CONTACT-FREE VIBRATION MEASUREMENTS WITH PARTICLE VELOCITY PROBES

D.Sadra, Hamburg University of Applied Sciences, Department of Automotive and Aeronautical Engineering, Berliner Tor 9, 20099 Hamburg, Germany

Abstract

To identify mode shapes of vibration structures, it is necessary to perform vibration measurements using structural sensors (such as accelerometers) attached to the vibrating structure. As a consequence, the structural response is modified due to the adding of extra masses. To avoid this effect, especially during the test of lightweight structures, it can be advantageous to replace vibration measurements by acoustic measurements. For this purpose, it is possible to use particle velocity (pu) probes in the very near field of the vibrating structure. To describe the theoretical approach, a simplified vibro-acoustic model for a one dimensional sound tube has been analyzed [1]. To be independent of a signal measured with sensors that are directly mounted on the structure, a coherence based measurement approach has been analyzed as the measurement error calculated.

1. INTRODUCTION

To identify mode shapes of vibrating structures, it is necessary to perform vibration measurements. In many cases structural sensors (such as accelerometers) are attached to the vibrating surface. As a consequence, the structural response of the system is modified.

To illustrate this effect, it is useful to analyze the first natural frequency of a simply supported beam structure (length: L, constant mass distribution: μ, bending stiffness: EI) with an additional discrete point mass m. The latter may be attached at $L/2$. Assuming a harmonic deflection shape such as

$$w(x,t) = \sin\left(\frac{\pi \cdot x}{L}\right) \cdot q(t) \qquad \text{[m]} \quad (1)$$

compare [1], with $q(t)$ being a generalized coordinate, the first natural frequency is given by

$$f = \frac{\pi}{2L} \cdot \sqrt{\frac{EI}{L}} \cdot \sqrt{\frac{1}{(\mu L + 2m)}} \cdot \qquad \text{[Hz]} \quad (2)$$

To avoid such an effect, especially during the test of lightweight structures, it can be advantageous to replace structural measurements with acoustic measurements. For this purpose, it is possible to use particle velocity probes (pu-probes), compare [2], in the very near field of the vibrating structure as discussed in the present paper.

2. SIMPLIFIED VIBRO-ACOUSTIC MODEL

In order to analyse the feasibility of contact-free vibration measurements using pu-probes, a simplified vibro-acoustical model, described in [3], has been used to perform numerical investigations. It describes one-dimensional propagation of plane waves through a sound tube. At the right hand side, the tube is terminated by an absorbing boundary with prescribed impedance. The acoustic field results from the harmonic motion of a piston at the left hand side. The latter is driven by an excitation force and connected to a spring-damper combination (parallel connection). All system parameters, taken from [3], are listed in Tab. 1.

Tab. 1: Physical properties of coupled system, [3]

Spring-damper-piston system (structure)		
Area of piston cross section	$A =$	0.000625m
Spring stiffness	$k =$	7474.75N/m
Mass of piston	$m =$	0.01kg
Viscosity of damper	$b =$	0.50kg/s
Sound tube (cavity)		
Length of tube	$l =$	1.25m
Area of tube cross section	$A =$	0.000625m
Speed of sound in air	$c =$	344m/s
Density of air	$\rho =$	1.205kg/m³

It should be noticed that the first natural frequency of the air-filled tube matches the natural frequency of the undamped spring-piston system (137.7Hz), if a sound hard termination is realized at the right hand side. For this reason, the first two natural frequencies of the sound hard terminated coupled system (128.3Hz and 147.2Hz, see Fig. 1) are typical for a strong vibro-acoustic coupling effect, compare [3].

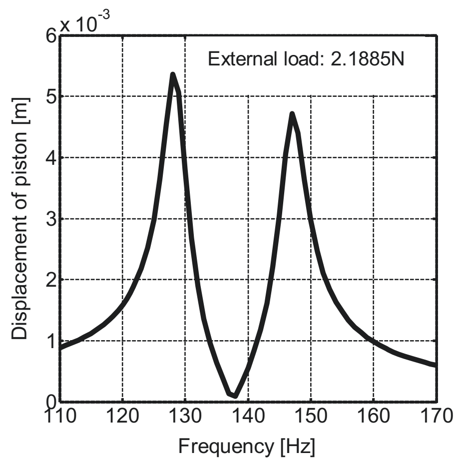

Fig. 1: Forced vibration of coupled system, compare [4]

A closed form solution considering arbitrary boundary conditions at the impedance boundary and time-harmonic excitation can also be found in [3] (equation 132–141). Unfortunately it turned out that the above named reference contained typesetting errors. Therefore, we authors would like to recommend two corrections

- Correction 1: Insert ν into equation 135 – see the sine-functions in the denominator,
- Correction 2: Change the numerator in equation 136 (definition of σ) from $-Z_0Z_1$ to $-Z_0Z_2$.

2.1. Theoretical investigations

To verify the feasibility of contact-free vibration measurements, the sound tube model has been analysed. As shown in Fig. 2, the relative deviation between the velocity of the piston and the particle velocity becomes significant, especially at the anti-resonances of the piston (found at 138Hz, 276Hz, and 414Hz), if the distance between piston and pu-probe increases.

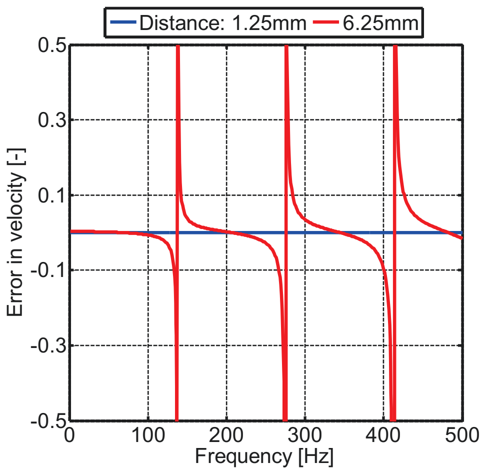

Fig. 2: Relative error in velocity measurements, see [4]

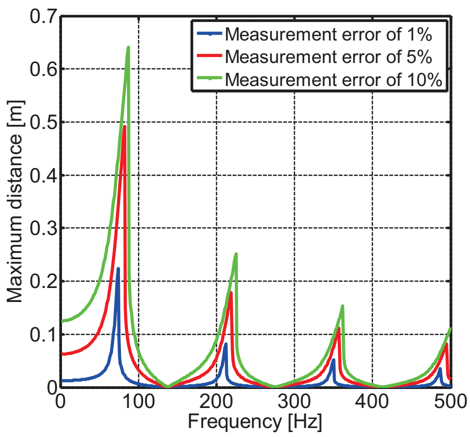

Fig. 3: Absolute values of relative error, [4]

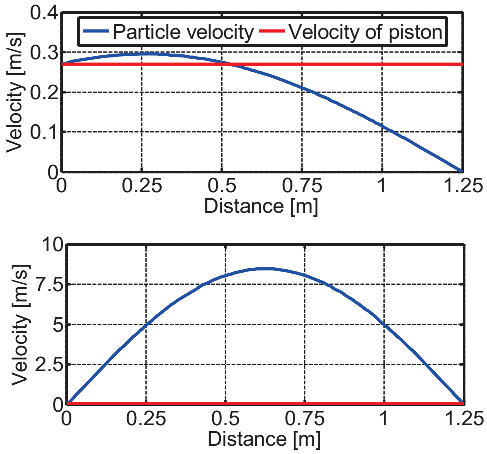

Fig. 4: Mode shapes for 87Hz (top) and 138Hz (below), see [5]

Upper bounds for the distance between pu-probe and piston that guarantee a defined deviation between the structural velocity and the particle velocity are shown in Fig. 3. It can be seen that a distance up to 0.5m is acceptable, if (i) a relative error of 5% can be tolerated, (ii) and the spatial gradient is small, compare Fig. 4 (top). Otherwise, compare Fig. 4 (below), a small distance is required.

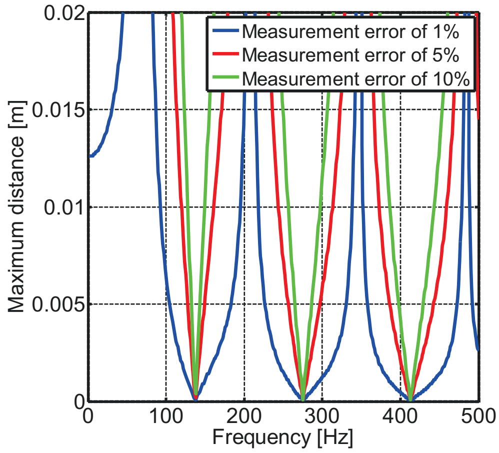

Fig. 5: Absolute values of relative error, compare [4]

If the maximum distance between the pu-probe and the piston is set to 0.01m, see Fig. 5, and an absolute value for the relative error of 10% can be accepted, it is possible to perform contact-free vibration measurements in a sub-band of in total 322Hz below 400Hz, as shown in the last row of Tab. 2.

Tab. 2: Frequency bands of defined errors (1cm distance), [5]

Absolute value of relative error: 1%		
0-92Hz	195-218Hz (Δ=23Hz)	338-352Hz (Δ =14Hz)
Absolute values of relative error: 5%		
0-120Hz	161-244Hz (Δ =83Hz)	313-373Hz (Δ =60Hz)
Absolute values of relative error: 10%		
0-127Hz	150-256Hz (Δ =106Hz)	297-386Hz (Δ =89Hz)

3. VIBRO-ACOUSTIC TEST ENVIRONMENT

The test-rig consists of an acoustic cavity (1.495m 0.95m x 0.2m). Inside this cavity a simply support beam structure (Aluminum, 0.895m x 0.03m x 0.002m) used to excite the sound field. A loudspeaker has be used to simulate the effect of background noise (BGI The measuring equipment given in Tab. 3 has been use

Tab. 3: Overview on the measuring equipment

Component	Type
pu-probe	USP-Regular (Microflown)
accelerometer	Bruel&Kjaer Type 4517
loudspeaker	Eighteensound 6ND430
modal analysis hardware	SCADAS mobile, LMS
power amplifier	Bruel&Kjaer Type 2706
audio amplifier	Kenwood KRF-V7020
electro-dynamical exciter	ELAC Standart, Autotune II

4. MEASUREMENT OF STRUCTURAL VIBRATION

4.1. Behavior of the test-rig

To analyze the behavior of the test rig, the natu frequencies have been determined.

Tab. 4: Resonances of the test rig

	cavity	beam structure
resonance	**f_n [Hz]**	**f_n [Hz]**
1	10.4	134.2
2	21.5	229.0
3	55.2	307.0
4	94.1	352.0
5	138.4	398.0
6	193.2	502.0

The resonances of the beam structure have been detected with a modal analysis system, the resonances of the cavity using the microphone signal of the pu-probe. The latter has been placed in a corner of the cavity. broadband signal has been used to excite the enclosu via the BGN-loudspeaker. Tab. 2 contains the first resonances of the uncoupled systems – beam structu and cavity.

4.2. Coherence based measurement techniqu

To be independent of a signal measured with senso that are directly mounted on the structure, a coheren based approach has been analyzed. At first t acceleration and the normal component of the pu ha been measured at a particular position consideri broadband structural excitation without BGN. T... coherence between these signals is shown in Fig. 6. Three different distances between the pu-probe and beam structure have been examined.

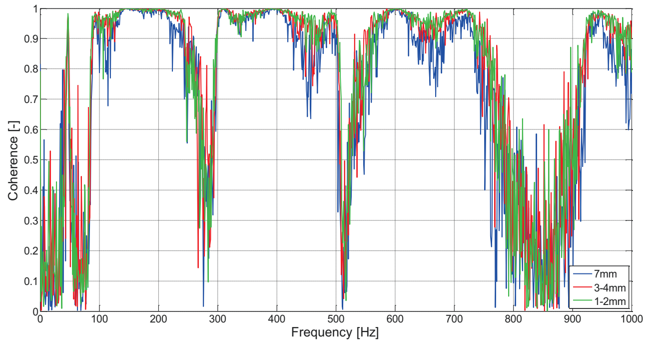

Fig. 6: Coherence between signal of accelerometer and normal component of particle velocity without BGN, [4]

Because the measurement has been performed close to the structure, both, normal and tangential component of the pu should be related to the structural vibration. Therefore, the coherence between these signals has been calculated (see Fig. 7).

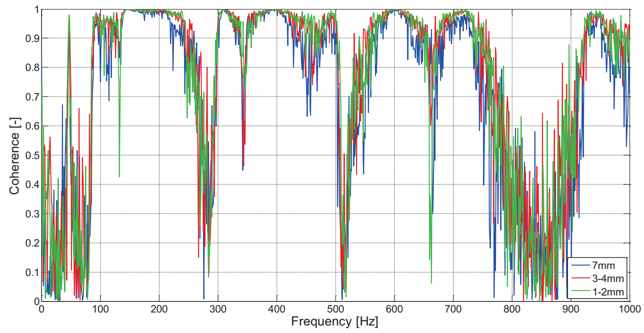

Fig. 7: Coherence between normal and tangential component of the particle velocity without BGN, [4]

It can be noticed that the curves shown in Fig. 6 and Fig. 7 are nearly identical. To quantify the effect of BGN, the measurement has been repeated. For this purpose, the BGN has been adjusted to 82dB sound pressure level (SPL). The results are shown in Fig. 8.

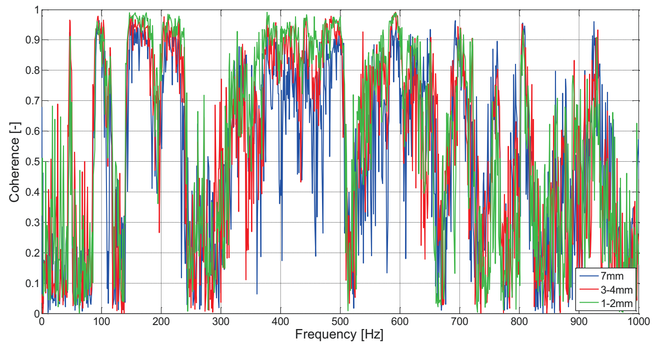

Fig. 8: Coherence between normal and tangential component of particle velocity with BGN, [4]

The BGN clearly affects the coherence. The same holds for the distance between probe and structure. This indicates, that the coherence between normal and tangential component of the pu can be used to quantify the effect of BGN.

4.3. Signal-to-noise ratio

According to the linear system theory, the signal-to-noise ratio (SNR) can be calculated as follows.

$$\mathrm{SNR} = 10\log_{10}\left(\frac{\left|H_1(j\omega)\right|}{\left|H_2(j\omega)\right|}\right)^2 + 10\log_{10}\left(\frac{S_{u1u1}(j\omega)}{S_{u2u2}(j\omega)}\right) \quad \text{[dB]} \quad (3)$$

H_1 is the complex transfer function between the driving signal of the exciter and the normal component of the pu. S_{u1u1} and S_{u2u2} are the auto spectral densities of the electrical voltage which drive the actuators. The first summand represents the effect of the transfer paths on the SNR. The effect of auto spectral densities of the actuator driving signals is represented by the second summand. Fig. 9 shows the measured transfer functions as well as the SNR calculated by Eqn. 3. If the magnitude of H_1 is below the one of H_2, a low SNR can be detected. Thus not all natural frequencies can be observed with the pu probe, if the SPL caused by the exciter is too low.

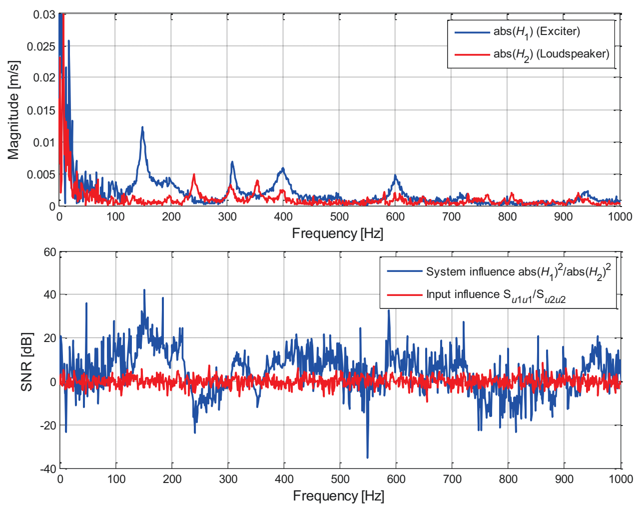

Fig. 9: Transfer functions and SNR, [4]

Finally the absolute error has been compared for both signals (Fig.10, acceleration a_s and normal component of

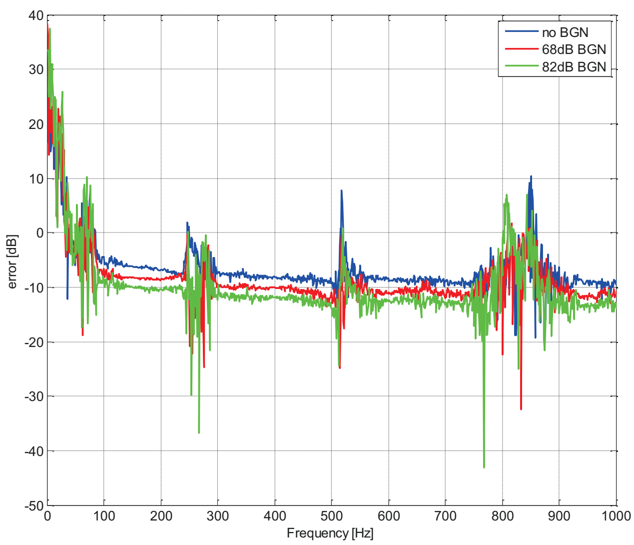

Fig. 10: Measurement error with/without BGN, [4]

Therefore the conversion given in Eqn. 4 has been applied.

$$T_{a_s v_s} = \frac{v_n}{a_s} = \frac{v_n}{j\omega \cdot v_s} = \frac{1}{j\omega}\frac{v_n}{v_s} = \frac{1}{j\omega}T_{v_s v_n} \quad \text{[m/s]} \quad (4)$$

The absolute error has been calculated using Eqn. 5.

$$\text{error} = 20\log_{10}\left(\left|T_{a_s v_s}\right| \cdot \omega\right) \text{ with } \omega = 2\pi f \quad \text{[dB]} \quad (5)$$

Without BGN, the loss in signal measured with the Microflown is (for most of the frequencies) -8dB. The error increases when the BGN increases, see Fig. 10.

5. CONCLUSION

At first, a theoretical model has been analyzed to get some information about measurement errors according to allowed distances for the use of the pu probe. For the experimental investigations, A coherence based approach has been used to proof that it is not necessary to use structural sensors, such as accelerometers, to measure the structural vibration. Furthermore, the influence of BGN has been examined. Moreover, a method for the qualification of the SNR using the transfer function and the auto spectral density has been analyzed.

6. ACKNOWLEDGEMENT

At first, I would like to express my deep gratitude to Prof. Dr.-Ing. (habil) Thomas Kletschkowski for his patient guidance, enthusiastic encouragement and useful critics throughout my master's thesis. Moreover, I would like to thank Microflown Technologies (Arnhem, the Netherlands) for lending the pu-probe as well as the Helmut-Schmidt University Hamburg, (chair for Mechatronics, Prof. Dr.-Ing. Delf Sachau) for lending additional parts of the measuring equipment.

7. LITERATURE

[1] Brommundt, E., Sachau, D.: Schwingungslehre mit Maschinendynamik, Teubner, 2008

[2] de Bree, H.-E.: The Microflown – an particle velocity sensor. Acoustics Australia (2003), pp 91-94, Vol. 31 No. 3

[3] Chargin, M., Gartmeier, O.: A Finite element procedure for calculating fluid-structure interaction using MSC/NASTRAN. California: NASA, December 1990. NASA Technical Memorandum 102857

[4] Sadra, D.: Contact-free vibration measurements with particle velocity probes. Master-thesis, Hamburg University of Applied Sciences, 2013

[5] Kletschkowski, T., Sadra, D.: Contact-Free Vibration Measurements with Particle Velocity Probes – Part II: Theory. Proc. of AIA-DAGA 2014, Oldenburg, Germany