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The measurement of complex intensity near the open end of a flanged cylindrical pipe

J. Prisutova and K. V. Horoshenkov

University of Bradford, Great Horton Road, BD7 1DP Bradford, UK
j.prisutova@student.bradford.ac.uk

The measurement of acoustic intensity assumes taking the time average of the energy flow and treating the intensity as a real quantity. However, one may extract additional information if the intensity is viewed as a complex quantity which contains information about the local mean energy flow and local energy oscillations. In this paper experiments are reported using a tri-axial Microflown intensity probe to measure the complex instantaneous intensity near the open end of a flanged pipe. The measurements are then used to quantify the energy travelling out of the pipe and the energy oscillating back and forth inside the pipe. This has a potential significance for acoustic detection of structural and operational conditions inside the pipe as one would expect oscillating energy to be increased and travelling energy to be decreased in the presence of a cross-sectional change. These experiments are carried out in the high frequency range to show the interaction between a plane wave and the first circumferential mode. Here, the amplitude of the circumferential mode is found to be strongly related to the relative position of the source in the pipe, and the complex intensity is seen to change in transverse as well as the axial direction.

1 Introduction

There is a considerable number of studies which focused on the sound pressure distribution near the open end of a pipe. The methods proposed as a result of these studies include the normal model decomposition [1], ray tracing approximation [2] and combination of normal mode decompositions with the finite element method [3]. There has been a number of mainly experimental studies [4]. Recent advancements in measuring instruments have offered an opportunity to investigate tri-axial components of acoustic velocity / intensity [6], which enabled authors to experimentally analyse the sound intensity distribution in presence of higher order modes. It is believed that its behaviour is complex and strongly depends on the boundary conditions on the open end [5, ch.6]. The purpose of this work is to compare the numerical model against the experimental data for the sound pressure and intensity obtained near the end of a 150 mm diameter pipe in which the sound field is multi-modal. A better understanding of this phenomenon can pave the way to the development of more accurate acoustic instrumentation to inspect the conditions in underground pipes.

2 Experimental setup

In order to study the behaviour of the acoustic intensity near the open end of the flanged pipe, a series of controlled experiments were performed in the Acoustic Lab at the University of Bradford. These experiments were aimed at determining pressure, velocity and intensity distribution at the frequency above the 1^{st} cross-sectional mode cut-off frequency.

For the experiments, clean cylindrical 6 m long 150 mm diameter pipe was used, one end of which was terminated with a Fane compression driver and another was left open and was enclosed in a flange. Figure 1 shows the installation of the compression driver which was held at the bottom of the pipe in order to excite higher order modes. The frequency of the 1^{st} cross-sectional mode of this pipe is 1340 Hz.

Microflown USP probe which is able to record pressure and tri-axial components of an acoustic velocity vector was used for signal recording. It was fixed to a rigid frame which kept the probe near the wall of the pipe (at 0.065 m radial distance) (see Figure 2).

The probe was inserted into the open end and measurements were performed at several axial distances. At each distance the probe was turned in 10 degree steps covering the whole circle in order to obtain the angular pressure / velocity distribution. Measurements were performed at frequencies from 1500 to 2000 Hz, covering the axial distance from 0.1 to 0.25 m from the open pipe end, but in this paper, only the frequency of 1800 Hz and axial distance of 0.2 m from the open pipe end will be considered.

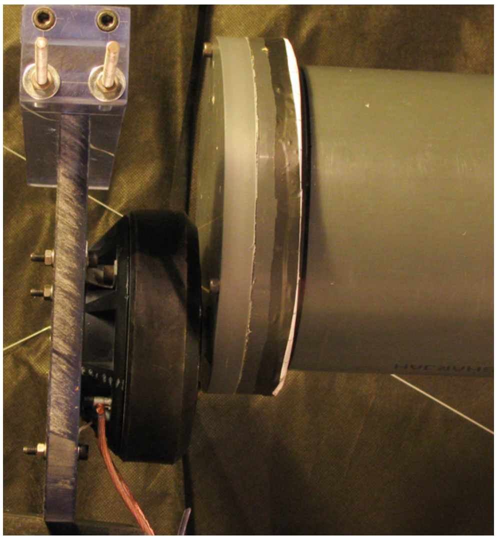

Figure 1: Loudspeaker installation

A signal used to excite sound field in the pipe was the Gaussian pulse. It was preferred to the continuous wave signal because it allowed to filter out unnecessary reflections and prevent the formation of standing waves. The Gaussian pulse was described using the following equation:

$$v(t) = A_0 \ \cos(\omega t) e^{-\frac{(t-t_0)^2}{\sigma^2}}, \qquad (1)$$

where t is time, t_0 is delay, ω is the angular frequency and σ is a temporal variable that controls the width of the pulse.

The signal was acquired using National Instruments DAQ NI PXIE-6358. LabVIEW software recorded it in data files at 48 kHz sampling rate and a Matlab code was used to analysed these files.

As the Gaussian pulse features more than one frequency, short-time Fourier transform was applied to the recorded signal in order to determine how the frequency content of a signal differs with time. Figures 3 and 4 show spectrograms at 1800 Hz, 0.20 m from the open end, angular position of 0 and 50 degrees respectively.

From the pictures, not only can it be clearly seen that the frequency content of the signal is not limited by the mode (1,0), but also that the spectrum varies from position to position. On Figure 3 there is a spectral line at about 2300 Hz, which corresponds to a cut-off frequency of a mode (2,0). In order to avoid its inclusion into analysis, the time history of the signal has to be cut before this spectral constituent ap-

Figure 2: Microflown USP probe, fixed to a frame and inserted into the open end of the pipe

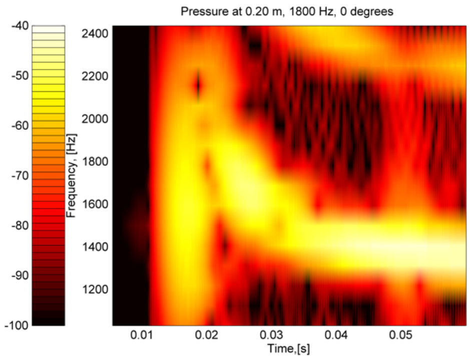

Figure 3: Pressure spectrogram at 1800 Hz, 0.2 m, 0 degrees

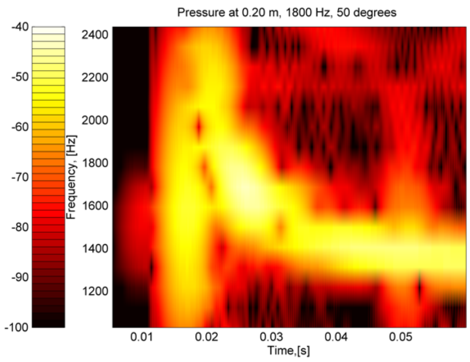

Figure 4: Pressure spectrogram at 1800 Hz, 0.2 m, 50 degrees

pears, which corresponds to the temporal interval from 0.01 to 0.03 s.

The example of time histories of the cut signal, which will be used in later analysis, are presented in Figures 5-7. Figure 5 shows the time history of pressure, recorded at angular position of 0 degrees. It can be seen that an incident pulse is followed by a modal disturbance, which is responsible for angular variations of sound field.

Figure 6 presents the time history of three components of acoustic velocity vector. The amplitudes of individual components are of the same order of magnitude, which is true for all considered angular positions. It means that none of them can be neglected in analysis (as opposed to plane wave regime, where the axial component alone gives plausible picture, see [4]).

Time histories of three components of the acoustic intensity can be seen in Figure 7. They were obtained by multiplying a time history of pressure by a time history of a corresponding velocity component:

$$I_j(t) = p(t)\, u_j(t), \quad j = 1, 2, 3. \tag{2}$$

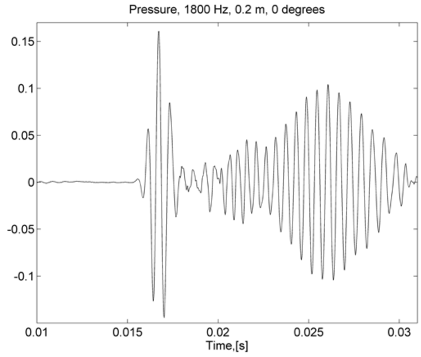

Figure 5: Pressure time history at 1800 Hz, 0.2 m, 0 degrees

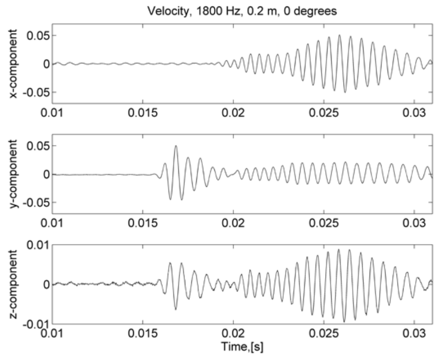

Figure 6: Velocity time history at 1800 Hz, 0.2 m, 0 degrees

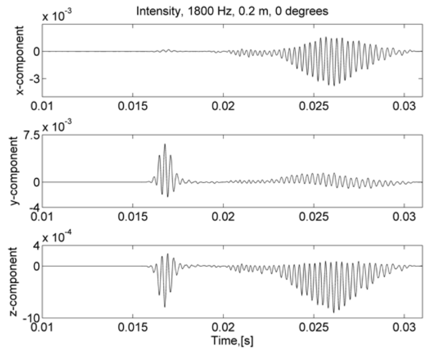

Figure 7: Intensity time history at 1800 Hz, 0.2 m, 0 degrees

It is also interesting to note how the time histories are changing with the altering angular position. Figure 8 shows time histories of radial intensity at 1800 Hz, 0.2 m axial distance, presented as a seismogram. Here it is easily noticeable how the oscillations following the plane wave pulse change in amplitude with an angle. Amplitude takes its maximum value on top and bottom of pipe cross-section (i.e. 90° and 270°) and decreases towards the sides of cross-section (0° and 180°). This effect can also be observed on Figure 11 which will be presented in Results and Discussion section.

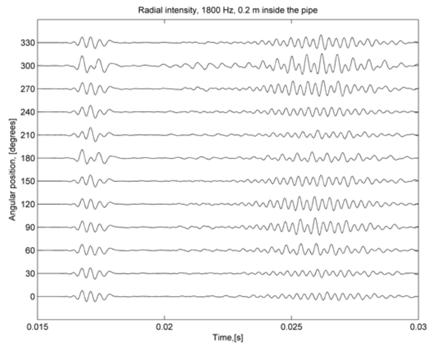

Figure 8: Radial intensity seismogram at 1800 Hz, 0.2 m

3 Numerical modelling

The numerical model was constructed at Brunel University, using the hybrid method described in [3]. One end of a pipe was terminated with a sound-hard boundary, and a monopole sound source was mounted on it. Another end was left open and was enclosed into an infinite flange. The pipe was divided into four regions. Two of them, containing the sound source and the end of the pipe with flange, were analysed using the finite element modelling, and other two were analysed using the normal mode decomposition method. After the solutions for each regions were obtained, the mode matching technique was used to match the solutions over the planes connecting the adjacent regions.

4 Results and discussion

After all time histories were acquired, Fourier transform was applied to each of them (see Eq. 3). Then, for each obtained spectrum the amplitude at frequency of interest was determined, giving the dependence of pressure / velocity amplitude on the angle.

$$
\begin{aligned}
p(\omega) &= \int_0^\infty p(t)\, e^{-i\omega t} dt, \\
u_j(\omega) &= \int_0^\infty u_j(t)\, e^{-i\omega t} dt, \quad j = 1, 2, 3.
\end{aligned} \tag{3}
$$

After this has been done for all circumferential points at one axial location and frequency, the intensity angular dependence was calculated by multiplying pressure and velocity angular dependences (Eq. 4). Then the results were normalised by the maximum value, plotted on the polar graph and compared with numerical predictions.

$$
I = \tilde{p}\tilde{u} \tag{4}
$$

Figure 9 shows the agreement between sound pressure as a function of angular position of the pipe computed by numerical model and measured in the laboratory. Figures 10 - 12 present similar result for three components of sound intensity - axial, radial and circumferential (angular). The results referred to in this paper are at a frequency of 1800 Hz and axial position of 0.2 m from the open pipe end. The reason for this particular choice was that it was experimentally observed that a frequency of 1800 Hz and a deeper axial position in the pipe give the better agreement with predictions.

There are several conclusions which can be drawn on the basis of Figures 10 - 12. Firstly, it was experimentally observed that variations in the intensity vector are rapid. It can undergo 100-fold change in only 50 degrees (see circumferential intensity component, Figure 12, $220° < \phi < 270°$). The same, although to a lesser extent, applies to radial and axial components. Such fast changes in the amplitude may present a difficulty to an acoustic reflectometry method (see [5]) as the quality of data gathered with it depends on the amplitude of intensity. So the amplitude variations have to be known beforehand, to enable compensation for them.

Secondly, both sets of data are in decent agreement. Experimental results clearly follow the same trend as numerical predictions, although they do not represent all the subtleties of the sound field in the vicinity of the open end. There might be several reasons for only partial agreement between predictions and experiment. It might be attributed to a presence of other frequencies in Gaussian pulse. Figure 3 distinctly shows that except from the 1^{st} cross-sectional mode resonant frequency, that of 2^{nd} mode is present in the recorded signal spectrum. The numerical model did not account for that. Also, the partial agreement may be induced by imperfect experimental setup. The recorded data seems to be quite capricious to the underlying measurement conditions. Such things as reflections from the walls and ceiling of the laboratory, minor vibrations of the pipe, even slight imperfections in pipe geometry may cause discrepancies in the experimental data. As the result depends very much on the ambient conditions, one has to account (and possibly compensate) for them.

Also, an inability of the laboratory results to fully follow the complex pattern of sound intensity field, reproducing all the minima accurately, may mean that present measuring instrumentation might be yet too rough to capture fast fluctuations of the sound field. Sound wave may increase in temperature while travelling along the pipe, which can cause faulty probe readings as well.

Finally, the raw recorded signal (as on Figures 5 - 7) presents itself as being rather complex. It is unclear what part of signal has to be taken for further analysis, as it is difficult to distinguish between contributions of different modes. One might need to precalculate their group velocities and apply this knowledge carefully to the recorded data to recognise individual modes.

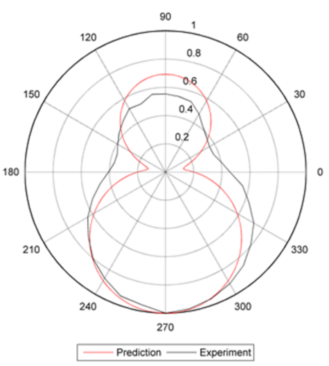

Figure 9: Pressure at 1800 Hz, 0.2 m

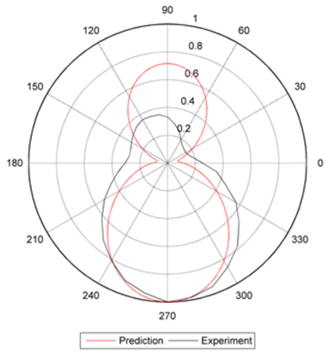

Figure 10: Axial intensity at 1800 Hz, 0.2 m

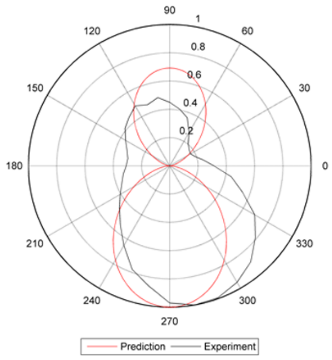

Figure 11: Radial intensity at 1800 Hz, 0.2 m

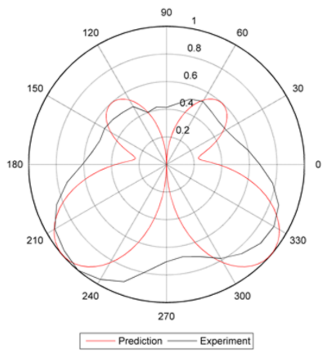

Figure 12: Circumferential intensity at 1800 Hz, 0.2 m

5 Conclusions

In this paper, an experimental method for sound field measurements above the 1^{st} cross-sectional mode is presented. The experimentally obtained data is compared to the results of the finite element model, which predicts sound pressure / velocity / intensity field in the vicinity of the open end of the pipe at frequencies between 1^{st} and 2^{nd} cross-sectional modes, which is $1340\,Hz < f < 2230\,Hz$ for the given pipe. Generally, two sets of data are in agreement with each other. Measured data follows the trend in numerical data, although fails to represent the reality correctly near the regions with fast field fluctuations. It may be caused by several reasons, such as the fact that Gaussian pulse contains other frequencies except from the frequency of interest, or imperfections in the laboratory setup, or insufficient advancement of measuring devices. Likewise, it is not entirely obvious which fragment of the recorded signal is carrying the useful information. Some future work might need to be done to understand that.

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