SIMOPERA – A RESEARCH PROJECT ON THE SOUND LEVEL REDUCTION IN THE ORCHESTRA PIT OF THE DEUTSCHE OPER BERLIN

A Schlesinger Beuth University of Applied Sciences, Berlin, Germany
JM Kimmich HTW University of Applied Sciences, Berlin, Germany
M Ochmann Beuth University of Applied Sciences, Berlin, Germany
S Frank HTW University of Applied Sciences, Berlin, Germany

1 INTRODUCTION

A research project on the simulation and optimization of the acoustics in large and complex shaped venues is presented. The project was initiated by the orchestra department of the Deutsche Oper Berlin (DOB), who seeks to control the sound pressure levels during performances in the orchestra pit. Excessive sound pressure levels shall be avoided without changing the acoustics in the hall.
In this project measurements were executed as a first step in order to capture the present situation of the DOB closely. Subsequently, wave based models and models of Geometrical Acoustics are fitted to the measurements and are then used for optimization. In the final stages, promising solutions shall be demonstrated by means of opto-acoustic simulation.
In this contribution the authors present recent work in a two year university project, which is named SIMOPERA (simulation and optimization of room acoustical field at the example of the Deutsche Oper Berlin). In earlier publications the authors gave a room acoustic analysis of the DOB and compared it to the work of Cremer et al. [1-3]. Particular interest was spent to the stage parameters. In that respect a comparison with other operas showed no exception. Moreover, a homogeneous distribution of acoustical parameters in the ISO recommended quality ranges of performance venues was verified for the modern acoustic layout of the DOB, which is consistent with public opinion.
In parallel a CAD model of the DOB was created and subsequently converted to the data formats of CATT Acoustic, a software program of geometrical acoustics, and COMSOL, a software program of – among other methods – of wave based acoustics.
Preliminary simulation approved the following challenges in this project. Mere absorption of sound energy in the orchestra pit will directly reduce the loudness of the hall and change the balance in favor of the stage performers to an improper extend [4-5]. Therefore the authors decided to follow the aim of improving the audibility among musicians by means of a higher level of reflected sound energy, thereby fostering a softer and more differentiated ensemble play, as suggested by the research of Schaerer-Kalkandjiev and Weinzierl [6]. Other well-known challenges rest for example in the model fitting of coupled rooms based on a set of available acoustic parameters that are generally not adequate for describing these complex sound fields. Therefore, the authors simulated different boundary conditions and interface definitions between linked rooms. This and the adaption of the wave based models to the measured modal field are topics of further investigation.
At present the authors study acoustical concepts that firstly increase mutual audibility in the orchestra pit and secondly enable a precise balancing of the low frequency sound pressure level in the orchestra pit. The former is based on reflectors that are installed in the proscenium, the latter is investigated by means of Helmholtz resonators. This paper summarizes the latest results in wave based modeling of coupled rooms and gives an initial study on Helmholtz resonators.

2 WAVE BASED ACOUSTIC SIMULATION OF COUPLED ROOMS

Methods based on Geometrical Acoustics as e. g. the raytracing method are suitable in the frequency range above the well-known Schroeder frequency. However, inside the orchestra pit the effective sound field is additionally strongly determined by room modes, oscillating below the Schroeder frequency. These can be modeled by wave-based methods, as e. g. the Finite Element Method (FEM).
A photo of the orchestra pit of the DOB is given in Figure 1. The floor space is about 150 m^2 with 33 m^2 lying under the stage, which is referred to as the overhang area. The overhang area is divided by the prompter's box in two parts. The standard setting of the stage lift is at -2.9 m below the stage. The side walls of the pit perpendicular to the long axis of the opera consist of wooden panels, whereas the smaller side walls are made from concrete. The pit wall directly under the stage and the ceiling of the overhang area are inclined towards the floor in order to avoid axial room modes.

Fig. 1 *The orchestra pit at Deutsche Oper Berlin with the overhang below the stage.*

In earlier studies the authors analyzed sound pressure levels in the decoupled orchestra pit with the FEM in the software package COMSOL (Multiphysics 5.3a) [4-5]. For decoupling the orchestra pit from the auditorium, an air impedance was at first applied to the ceiling of the pit. Recently, this interface layer was replaced be a perfectly matched layer. By such means, the incorrect reflection from only non-planar waves impinging on a cap with air impedance was avoided.
In order to improve computation speed, a combination of FEM in the orchestra pit (with the essential degree of detail) and Boundary Element Method (BEM) in the auditorium was implemented. Such linkage proved however unsuited for the following simulation of eigenfrequencies. Therefore the following study on eigenfrequencies, which shows the necessity of rendering the entire room rather than the decoupled pit, was based on FEM only.
The simulation setups shown in Figure 2 indicate different pressure distributions at close frequency nodes for (B) the pit coupled to the auditorium of the DOB and (C) the pit alone. Both conditions are summarized in the room transfer functions of Fig. 2 (A). Although the wall impedance was, in absence of measurement values, set to be sonically hard this simplification illustrates the rule.

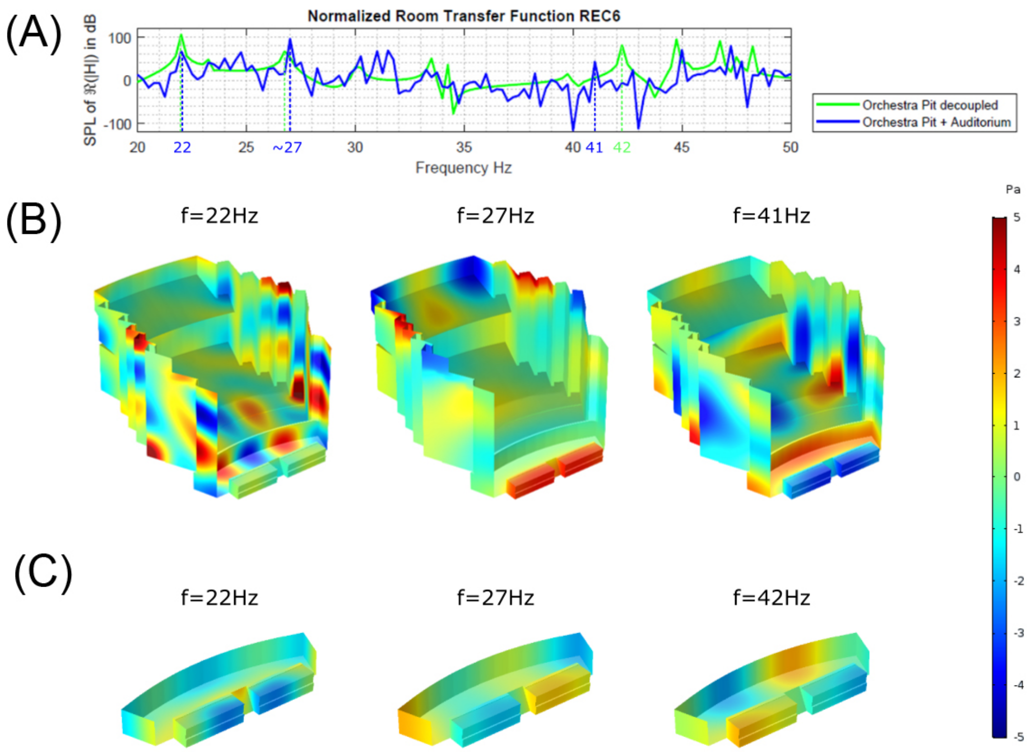


Fig. 2 *(A) Analysis of the room transfer function for the auditorium with orchestra pit (blue) and the decoupled orchestra pit (green). (B) Sound pressure distribution at the walls of the auditorium with orchestra pit and (C) sound pressure distribution at the walls of the orchestra pit, only. (B) and (C) show each three eigenfrequencies of the room.*

3 WAVE BASED ACOUSTIC SIMULATION OF HELMHOLTZ RESONATORS

One of the means to suppress increased sound pressure levels of room modes within the orchestra pit are Helmholtz resonators. In order to include them in the FEM model, a preliminary analysis was executed. A Helmholtz resonator consists of a fluid filled combination of a volume and an open cavity. By describing the fluid in the open cavity as a mass and the fluid in the volume as a spring [7], one obtains a resonating mass-spring system the resonance frequency f_R

$$f_R=\frac{c}{2\pi}\sqrt{\frac{S}{(l+2\Delta l)V}}, \tag{1}$$

where S is the cross section of the open cavity, for a circular neck $S=\pi r^2$, with r the radius and l the length of the neck. V is the volume of the resonator and Δl the end correction, which is $\Delta l=\gamma r$ with the end correction factor $\gamma=0.85$ [8].
However, this derivation does not take thermoviscous losses for wave propagation in narrow tubes into account, which can be significant for low frequencies.

A general consideration of thermoviscous acoustical problems makes use of the linearized Navier-Stokes-Equation for the temperature, velocity and pressure. Their numerical solution is computationally intensive. When treating viscothermal acoustic problems in tubes, the low reduced frequency model [9-10] can be used under specific assumptions: firstly the viscous and thermal boundary layer thickness must be much smaller than the acoustic wavelength. The viscous boundary layer thickness δ_v and the thermal boundary thickness δ_h are in accordance with [11]

$$\delta_h = \sqrt{\frac{2\mu}{\omega\rho}} \quad \text{and} \tag{2}$$

$$\delta_v = \sqrt{\frac{2k}{\omega\rho C_p}} \, , \tag{3}$$

where μ is the dynamic viscosity, ρ the density, k the thermal conductivity and C_p the heat capacity at constant pressure. When acoustical waves propagate through a tube, losses depend on the frequency and occur when the dimension of the cross-section of the tube tend to the thermal and viscous boundary layer thickness. The frequency dependence of both quantities is shown in Figure 3.

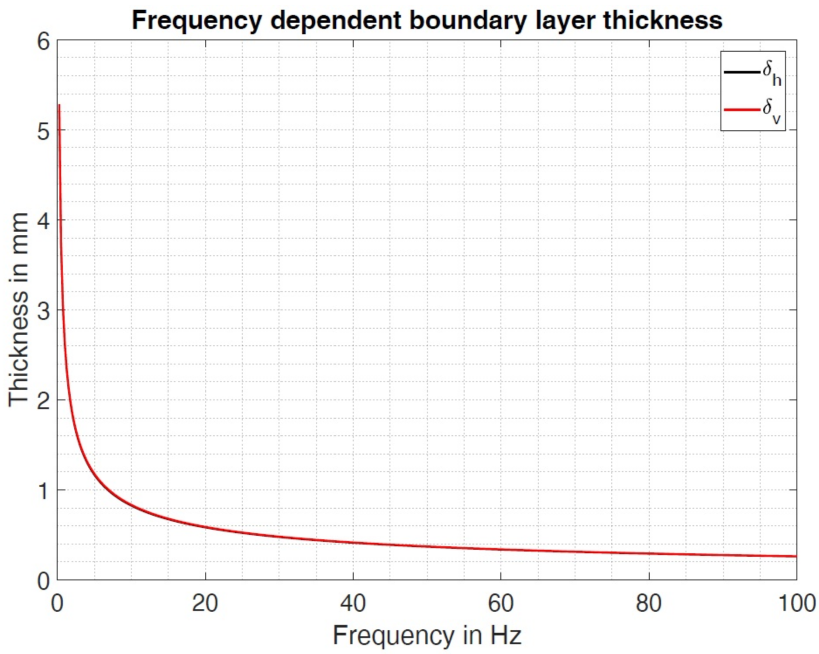


Fig. 3. *Frequency depended boundary layer thickness in air: The black and red lines show the thermal and viscous boundary layer thickness respectively.*

For example in air where $\delta_h \approx \delta_v \approx 0.8\,mm$ the ratio for a tube of 1 cm radius is $\delta / r = 12.5$. As shown in [12], for air the first assumption holds in the whole audible frequency range. Secondly, the cross section of the tube must be much smaller than the acoustic wavelength. This assumption holds for the tube dimensions normally in the order of $r = [0.01 \ldots 0.1\,m]$ in Helmholtz resonators and frequencies below 150 Hz where $r / \lambda = 0.1\,m / 0.2\,m = 0.05 \ll 1$.

In COMSOL Multiphysics 5.3a a Narrow Region Acoustics module is available to treat the above mentioned cases. For comparing sound propagation within a Helmholtz resonator with and without losses the model shown in Figure 4 was built. This Helmholtz resonator offers the simple geometry of a cube with side length $a=0.38\,m$, a neck consisting of a circular tube with varying radius r and length l. For the case of losses in the tube the Narrow Region Acoustics module of COMSOL for a circular duct was applied to the green region in Figure 4. On the red surface a pressure boundary condition $p=1\,Pa$ is applied for excitation.

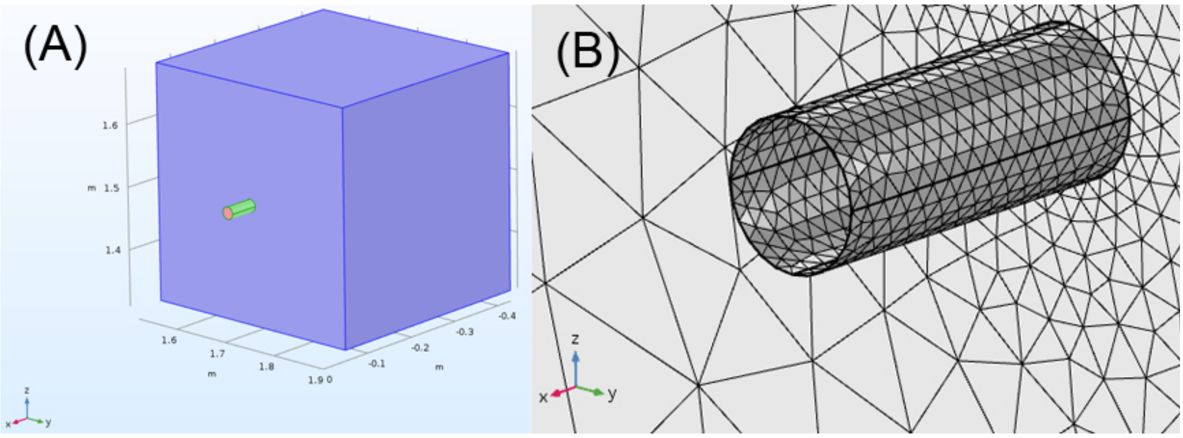


Fig. 4. *(A) Domains of the FEM model of a Helmholtz resonator in COMSOL. The blue area is the volume of the Helmholtz resonator, the green area the neck consisting of a circular tube. On the red surface a pressure boundary condition* $p=1\,Pa$ *is applied for excitation. In the green neck area the Narrow Region Acoustics module for circular duct is used. (B) Meshing in the neck area with finer elements for the neck and the orifice and growing element size for the volume of the resonator.*

For evaluating the effectiveness of the simulated resonator, the normalized average sound pressure level in the resonator volume was calculated with

$$\langle p\rangle=\frac{1}{V}\int p^2\,dV \quad \text{and} \tag{4}$$

$$L_p=20\,lg\left(\frac{\langle p\rangle}{p_0}\right), \tag{5}$$

with $p_0=1\,Pa$. In addition the eigenfrequencies were calculated numerically. For radii varying from 0.01 to 0.1 m the results are depicted in Fig. 5.

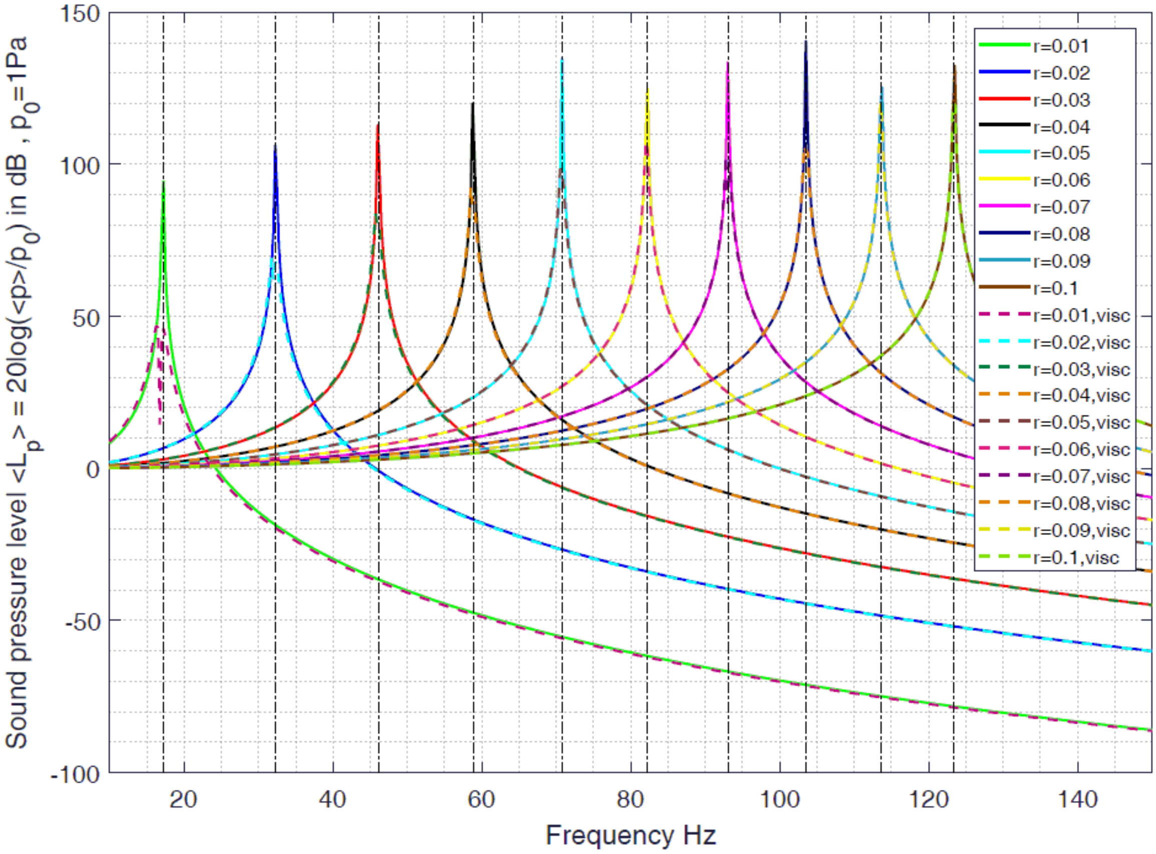


Fig. 5. *Averaged sound pressure levels for the volume of Helmholtz resonators without (solid dotted lines) and with (dotted, colored lines) thermoviscous losses with varying neck radii from 0.01 to 0.1 m and constant neck length of* $l=0.05\,m$ *and Volume* $V=0.055\,m^3$ *. The vertical black dotted lines show the eigenfrequencies.*

The results of this study show that firstly the calculated eigenfrequencies match the calculated resonance peaks well, secondly good agreement between both models exists for radii from 0.1 to 0.09 m. In case of radii from 0.08 to 0.02 m the simulation models with losses show a drop of the resonance peak of about 30 dB. In case of 0.01 m the deviation rises to about 50 dB between both models.

When varying the length of the resonator neck with constant radius $r=0.02\,m$ one observes in Figure 6 a deviation for the entire series of eigenfrequencies between the models. The resonance curves with losses are shifted towards lower frequencies as compared to the models without regard for losses.

However, with both models, the resonance frequencies do not match with Equation 1. While this can be corrected for the variation in terms of the radius, using a different end correction factor, there is no correction for the length mismatch. These difference are subject to further investigation.

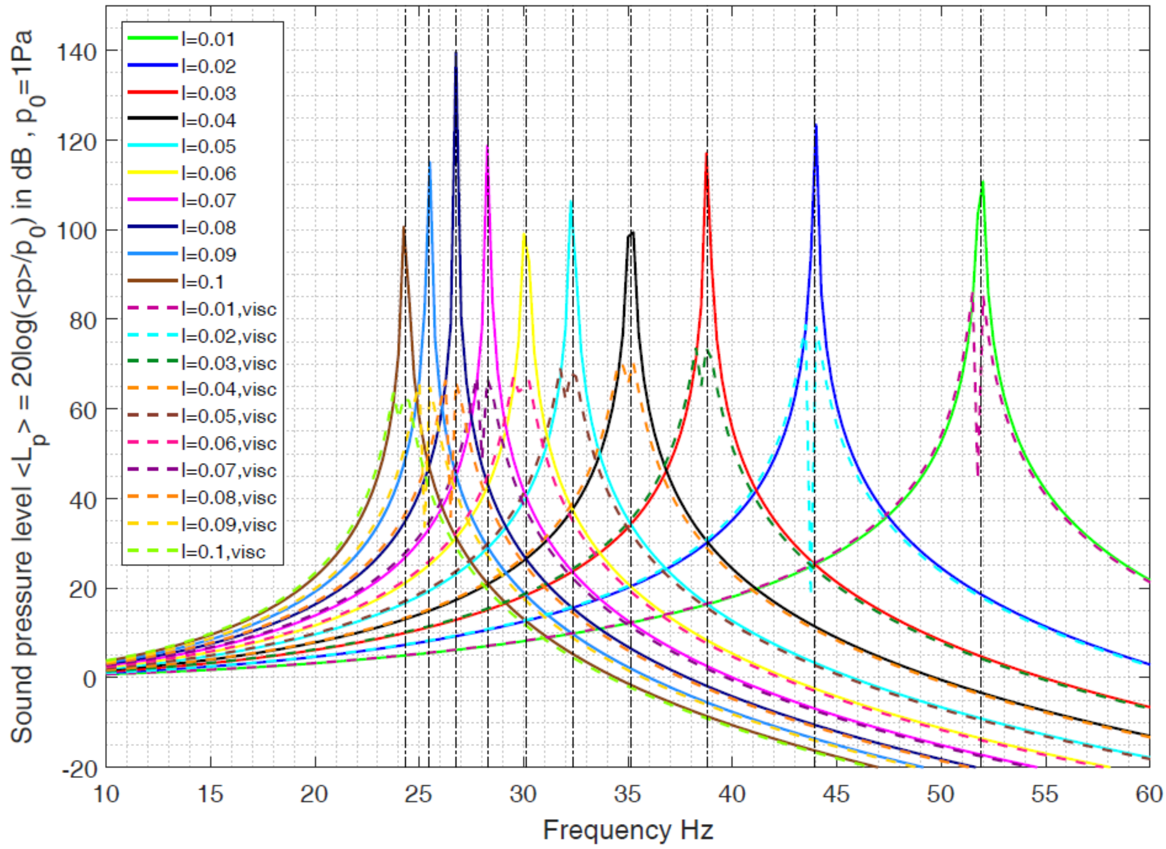
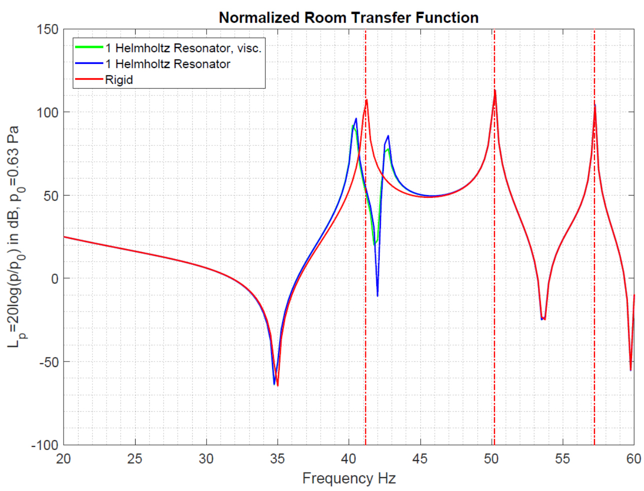


Fig. 6. *Averaged sound pressure levels for the resonator volume of Helmholtz resonators without (solid colored lines) and with (dotted, colored lines) thermoviscous losses with varying neck length from 0.01 to 0.1 m with constant neck radius of 0.02 m and a volume of 0.055 m$^{3.}$ The vertical black dotted lines show the eigenfrequencies.*

Finally, we apply a Helmholtz resonator to a test room. When coupled to a model room, one can observe the effect of the Helmholtz resonator tuned to the first axial room mode in Figure 7. The room is excited by monopole point source with $L_p(1m)=90\,dB$ under free field conditions. The room transfer function is normalized to this value.

Fig. 7. *Normalized Room Transfer Function with and without the Helmholtz resonator coupled to the cuboid test room. The room dimensions are 4.17, 3.42 and 3 m, respectively, with walls being sonically hard. The first axial mode in x-direction lies at $f=41.15\,Hz$.*



4 CONCLUSION AND FUTURE WORK

In this paper an analysis on eigenfrequencies with wave based simulation was given. It was illustrated that the simulation cannot be executed in a decoupled room alone, here the orchestra pit. In the second part, the modeling of Helmholtz resonators based on direct coupling with a FEM model was shown. A comparison between lossless resonators and such with thermoviscous losses in the neck, modeled with the Narrow Region Acoustics module in COMSOL, revealed differences for radii smaller than 0.02 m. These differences need to be further investigated. Therefore simulations without simplifications for sound propagation in the neck using COMSOL's thermoacoustic module are intended. Furthermore, a comparison with equivalent circuit models for calculating impedance boundary conditions for resonators are going to be executed.
The aim of these studies is to better understand the effect of Helmholtz resonators and their adjustment for applications in room acoustics. Once the inner workings of Helmholtz resonators can be fully rendered with FEM, it is planned to increase their frequency range of absorption by means of porous material in the neck or the volume [13]. Subsequently, the authors intend to study the behavior of arrays of equal and unequal Helmholtz resonators and to apply them to the model of the orchestra pit.

5 REFERENCES

1. Cremer, L., Nutsch, J. and Zemke, H.J. (1962) Die akustischen Maßnahmen beim Wiederaufbau der Deutschen Oper Berlin. Acustica 12, 428-433
2. Schlesinger, A., Kimmich, J. M., Ochmann, M., Frank, S. (2018), Messtechnische Untersuchung der Raumakustik der Deutschen Oper Berlin. In: Fortschritte der Akustik - DAGA 2018, München, Deutschland, 2018.
3. Schlesinger, A., Ochmannn, M., Kimmich, J. M., Frank, S. (2018), Controlling multiple acoustic objectives during the implementation of the European directive 2003/10/EC at an opera house. In: Euronoise 2018 Conference Proceedings, Heraklion, Greece
4. Kimmich, J.M., Frank, S., Schlesinger, A., Ochmann, M. und Tschaikner, M. (2018), Berechnung des Schallfeldes in der Deutschen Oper Berlin mit Raytracing und der Finiten Elemente Methode. Tagungsband der DAGA 2018, München, Deutschland, 2018.
5. Kimmich, J.M., Schlesinger, A., Tschaikner, M., Ochmann, M. Frank, S., Acoustical Analysis of coupled rooms applied to the Deutsche Oper Berlin. Proceedings of the Polish-German Structured Conference on Acoustics, Ustka, Poland, 2018.
6. Schaerer Kalkandjiev, Z., Weinzierl, S. (2013), The influence of room acoustics on solo music performance: An empirical case study. Acta Acust. united with Acust., 99(3), pp. 433 - 441
7. D. T. Blackstock, Fundamentals of Physical Acoustics, New York –- Wiley, 2000.
8. T. J. Cox and P. D'Antonio, Acoustic Absorbers and Diffusers, 2nd ed., Abingdon: Taylor & Francis, 2009.
9. M. R. Stinson, "The propagation of plane sound waves in narrow and wide circular tubes, and generalization to uniform tubes of arbitrary cross-sectional shape," J. Acoust. Soc. Am. 89(2), February 1991, pp. 550–558, 1991.
10. H. Tijdeman, "On the Propagation of Sound Waves in Cylindrical Tubes,", J. Sound Vib. 39(1), 1975, pp. 1–33
11. P. M. Morse and K. U. Ingard, Theoretical Acoustics, 1st Princeton University Press ed., Princeton: Princeton University Press, 1986.
12. W. R. Kampinga, Viscothermal acoustics using finite elements – Analysis tools for engineers, Ph.D. dissertation, University of Twente, Enschede, The Netherlands.
13. A. Selamet, M. B. Xu, I.-J. Lee and N. T. Huff, "Helmholtz resonator lined with absorbing material," J. Acoust. Soc. Am. 117(2), February 2005, pp. 725–733, 2005.