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REDUCTION OF THE NOISE TRANSMISSION THROUGH A DOUBLE GLAZED WINDOW DUE TO ADAPTIVE MEASURES

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Abstract. *In this work the sound transmission through a double-glazed window is reduced due to adaptive methods with a focus on an adaptive Helmholtz resonator acting on the sound field in the air gap between the two glass plates. As test-rig a double glazed window is mounted on a rectangular cavity. The system is experimentally characterised and simulated using the commercial numerical software ANSYS. In termes of the noise transmission the gained model is reduced using model order reduction techniques and transferred to MATLAB/Simulink. Evaluating the transmitted noise the radiated sound pressure is calculated using the acoustic monopole synthesis.*

1 INTRODUCTION

Working against growing noise levels windows turn out as great weak spots regarding the sound transmission paths into rooms. Incoming sound waves excite the outer glass plate. The vibrations are transmitted by the acoustic fluid inside the window to the inner glass plate. Although the acoustic energy is decreased by each impedance step the vibrations effect a considerable sound radiation inside the building.

By the use of adaptive methods, affecting the acoustic fluid between the glass plates, it gets possible to reduce the sound transmission through the window [1]. For the investigation of this method a cavity with a mounted double-glazed window is considered. In this function the walls have a hard sound reflecting characteristics that have been validated through experimental analysis [10]. Exciting the acoustic fluid inside the cavity, measures reducing the sound transmission outwards are designed and tested.

Evaluating adaptive methods high efficient and precise simulation models are needed. In this manner a fluid-structure interacting Finite-Element model is created using ANSYS. This model is validated performing both structural and acoustical experimental modal analysis. The gained model is reduced using model order reduction methods [16]. For the calculation of the acoustic radiation of the outer glass plate the reduced model is transferred to MATLAB. Here the produced sound pressure in the far field is calculated discretising the surface to connected monopoles. The sound radiation of each monopole can be calculated deriving the displacement of the reduced Finite-Element model performed by the surface of each radiator. The spatial radiated sound-pressure is gained by the superposition of each monopole on the surface [6, 14]. Using Simulink adaptive methods are evaluated reducing the acoustic emission of the designed test rig. This way the design and evaluation of different adaptive methods and the regarding control strategies reducing the transmitted sound-energy are realised.

2 DESIGN AND EXPERIMENTAL CHARACTERISATION OF THE TEST-RIG

Investigating the interaction of an elastic structure with the acoustic fluid inside a cavity the acoustics demonstrator of the LOEWE Centre AdRIA was designed as simplification of a real room with given dynamic structural and acoustic specifications (See Figure 1). It profits of a numerical and analytical reproducible behaviour in the frequency domain up to 500 Hz [10]. Until that frequency range a couple of cavity modes are present and no destructive interaction with structural modes occure. Mounting different elastic structures on the demonstrator different approaches regarding Smart Structures can be investigated. The main goal here is the reduction of the transmitted sound energy through the elastic structure. By this, the noise source is located in the cavity with the inner dimensions of 870x620x750 mm³.

In the current investigations a double-glazed window is mounted on the five rectangular rigid sound hard walls (See Figure 2) using a two part frame structure made of stainless steel. The frame has an overall thickness of 50 mm and a width of 60 mm and is interconnected to the demonstrator with 28 screws. A double-glazed window with a symmetric design consisting of two glass plates with a dimension of 900x650x4 mm³ with an air gap of 8 mm is used.

Existing soundproof windows suffer of a drop in the sound reduction index for a frequency domain approximate below 300 Hz [5]. As the the acoustic fluid inside the demonstrator shows the first natural frequency at 196.8 Hz (0,1,0-mode) it seems reasonable, that a frequency domain up to 200 Hz is investigated in the current work where no negative influence of the cavity with the structural modes can be expected. The attenuation of the system can either be done by shaker excitation of the double glazed window in termes of the characterisation of the struc-
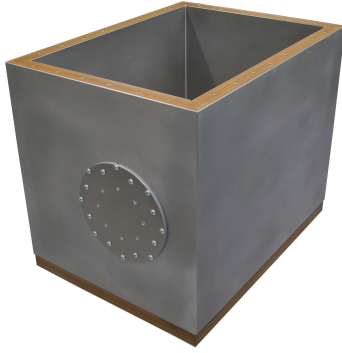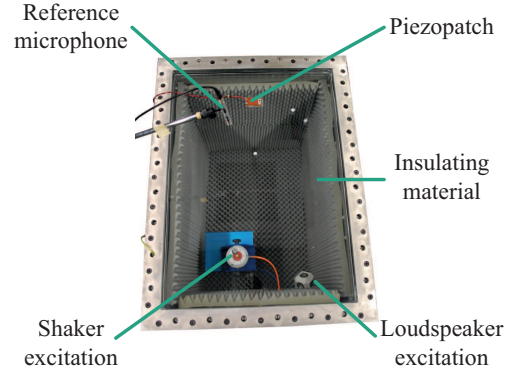
Figure 1: Acoustic demonstrator

Figure 2: Realised test-rig of the double glazed window mounted on the acoustics demonstrator

tural behaviour or a loudspeaker regarding the acoustic characterisation and investigations of the transmission of the system. The corresponding coordinates of the excitations are listed in Table 1. Obtaining a direct noise emission to the window from the loudspeaker, insulation material is mounted on the walls of the demonstrator.

Parameter	Coordinate in mm		
	X	Y	Z
$HR_{in,1}$	620	0	758
$HR_{in,2}$	620	870	758
$HR_{in,3}$	0	0	758
$HR_{in,4}$	0	870	758
q_{in}	620	0	0
f_{in}	390	690	750
$u_{out,lower}$	390	690	750
$u_{out,upper}$	390	690	766
p_{out}	0	40	758

Table 1: Coordinates of the in- and output interface position

Generating detailed numerical simulation models, (see Section 3) the behaviour of the coupled acoustic and structural systems have to be characterised. The experimental modal analysis is performed using a 3D scanning laser vibrometer with a force excitation at the position f_{in} seen in Table 1. This method has the advantage, that the in-plane and out-of-plane waves of both glass plates can be measured contactless without tearing the test-rig apart. The modal analysis of the measured date is performed using LMS Test.Lab. The resulting Eigen frequencies and the corresponding modeshapes are listet in Table 3. In addition the frequency response functions (FRF) to $u_{out,lower}$ and $u_{out,upper}$ are pictured in Figures 3 and 4. Considering the real part a phase shift can be observed for some Eigen frequencies. Here the upper and the lower plate vibrate in opposite phase. This characteristic underlies the symmetrical set-up of the double-glazed window resulting in a high sound pressure in the air gap of the window. Therefore the chosen reduction method will have its most effectiveness in this frequencies. The corresponding opposite vibrating modes are marked in Table 3 with a (*).
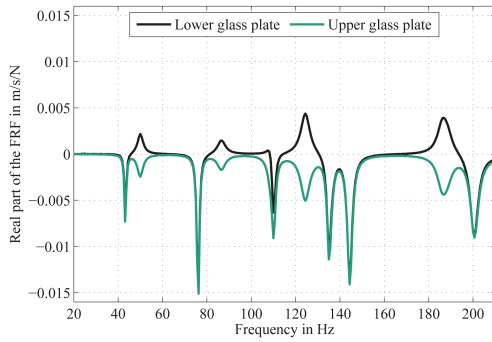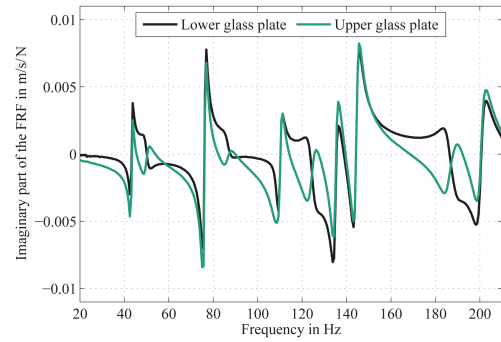
Figure 3: Real part of the measured FRF

Figure 4: Imaginary part of the measured FRF

3   SIMULATION OF THE NOISE TRANSMISSION

The design of adaptive methods for the reduction of noise and vibrations demand high accurate and efficient simulation models. By this means the vibro-acoustic system of the acoustics demonstrator coupled with the double-glazed window is implemented by Finite-Element (FE) software. The estimation of the radiated sound pressure is a highly memory and calculation intensive process. In FE-models the analysis often produce models with more than $2 \cdot 10^6$ degree of freedoms. Therefore the gained model is reduced using model order reduction methods and transferred to MATLAB/Simulink where the acoustic radiation is calculated using the less intensive acoustic monopole synthesis

3.1   FE-Simulation of the coupled system

The fundamental work of the simulation is performed in ANSYS. Here the acoustic fluid inside the acoustics demonstrator coupled with the double-glazed window is modeled. For the model evaluation the force excited glass plate at f_{in} is adapted to the FRF at $u_{out,lower}$ and $u_{out,upper}$ to the laser scanning vibrometer measurements gained in Section 2. The material properties listed in Table 2 are used.

Property	Value
Density air	$1.1886 \frac{\text{kg}}{\text{m}^3}$
Sonic speed air	$343.5 \frac{\text{m}}{\text{s}}$
Dynamic viscosity air	$17.1 \mu \text{Pa} \cdot \text{s}$
Density glass	$2500 \frac{\text{kg}}{\text{m}^3}$
Young's modulus glass	$70 \frac{\text{kN}}{\text{mm}^2}$
Poissons's ratio	0.19
Axial stiffness	$250730 \frac{\text{N}}{\text{m}}$
Normal stiffness	$452980 \frac{\text{N}}{\text{m}}$
Tangential stiffness	$250730 \frac{\text{N}}{\text{m}}$

Table 2: Physical properties of the used materials

Both the sound field and the structure are implemented using quadratic elements. The design of the fluid structure interaction problem is performed with FLUID220 elements for the acoustic

fluid and SOLID186 elements in terms of the structure [2]. The elastomer mounting of the glass plates are evaluated using MATRIX27 elements. This way the behaviour of the simulated window can be adapted to the real one by adjusting the stiffness in normal, tangential and axial direction. The optimal results of the performed parameter optimisation can be see in Table 2. The resulting Eigen frequencies of the FE-simulation are listed in Table 3. Here an average divergence of 2.7% can be gained comparing the simulation with the measurements implying a sufficient accuracy for the subsequent investigations.

| Rank | Mode | Measured | | Simulated | Divergence in % |
		Natural frequency in Hz	Modal damping in %	Natural frequency in Hz	
1	1,1	43.2	1.10	43.2	0.1
2	1,2*	50.0	3.07	47.6	5.0
3	1,2	76.0	0.90	76.7	0.9
4	2,1*	86.5	2.55	82.7	4.6
5	1,3*	108.7	1.59	105.8	2.7
6	2,1	110.1	0.87	113.1	2.7
7	2,2*	124.4	2.01	119.7	3.9
8	1,3	135.1	0.89	138.0	2.1
9	2,2	144.4	0.92	149.8	3.7
10	2,3*	186.2	1.49	179.2	3.9

Table 3: Comparison of the Eigen frequencies gained from the FE simulation with measurements of the mounted double-glazed window (*-corresponding opposite vibrating modes)

3.2 MODEL ORDER REDUCTION OF THE VIBRO-ACOUSTIC SYSTEM

For further investigations of the numerical system in MATLAB/Simulink the model has to be reduced using model order reduction techniques. In the current work a variation of the balanced truncation, the modal truncation, is used for the reduction of the coupled vibro-acoustic model. Therefore the system has to be transferred from physical to modal coordinates using orthogonality relationships [6, 16] resulting in

$$G(\omega) = \sum_{i=1}^{n} \frac{\Phi_i \overline{\Phi_i}^T}{\mu_i \omega_i^2} D_i(\omega) \quad \text{, where} \quad D_i(\omega) = \frac{1}{1 - \frac{\omega^2}{\omega_i^2} + 2\mathrm{i}\xi_i \frac{\omega}{\omega_i}} \tag{1}$$

is the dynamic amplification factor of mode i. This way the system can be described using the FE-model with n modes with the corresponding modal mass μ_i, mode shape Φ_i, natural frequency ω_i and modal damping ratio ξ_i estimated through the previous experimental modal analysis [3]. As a result of the fluid structure interaction the coefficient matrices are not symmetrical resulting in problems regarding the orthogonalisation through the left and right multiplication by the Eigen modes of the system. Therefore the matrices have to be left multiplied by the transposed revised orthogonality revision [12]

$$\overline{\Phi_i} = \left\{ \begin{array}{c} \Phi_{\mathrm{s},i} \\ \frac{1}{\omega_i^2} \Phi_{\mathrm{a},i} \end{array} \right\} \tag{2}$$

5

with the corresponding structural Eigen modes $\Phi_{s,i}$ and the acoustical Eigen modes $\Phi_{a,i}$. In Figures 5 and 6 the resulting FRF of the reduced simulations are compared to the measured functions of the lower and upper glass plate. It can clearly be seen that the simulation results matches the results of the real system and are therefore used as a basis for the calculation of the radiated noise.
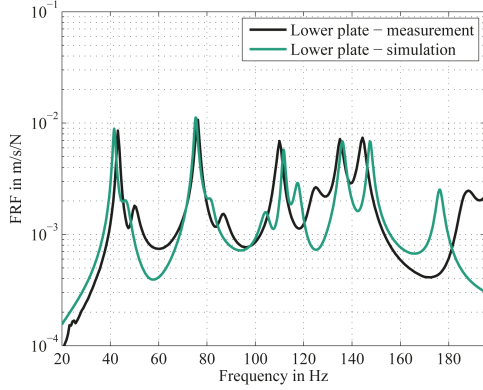

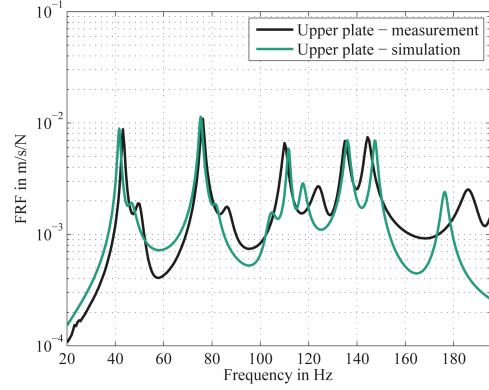Figure 5: Mean FRF of the force excitation on the lower glass plate

Figure 6: Mean FRF of the force excitation on the upper glass plate

3.3 CALCULATION OF THE ACOUSTIC RADIATION

The radiated acoustic pressure is calculated dividing the surface in small monopole sources emitting the noise synthesised for each position in the considered far field [6, 14]. For a low frequency domain, thin surfaces vertical vibrations dominate as contrast to the flexural rigity. The surface velocity can be assumed equal to the sound particle velocity of the surrounding medium and the radiated acoustic pressure can be calculated as summation of each monopole

$$p(x,y,z) = \frac{i\omega\rho_0}{4\pi}\left(\frac{q_1 e^{-ikr_1}}{r_1} + \ldots + \frac{q_m e^{-ikr_m}}{r_m}\right) = \frac{i\omega\rho_0}{4\pi}\sum_{n=1}^{m}\frac{q_n e^{-ikr_n}}{r_n} \tag{3}$$

with the volume displacement source of each monopole $q_n = S_n v_n$ where S_n is the surface and v_n the velocity of each source, the radius r_n of the point considered for the sound radiation and the corresponding wave number k. Calculating the acoustic radiation of three dimensional spatial extensive structures also the visibility of the observation point regarding the vibrating surface has to be taken into consideration. Therefore the view factor, commonly used in the field of illuminating engineering and radiated heat transfer is introduced to the calculations of each radiated monopole

$$F_{A_n-A_o} = \frac{1}{\pi A_n}\int_{A_n}\int_{A_0}\frac{\cos\varphi_n\cos\varphi_0}{r_{n,0}^2}dA_o dA_n \tag{4}$$

depending to the size A_0, the distance $r_{n,0}$ and the orientation φ of the surface [4]. In Figure 7 the radiated sound power on a half sphere with a radius of 1000 mm around the center of the upper glass plate excited using a force of 1 N at f_{in} calculated using MATLAB is compared to the solution gained using LMS Virtual.Lab 10-SL3. The software uses the boundary element method to calculate the radiated sound field. Based on its acceptance it can be seen as reference for the simulations of the radiated sound energy. It can be seen, that the simulations cope very

good keeping in mind the simplifications used in the monopole synthesis resulting in a much higher computation efficiency.
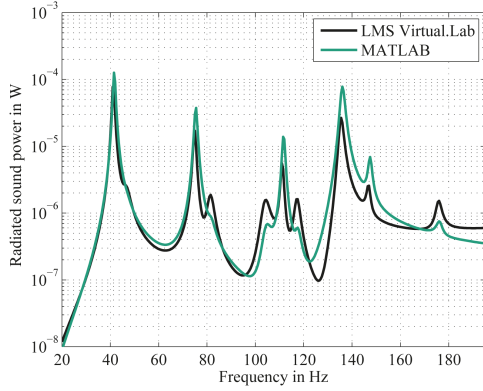

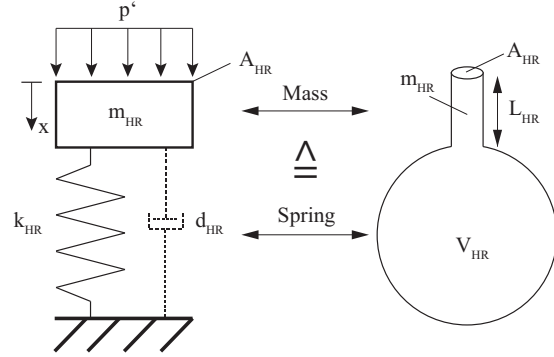Figure 7: Comparison of the in calculated sound radiation

Figure 8: Analogy of a Helmholtz resonator to a mass damper

4 ADAPTIVE METHODS FOR THE REDUCTION OF THE NOISE TRANSMISSION THROUGH DOUBLE-WALLED STRUCTURES

Reducing the transmitted sound power through a double walled structure adaptive measures can be applied on the structural or the fluid domain. The application of structural actions like tuned mass dampers or piezoelectric transducers [13, 15] would interfere with the transparacy of the window. Therefore the action on the acoustic fluid between the two glass plates is preferred. Working on windows with a large air gap between the plates, loudspeakers can be placed inside in terms of active noise control [8]. The used window has a air gap of 8 mm what is inadequate for the use of a loudspeaker in the desired frequency domain. Helmholtz resonators (HR) can be used for the passive reduction of the transmitted sound energy [1, 11, 15]. In the current work the HR is considered as adaptive system with the effecting frequency optimally tuned to the governing sound field through an active change of its geometry [10].

4.1 CONCEPT AND CONTROL OF THE ADAPTIVE HELMHOLTZ RESONATOR

Helmholtz resonators are used for the narrow band reduction of airborne sound. The setup can be devided in a neck and a body section (see Figure 8), whereby the analogy to a mass loaded mass-damper can be used. Here the moving air in the neck can be considered as mass moving in a damped vibration due to viscous effects, whereby the body acts as an air-spring [7, 9]. This way the behaviour can be described due to

$$m_{HR}\ddot{x} = A_{HR}p' - k_{HR}x - d_{HR}\dot{x}, \tag{5}$$

$$m_{HR} = \rho_0 L_{HR}^* A_{HR}, \tag{6}$$

$$k_{HR} = \frac{A_{HR}^2 \rho_0 c^2}{V_{HR}}, \tag{7}$$

$$d_{HR} = A_{HR}\sqrt{2\mu\rho\omega}\left(\frac{L_{HR}}{R_{HR}} + 1\right) \tag{8}$$

with the physical properties of dry air at 20° C listed in Table 2, the opening surface A_{HR}, the volume inside the body V_{HR}, the length L_{HR} and the radius R_{HR} of the neck and the corrected neck length $L_{HR}^* = L_{HR} + \frac{\pi}{2}R_{HR}$ considering the mass outside the opening and inside the body that contributes to the mass inside the neck. Solving Equation 5 results in the Eigen frequencies

$$f_{0,HR} = \frac{\omega_{0,HR}}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{k_{HR}}{m_{HR}}} = \frac{1}{2\pi}\sqrt{\frac{c^2 A_{HR}}{V_{HR}L_{HR}^*}}. \tag{9}$$

One can easily see, that the tuning frequency of the HR can be variated either by changing the mass in the neck or the stiffness in the body. Regarding Equation 8 a change of the neck geometry results in an unwanted alteration of the viscous damping. Therefore the body is considered as cylinder, changing the volume through an axial piston. The resulting Simulink model is pictured in Figure 9. The associated positions of the four HRs are listed in Table 1. The stiffness of the HR here is controlled using a phase estimator keeping the phase of the incoming sound pressure and the signal in the HR at a value of 90° where the HR is tuned [10].
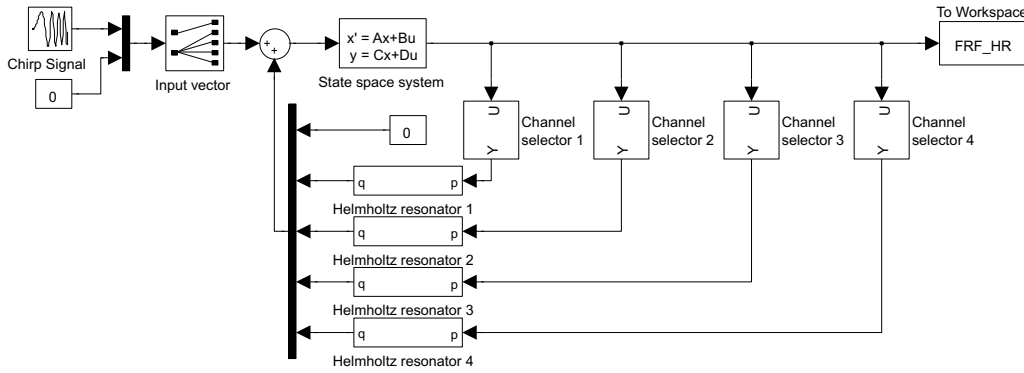

Figure 9: Simlink model of the reduced system with the Helmholtz resonators

4.2 EFFECT OF THE ADAPTIVE HELMHOLTZ RESONATOR ON THE DOUBLE-GLAZED WINDOW

In Figures 10 to 13 the simulated effect of four HRs placed at the positions in Table 1 on the vibrations of the double-glazed window can be seen. It can be observed that the Eigen modes of the glass plates vibrating in opposing phase result in high sound pressure in the air gap. Here the HRs have its most effectiveness. Regarding the transmitted vibrations to the upper glass plate effected by a force excitation at f_{in} with 1 N, it can be stated, that these opposing modes have a lower amount of transmitted energy in Figure 11, based on the higher resistance, that the glass plates experience in this frequencies. This fact can also be observed in the radiated sound energy in Figure 12. Investigating the effect of the HRs on the vibrations of the upper plate and the radiated sound energy, it can be determined that the reduction has it's only effectiveness at the opposing modes, where the resonance amplitudes can be reduced, so that they do not have any effect on the radiated sound energy. Considering the effect on the transmitted sound energy based on the excitation at q_{in} with an amplitude of 1 m³ it can be seen, that the radiated system has a high sensibility for the first and the eighth Eigen mode. Here the

coupling of the cavity inside the acoustics demonstrator and the double-glazed window has its highest sensibility. Unfortunatally the HRs have no significant effect, caused by the equiphase virtue of these modes. But a reduction effect on the fifth Eigen mode can be seen, that also has no small concern to the radiated sound power.

All in all it can be stated that the HRs are valuable adaptive measures for the semi-passive reduction of the transmitted sound energy. The most effectiveness is achieved for modes that are not vibrating in a equiphase virtue based on the high sound pressure resulting in the air gap between the two glass plates. Equiphase modes cause a low sound pressure whereby the HRs have a low efficiency. A larger effect of the HR on the sound transmission through sound proof windows, that have a unsymmetrical setup and therefore no eqiphase Eigen modes, is expected in further investigations.
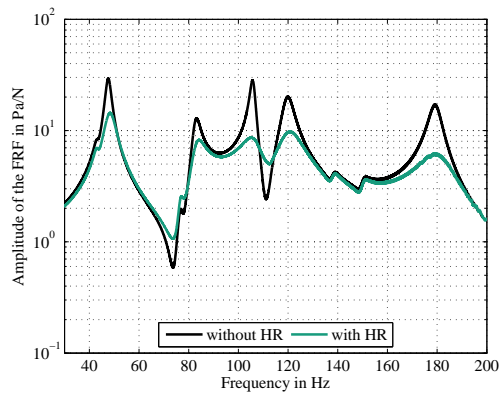

Figure 10: FRF of the force excitation to the sound-pressure in the air gap of the window at p_{out}
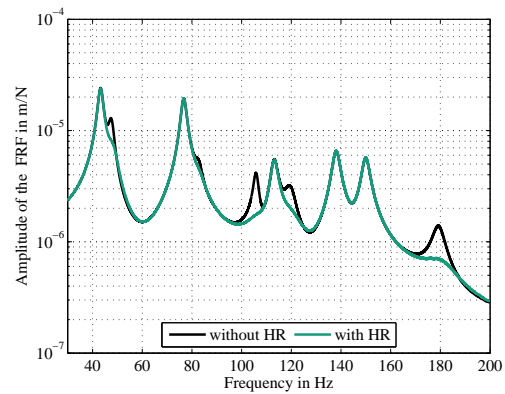

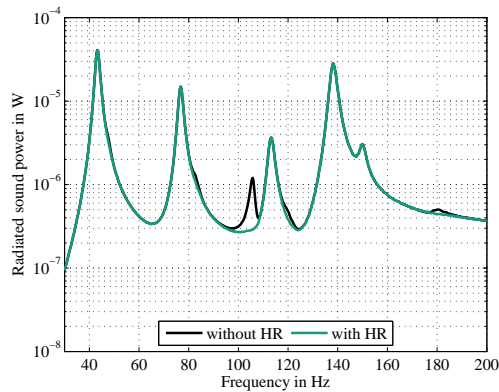Figure 11: FRF of the force excitation to the mean displacement of the upper glass plate

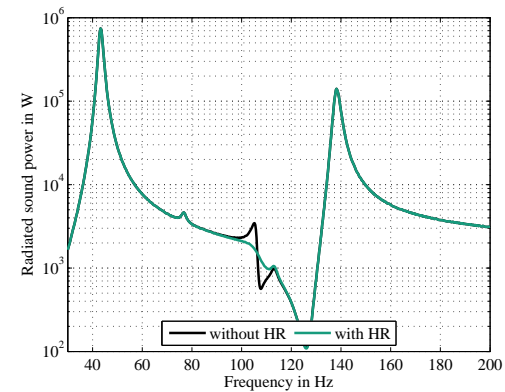Figure 12: Radiated sound power with the force excitation at p_{out}

Figure 13: Radiated sound power with volume excitation at q_{in}

5 ACKNOWLEDGEMENTS

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