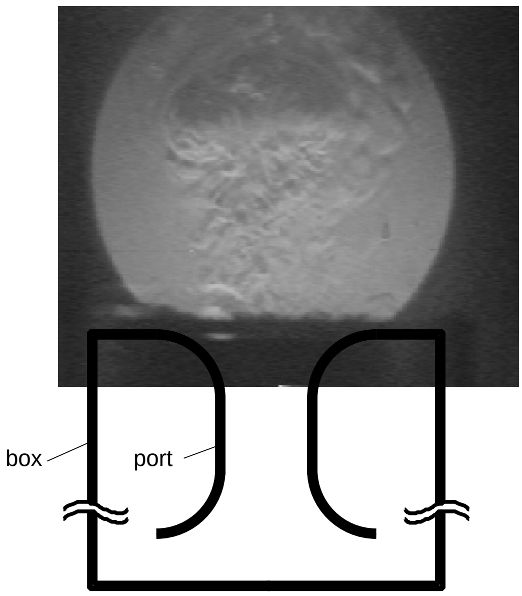

(d)

Figure 3: Schlieren visualization of the vortex shedding for successive instants of the half period of the harmonic cycle; a) the vortex develops, b),c) and d) the vortex moves away from the port.

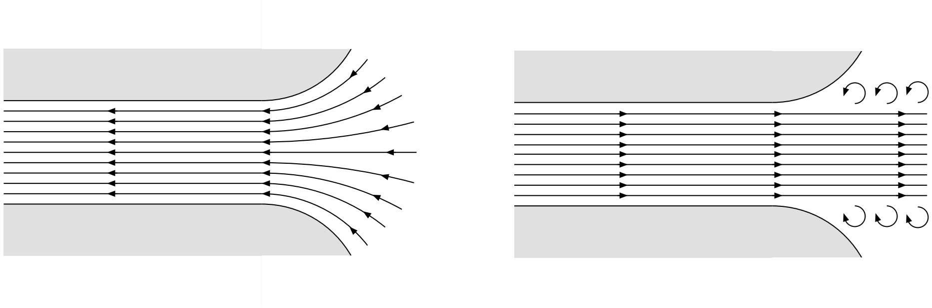

Figure 4: *Sink-like inflow (**left**) and jet-like outflow (**right**).*

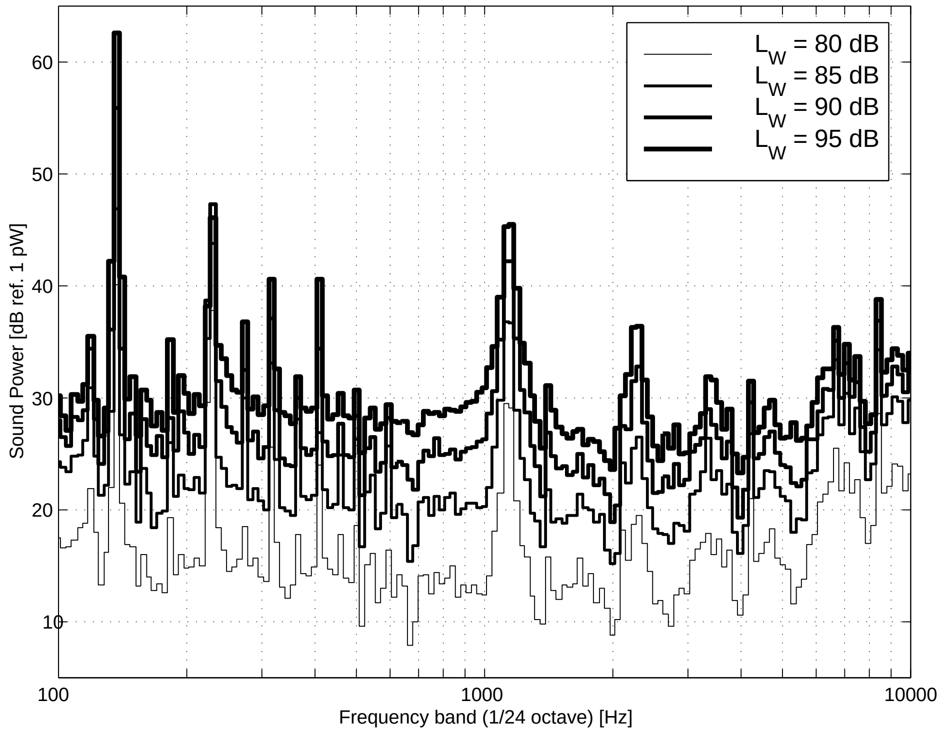

Figure 5: *Sound power of blowing sounds of a cylindrical bass-reflex port with sharp edges (port 0) for the sound power levels 80, 85, 90 and 95 dB at the driving frequency. Physical port length L=0.13 m. port cross-sectional radius a=0.014 m, Loudspeaker driven with a single frequency of 45 Hz.*

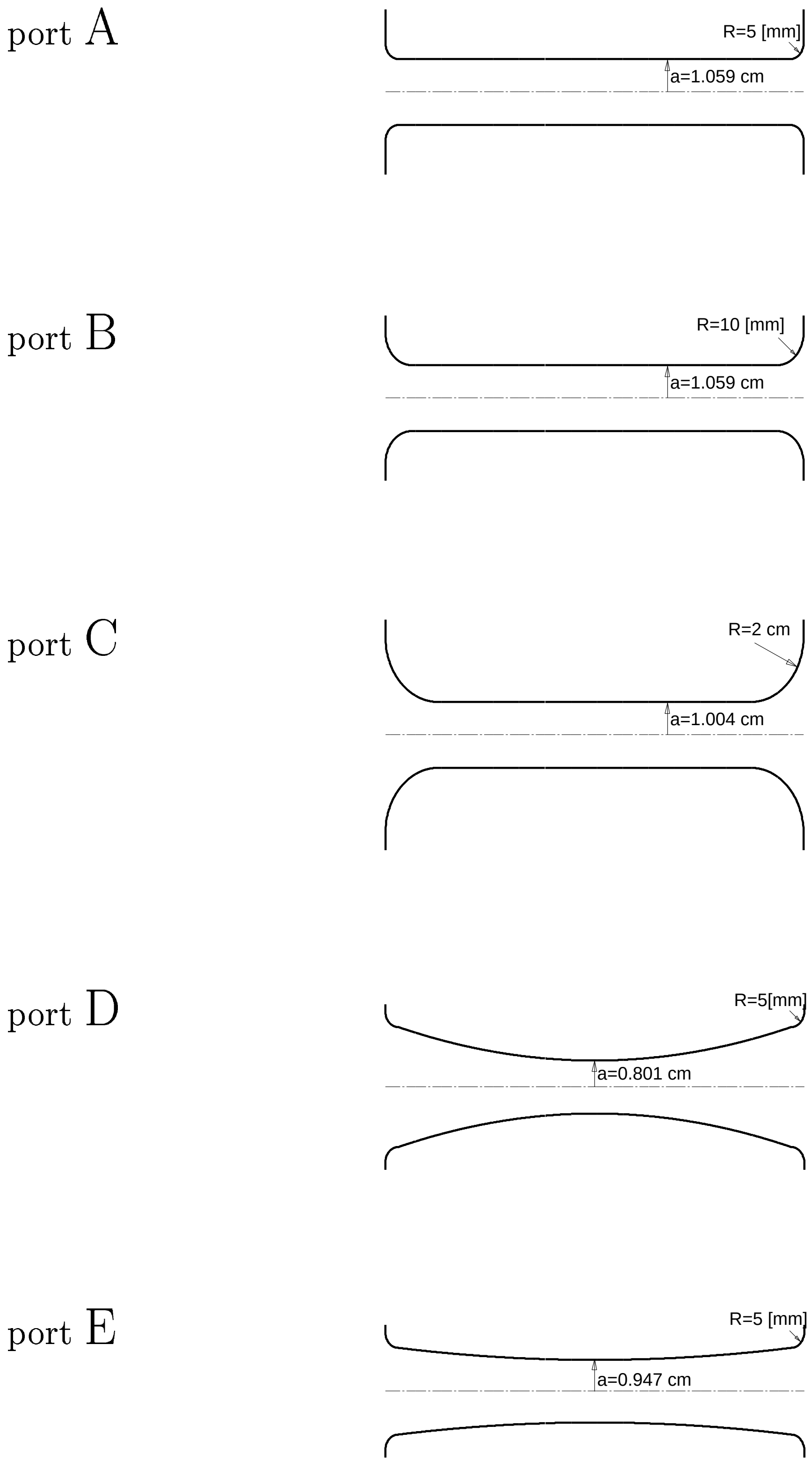

Figure 6: Port geometries.

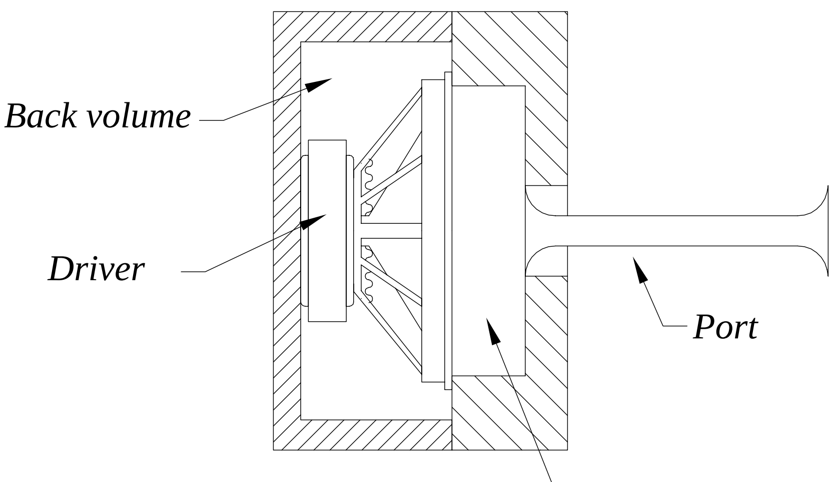

Figure 7: *Band-pass loudspeaker system.*

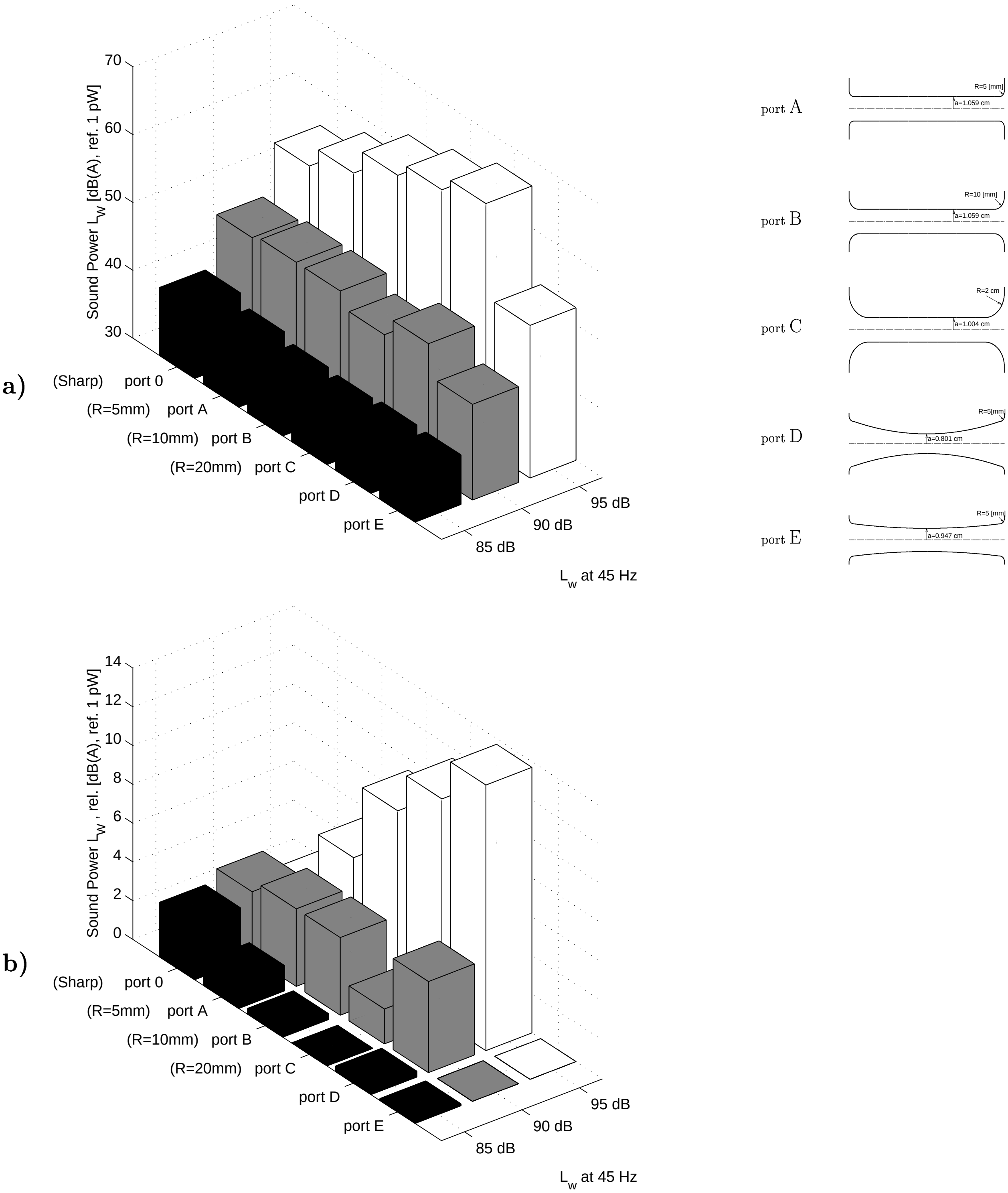

Figure 8: *Sound power of the blowing sounds for port 0, A, B, C, D and E, for equal sound pressure levels at the driving frequency at 1 m distance in an anechoic room,* **a)** *absolute sound power,* **b)** *sound power relative to port E.*

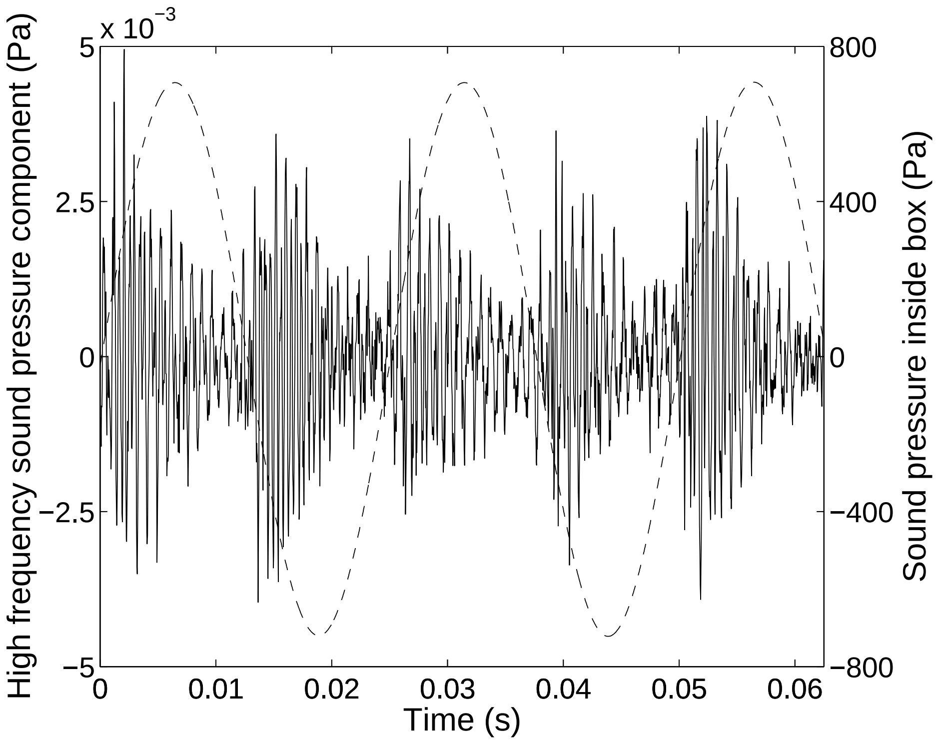

Figure 9: *High-pass filter results at the same sound pressure levels in the loudspeaker box at 40 Hz and at the same effective port lengths L_{eff}. Driving frequency 40 Hz. Cylindrical port geometry with sharp ends. Physical port length L=0.15 m, port cross-sectional radius a=0.01 m. Solid line: high-frequency sound pressure signal at 1 m distance outside box. Dashed line: sound pressure inside box.*

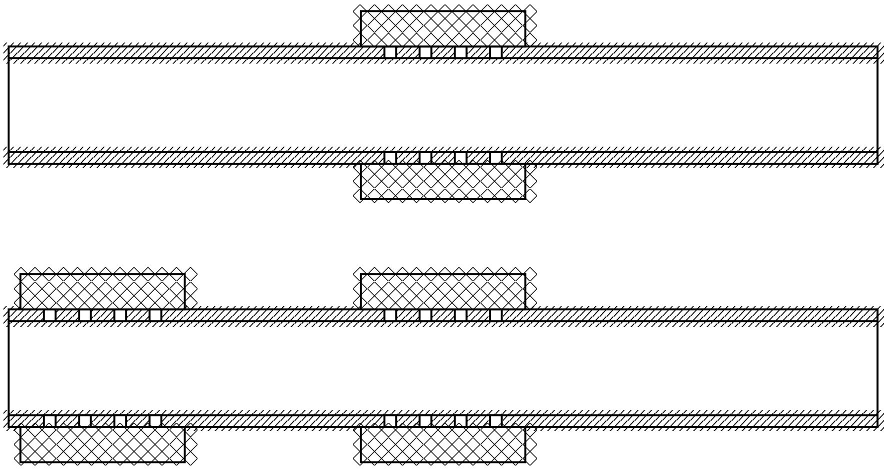

Figure 10: *Perforated cylindrical sharp ended port with foam wrapped around the port. Upper sub-figure: perforations and foam at* $x = L/2$*. Lower sub-figure: perforations and foam at* $x = L/2$ *and at* $x = 3L/4$*.*

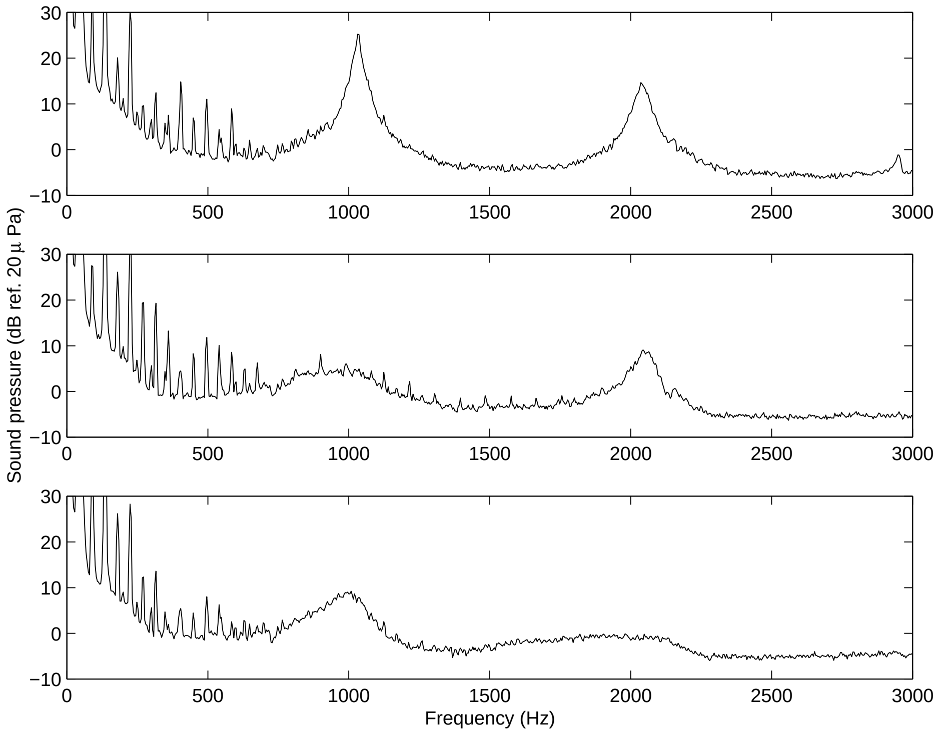

Figure 11: *Sound pressure at a distance of 1 m from a cylindrical sharp ended port, with and without perforations and foam. Upper sub-figure: no perforations. Middle sub-figure: perforations and foam at* $x = L/2$*. Lower sub-figure: perforations and foam at* $x = L/2$ *and at* $x = 3L/4$*.*