Numerical analysis of thin-walled component structure

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Abstract. Thin-walled components are commonly employed in the broadly understood engineering applications. From building construction, to machine building and acoustic system components. The design calculations for this type of structures are complex and time consuming. The paper presents an MES analysis of a thin-walled component in the example of a speaker membrane. The aim of the analysis is to determine the deformation characteristics of the membrane as a function of its deflection by the voice coil. The examined component is constructed from two materials, which was reflected in the model. The paper presents the methodology of building the MES model, modeling of selected materials forming the structural composition of the membrane as well as interpretation of results. The contents provided in the paper may be of use for the purpose of determining the correct construction of these components in order to improve their working efficiency.

1. Introduction

The principle of operation of a speaker, i.e. an electro-acoustic transducer involves directing a current to the speaker coil which is placed in a magnetic gap between the pole piece and the pole piece plate. This leads to the electromagnetic field generated by the coil to be superimposed on the constant field of a magnet. This causes the coil wire to be affected by force which sets it to motion together with the membrane, which causes a change in acoustic pressure and the emission of sound [1, 2].

The main requirements for speaker membranes are to ensure simultaneous:

- lightness,
- stiffness.

These two requirements are contradictory, which presents a constant challenge to the speaker designers to achieve an endless compromise between the lowers possible weight and stiffness.

In order to achieve such compromise, various materials are utilized for membrane materials, e.g. [1, 2]:

- paper,
- polypropylene,
- Kevlar,
- aluminum,
- a combination of the above (as a multilayer structure).

From the point of view of mechanics, to increase membrane stiffness, additional surface folds and embossing are used [1, 2].
The above solution is the result of an assumption that the membrane moves as an inflexible piston, i.e. reciprocal motion with equal vibration amplitude on the entire surface. However, in practice it turns out that bending waves spread in the membrane material, from the drive (vibrating coil) towards the external edge, where they reverberate and are superimposed on incoming waves [1, 2].

As a result, a part of the membrane shifts in one direction, and simultaneously another part of the membrane shifts in the opposite direction. Figure 1 demonstrates the shifting membrane modes depending on the direction of vibration propagation. Resulting from the above, we observe a decay in sound emission and an uneven acoustic pressure frequency characteristic [1, 2].

![Membrane zones vibrating with opposite phases](image)

This can be mitigated by stiffening the membrane by [1, 2]:

- changing its surface shape,
- selecting a suitable material,
- additional surface coating with suitable material,
- utilizing a multi-layer structure.

The authors of this article focused on examining the working characteristics of the speaker membrane in the factory-made variant in order to modify the membrane construction in the future in order to improve the frequency characteristic of the examined speaker so as to improve sound quality and expand its acceptable range.

2. Methodology of the research

Figure 2 presents an assembly of selected speaker components to be studied. For the purpose of calculations, the finite element method implemented in the Abaqus software was utilized. It is software that is used when modeling and analyzing many engineering issues [3–5]. The three major constituents of the speaker were modeled: surround, cone and rigid sleeve. Figure 3 presents a 3D model together with a FEM model of the analyzed speaker components. The surround component is made of low density polyurethane foam, the cone is cardboard, whereas the rigid sleeve is made of an aluminum alloy. There are many different models implemented in the Abaqus software which can be used for modeling the behavior of low density polyurethane foam and similar materials under external load, for example Ogden’s models [6]. In the considered example, the influence of membrane (cone) displacement on the deformation of the surround component is not the object of study, but the deformation of the membrane itself as a result of the force of inertia during its displacement by the speaker coil. Additionally, the dimensions of the surround component allow the membrane to move out freely without affecting its deformation. Based on the above, it was decided to simplify the material model for the surround component and utilize a regular flexible model with parameters $E = 2.6$ MPa and $\nu = 0.4$. The membrane material (cardboard) exhibits anisotropic characteristics, it is therefore necessary to assign correct characteristics in each direction. Table 1 presents parameter values to characterize the parameters for cardboard material. The rigid sleeve was modeled as. The basis of this assumption is the significantly higher rigidity of the material used to manufacture this component in comparison to the other materials. For the purpose of calculations, C3D8R elements were used (an 8-node linear brick, reduced integration, hourglass control), care was taken to ensure that at least 5 elements are present on the thickness of each assembly component. The density of membrane material was set as 450 kg m\(^{-3}\).
Figure 2. An assembly of selected speaker components for analysis, diameter 150 mm: 1 – surround, 2 – cone, 3 – rigid sleeve.

Figure 3. An assembly of selected speaker components for analysis, diameter 150 mm, a) 3D model, b) FEM model, 1 – surround, 2 – cone, 3 – rigid sleeve.

Figure 4 presents two extreme positions of the membrane. The state of deformation of the membrane under forced displacement by the sleeve was examined. The sleeve moved by 10 mm during the time of the coil input with set frequency. The examined frequency range is 45 Hz – 1750 Hz. The displacement of points P1,5 was checked after every movement/deformation of the membrane.
Table 1. Properties of material used for modeling of the membrane [7].

|   | \( E \) (GPa) | \( G \) (MPa) | \( \nu \) (-) |
|---|---|---|---|
| 1 | 8.25 | 1890 | 0.43 |
| 2 | 2.9  | 7    | 0.01 |
| 3 | 2.9  | 70   | 0.01 |

Figure 4. Parameters of analysis: maximum membrane displacement -10 mm, \( P_{1.5} \) – points in which membrane displacement was measured.

3. Results of the FEM analyses
Figures 5–12 demonstrate the deformation state of the membrane resulting from the displacement in time matching the frequency, together with the displacement of measuring points 1–5 as a function of sleeve displacement.

Figure 5. Membrane deformation state with displacement of measuring points during membrane motion in time matching the frequency of 45 Hz.

Figure 6. Membrane deformation state with displacement of measuring points during membrane motion in time matching the frequency of 150 Hz.
Figure 7. Membrane deformation state with displacement of measuring points during membrane motion in time matching the frequency of 400 Hz.

Figure 8. Membrane deformation state with displacement of measuring points during membrane motion in time matching the frequency of 500 Hz.

Figure 9. Membrane deformation state with displacement of measuring points during membrane motion in time matching the frequency of 800 Hz.

Figure 10. Membrane deformation state with displacement of measuring points during membrane motion in time matching the frequency of 1000 Hz.
Figure 11. Membrane deformation state with displacement of measuring points during membrane motion in time matching the frequency of 1750 Hz.

Figure 12. The displacement of measurement points as a function of frequency determining the time of the coil sleeve displacement: a) $P_1$, b) $P_3$, c) $P_5$.

4. Conclusion
Based on the results for the frequency higher than 250 Hz, the displacement of the points $P_1$-$P_5$ differs from the constant kinematic distortion of 10 mm due to the membrane deformation. In result the velocity vector distribution along the edge of the membrane axial cross-section is non-uniform, which leads to the divergence between the resultant velocity vector of the body and the desired one. Since the velocity of the points is connected with the frequency of the vibrating loudspeaker membrane, the sound generated for it will have lower frequency. This induces that using stiffer materials for the membrane should lower the power loss and improve the sound quality and the bandwidth of the speaker.

On the other hand increasing the spring constant $k$ by either changing the material or the geometry of the membrane, will affect in nonlinear growth of its natural frequency $\omega_0$. Since there is a negative correlation between the Q-factor of the speaker and the natural frequency and the decrease of the Q-factor will increase the vibration energy loss, this approach is not advantageous. The Q-factor can be increased by damping constant $\beta$. Additionally the viscoelastic properties of the membrane material will
increases the resistance of the material along with the frequency growth since there is a linear correlation between the strain rate $\dot{\varepsilon}$ and the generated stress. It means that in order to maintain the good sound quality, the compromise between the stiffness and the damping of the membrane has to be found.

The last parameter of the membrane which should be considered in such analysis is its mass. In dynamic analysis of the vibration of the membrane the mass increase for more rigid construction will at some level compensate the natural frequency growth caused by the higher stiffness. However if we analyze the deformation of the membrane, the elastic buckling of the membrane is caused by the inertia force, which rise along with mass or acceleration. Since for the vibrating motion the peak acceleration is in quadratic function of the frequency ($a \sim f^2$) the mass effect will be much less important than the compromise between the stiffness and damping of the membrane.

The methodology of study presented in this work may be utilized at the preliminary stage of research to improve the acoustic properties of speaker membranes. The proposed model can be employed to determine the state of membrane deformation, which allows to calculate the displacement characteristic of the membrane and estimate the behavior characteristics of a given membrane shape and structure undergoing displacement in a given time at an early stage of study. The presented results allow to make the decision about which areas of the membrane should be reinforced to facilitate its full outward motion without excessive deformation. The examined membrane demonstrates a tendency for deformation in the area of the lower connection with the rigid sleeve. The development of the presented model will contribute to a more accurate representation of membrane component behavior characteristics under specific forced displacement.

5. References
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