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Piezoelectric two-layer plate for position stabilization

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Abstract

Numerous vibrating electromechanical systems lack a rigid connection to the inertial frame. An artificial inertial frame can be generated by a shaker, which compensates for vibrations. In this article, we present an encapsulated and perforated unimorph bending plate for this purpose. Vibrations can be compensated up to the first eigenfrequency of the system. As basis for an efficient system simulation and optimization, a new three-port multi-domain network model was developed. An extension qualifies the network for the simulation of the acoustical behavior inside the capsule. Network parameters are determined using finite element simulations. The dynamic behavior of the network model agrees with the finite element simulation results up to the first resonance of the system. The network model was verified by measurements on a laboratory setup, too. Furthermore, the network model could be simplified and was applied to determine the influence of various parameters on the stabilization performance of the plate transducer. The performance of the piezoelectric bending plate for position stabilization had been in addition investigated experimentally by measurements on a macroscopic capsule.

Keywords

Modeling, unimorph, network methods, finite element methods, electromechanical transducer

Introduction

It is common for freely vibrating electromechanical systems, for example, cameras or hard disk drives, to use additional actuators to keep them in position (Chiu et al., 2007; Sasaki et al., 2012). Widely used actuation principles involve cantilever beams, electrostatic comb structures, voice coils, or piezo blocks. In this article, a new approach is presented, where a perforated, piezoelectric unimorph bending plate—shown in Figure 1—is used as stabilization unit. In precision and microsystems engineering, such unimorph plates are used for transduction of volume flows or pressures. Other applications can be found in medical devices.

In this article, a compensation arrangement is investigated and optimized where a piezoelectric bending plate is fixed in a rotational symmetric capsule providing two cylindrical cavities V_1 and V_2 in the front and rear of the unimorph. A perforation of the bending plate allows an air exchange through the plate in order to achieve acoustic transparency. The encapsulation protects the plate from environmental influences. This way, the stabilization unit can be easily attached to the system to be stabilized, too.

The working principle of the stabilization unit relies on an acceleration force. This force $\underline{F}_c$ is induced into

the capsule when a voltage $\underline{V}_{SU}$ is applied at the piezoelectric element such that the unimorph is bending. A concentrated mass m_{add} , which is attached to the middle of the bending plate, increases the force and thus the performance of the stabilization unit. The magnitude of $\underline{F}_c$ depends on the geometry of the stabilization unit, the mass of the capsule, the attached mass, and the operation frequency. A system attached to the capsule can be stabilized based on the force into the capsule and the resulting velocity $\underline{v}_c$, respectively, when the excitation voltage is controlled based on the measured acceleration $\underline{a}_c$ of the capsule and a control function $\underline{x}$. The system exhibits resonances, which limit the applicable frequency range.

One goal of the presented work was to improve the capsule performance based on behavioral simulations.

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Figure 1. Schematic cross-sectional view of the stabilization unit inside the axisymmetric capsule and attached impedance of the system to be stabilized (left) and 3D view of the stabilization unit (right).
3D: three-dimensional.

In comparison to finite element (FE) models, network models consume less computational time for simulations in the frequency domain and time domain. Therefore, network simulations can be utilized for an effective prediction of the functional parameters of the stabilization unit. The simplicity of the network model is paid with less accuracy in comparison to a FE model, but it is a prerequisite to an efficient system optimization and to an understanding of the physical phenomena. The chosen optimized variant can be conclusively simulated with FE to refine the design. Actually, no network model is known of the piezoelectric unimorph plate with respect to the electrical, mechanical, and acoustic system properties, which are required for dynamic analyses of the arrangement in Figure 1. In the following sections, the dynamic network model of the stabilization unit is developed and applied to improve its performance. Thereby, the network has to take account of electrical, mechanical, and acoustic properties of the stabilization unit because an applied electrical voltage at the piezoelectric element causes a mechanical displacement as well as an acoustic difference pressure $p_d = p_2 - p_1$ across the unimorph. The mechanical and electrical properties are influenced by the perforation, as well. Therefore, a good acoustic behavior of the unimorph comes with the expense of the electromechanical stabilization performance. Nevertheless, FE simulations were involved to determine network parameters following the approach of a *combined simulation*. Additionally, the network model was verified by measurements on a laboratory setup. Measurements were also taken from a macroscopic capsule to determine the performance of the piezoelectric bending plate for position stabilization.

Approach for a multi-domain network model of the stabilization unit

In this section, a new approach for a three-port multi-domain network model of the stabilization unit is presented. An extension of the network also takes account of the perforation of the bending plate and cavities inside the capsule.

The network model requires system points and involved physical flow and differential quantities. Figure 1 provides an overview of all quantities and system points (1, 2), too, which will be considered by the network model. The physical coordinates can be grouped into electrical, mechanical, and acoustic ones. Electrical coordinates are the voltage V_{SU} and current i at the piezoelectric element. Since the stabilization unit operates as an electromechanical actuator, the piezoelectric element represents an electrical source. The capacity C_b of the piezoelectric element can be measured in the blocked mechanical ($v_c = 0$) and open acoustic case ($q = 0$). Mechanical coordinates are the velocities v_m and v_c as well as the forces F_m and F_c in the middle of the bending plate and the capsule (see Figure 1). The mechanical differential and flow quantities are aligned in the same direction. The acoustic difference pressure p_d across the bending plate and the acoustic volume flow q are aligned in opposite directions. Whereas the volume flow is as positive directed as the mechanical parameters, the difference pressure is defined in the opposite direction. Thus, a positive volume flow causes a positive difference pressure.

Model of the piezoelectric bending plate

While capsule and additional mass can be described as concentrated mechanical masses m_c and m_{add} , the

Figure 2. Multi-domain network model of the unimorph including capsule mass and attached mass.

network model of the piezoelectric bending plate is more complex. Aspects of bending plates and piezoelectric disks have been already discussed by Lenk et al. (2010). There, a bending plate is modeled as mechanoacoustical transducer where mechanical and acoustical domains of this network are connected by the effective piston area A_{eff} of the bending plate, that is, the lower right transducer in Figure 2. The connection between the electrical and mechanical domains has been realized by the electromechanical transducer of the disk, which is represented by the lower left transducer in Figure 2. The remaining (upper) transducer describes the electroacoustic network approach for a piezoelectric unimorph bending plate, developed by Starke et al. (2011a) and Prasad et al. (2002). However, the existing models of Lenk et al. or Starke et al. (2011b) can only simulate electromechanical or electrical-acoustical interactions at a time. Hence, these previous network approaches of the piezoelectric bending plate cannot be utilized on its own for simulating the dynamical behavior of the arrangement in Figure 1 in all three physical domains simultaneously, but a combination of the known models is needed, as shown in Figure 2.

This model connects all the existing approaches to a three-port network model of the piezoelectric bending plate with respect to the electrical, mechanical, and acoustic domains. The model is valid up to the first mechanical and acoustical resonance frequencies. Nevertheless, the one-dimensional network describes the physical behavior only in the direction of the symmetry axis. In comparison to the existing network models, the network in Figure 2 is suited for simulating electrical, mechanical, and acoustical reactions at the

same time. Furthermore, the developed network simulates the acoustical reaction in the case of an applied voltage at the piezoelectric element even if the mechanical system points 1 and 2 in Figure 1 are fixed. These improvements are necessary to achieve the requirement of simulating all physical interactions of the encapsulated piezoelectric bending plate.

In the mechanical domain, the network consists of the compliance n_K (m/N) as well as the effective mass fractions m_m (kg) and m_e (kg) of the bending plate. The quality factor of the plate can be adjusted by impedance h_{bp} (s/kg). A typical quality factor is $Q = 80$. Furthermore, the mechanical part of the network model contains the masses of the additional body m_{add} (kg) and the capsule m_c (kg). The acoustical series resonance circuit of the model includes compliance N_{ac} (m^3/Pa), effective mass M_{ac} (kg/m^4), and impedance Z_{ac} ($\text{kg}/\text{s m}^4$), which influences the quality factor, in addition to h_{bp} . The reciprocal connections between the three domains of the network are realized by the transduction coefficients Y ($\text{m}/\text{A s}$), A_{eff} (m^2), and X (Pa/V). Hence, the equations in the gray boxes in Figure 2 describe the transfer characteristics.

Acoustical network of the stabilization unit

The possibility of an air exchange between the cavities of the capsule through the holes inside the bending plate assures lower acoustic pressures $\underline{p}_1$ and $\underline{p}_2$ for displacements $\underline{\xi}_m$ of the stabilization unit. A high difference pressure $\underline{p}_d$ operates as an additional load on the unit and could decrease its performance. In comparison to a bending plate without perforation, the network in

Figure 3. Extension of the network of the stabilization unit with respect to the acoustic behavior of the holes inside the bending plate and the air cavities of the capsule.

Figure 2 has to be extended in order to regard the air exchange between the cavities. The partial network considering the acoustical behavior inherent in the capsule is shown in Figure 3.

The elements $N_{ac,V1}$ (m^4/Pa) and $N_{ac,V2}$ model the compliant character of the air volumes inside the capsule. Furthermore, $M_{ac,V1}$ (kg/m^4) and $M_{ac,V2}$ describe the moving air masses of the front and rear side of the bending plate. The parameters that allow pressure equalization and a volume flow between the air cavities are acoustic mass $M_{ac,h}$ (kg/m^4) and acoustic damping $Z_{ac,h}$ ($\text{kg}/\text{s m}^4$).

FE models for the derivation of the network elements

Some of the network elements in Figures 2 and 3 can be calculated analytically only with high effort. Therefore, the network elements were determined via FE simulations. The method of deriving network parameters from FE simulations is known as combined simulation (Starke et al., 2011a). The boundary conditions of these simulation experiments are determined by hand of the network model itself. By setting mounting, shortcircuits and excitation network elements can be separated selectively. These boundary conditions are assigned to the FE model. Figure 4 gives a detailed view of the FE model, which has been implemented in the Workbench environment of ANSYS® as basis for the combined simulation. Selected network parameters are determined exemplarily next.

Electromechanical FE model of the bending plate

Several network parameters can be determined by FE simulations of the static behavior of system components. Additional modal analyses are necessary to calculate the parameters of resonating system parts. For example, the masses m_m and m_e are derived from modal analyses.

Figure 4. Section of the electromechanical finite element model of the piezoelectric bending plate (Steinert, 2012).

An example for a static ($f = 0$) simulation experiment is the derivation of the acoustic compliance N_{ac} . As Figure 2 shows, the element can be isolated obviously by fixing the mechanical system points ($v_m, v_e = 0$). Also, the piezoelectric element has to be shortcircuited ($V_{SU} = 0$). After setting these boundary conditions, $N_{ac} = V_L/p_d$ can be determined by applying an acoustic pressure p_d and calculating the resulting static volume flow V_L of the bending plate. The resonance frequency $f_{0,ac}$ of a modal analysis with the same boundary conditions gives the acoustic mass element $M_{ac} = [(2\pi f_{0,ac})^2 \cdot N_{ac}]^{-1}$. All the other network elements can be determined in a similar way with adapted boundary conditions and load.

Figure 5 shows magnitude and phase angle of the volume flow q of the bending plate in the case of an electrical excitation. The diagram shows a good agreement of the network and FE solution up to the first resonance of the system. Differences between both simulation methods until the first resonance are due to different damping coefficients in the network and FE simulation.

Acoustic FE model

For the determination of the acoustic parameters in Figure 3, several dynamic FE simulations with different

Figure 5. Velocity response in the middle of an exemplary bending plate with a radius of 30 mm simulated with network and FE methods (Steinert, 2012).
FE: finite element.

Figure 6. Acoustic difference pressure response divided by the volume flow inside the air cavities with a radius of 30 mm simulated with network and FE methods (Steinert, 2012).
FE: finite element.

air cavity volumes and hole sizes and numbers were accomplished. The simulations show that the dominant acoustic element is the moving air mass $M_{ac,h}$ inside the holes. Based on the results, an equation was derived, which calculates approximately the parameter according to the geometry of the piezoelectric bending plate (Steinert, 2012)

$$M_{ac,h} \approx \rho_{air} \cdot \frac{2 \cdot 0.428}{\pi \cdot r_{pl}} \cdot \left[N_h \cdot \left(\frac{r_h}{r_{pl}} \right)^2 \right]^{-0.383} \quad (1)$$

where ρ_{air} is the density of the air, N_h is the number of the holes inside the bending plate, r_h is the radius of the holes, and r_{pl} is the radius of the bending plate. The acoustic compliances of the air cavities can be calculated by the equation (Lenk et al., 2010)

$$N_{ac,V1/2} = \frac{\pi \cdot r_{pl} \cdot l_{V1/2}}{\kappa \cdot p_0} \quad (2)$$

where $l_{V1/2}$ is the length of the cavities and p_0 is the normal pressure. The FE simulations confirm that the relation is also valid for a bending plate with holes. For the moving air masses of the front and rear side of the plate, a simulation-based equation could be developed, too. These can be described approximately by the equation

$$M_{ac,V1/2} \approx \frac{32 \cdot \rho_{air}}{15 \cdot \pi^2 \cdot r_{pl}} \quad (3)$$

The acoustic damping of the holes $Z_{ac,h}$ becomes relevant for small holes with a height of the bending plate $h_{pl} \gg r_h$. In most cases, the damping (Lenk et al., 2010)

$$Z_{ac,h} \approx \frac{8 \cdot \mu \cdot h_{pl}}{N_h \cdot \pi \cdot r_{pl}^4} \quad (4)$$

is negligible small in comparison to the moving acoustic mass $M_{ac,h}$. The diagram in Figure 6, which compares the results of the acoustic simulations of network

and FE model, show that the developed equations and the according network in Figure 3 provide the possibility of a fast dynamic simulation of the acoustic behavior inside the capsule.

As shown in Figures 5 and 6, the simulation results of the reduced network model agree with the results of the complex FE simulation up to the first mechanical and acoustical resonance frequency. Compared to the FE simulation, the advantage of the network simulation is the short computing time. For example, the calculation of the complete frequency responses in Figures 5 and 6 using network methods takes about 1 s, whereas the calculation of *one* frequency step of using FE methods takes about several seconds on a normal personal computer (PC). Depending on the chosen frequency resolution, the time advantage using the presented network model of the piezoelectric bending plate can be up to the factor of 1000.

For the simulation of the transient behavior of the piezoelectric plate, the time advantage using network methods is even bigger. The calculation time of a sinusoidal load (10 cycles) at the piezoelectric element is about 1 h using FE methods and a time resolution of at least 10 steps per cycle. The same transient simulation using network methods takes about a few seconds. These examples show that the simulation of the dynamic behavior of the piezoelectric plate transducer using network simulations is much more efficient than FE simulations. Hence, the simulation of the stabilization performance of the piezoelectric bending plate in the following sections is realized using the previously described network models.

Experimental verification of the network model

To prove the network models of the previous sections, a test setup was constructed. There, the mechanical and acoustic reactions v_m/V_{SU} and p_v/v_m of an electrically

Figure 7. Schematic drawing of the measurement setup (Steinert, 2012).

Figure 8. (a) Encapsulation of the piezoelectric bending plate (Steinert, 2012) and (b) piezoelectric bending plate used in the measurements (Steinert, 2012).

stimulated bending plate were measured. The aim of the experiments is to demonstrate the coupling of the electrical, mechanical, and acoustic domains.

Setup

The laboratory measurement setup is shown in Figures 7 and 8. The piezoelectric bending plate is firmly clamped between two pipes of variable length. Thus, on both sides of the plate, a volume of defined size is introduced. On one side, the pressure in the volume is measured by a microphone. On the other side, a hole in the housing enables the measurement of the velocity by a vibrometer. The magnitudes as well as the phase angle of velocity and pressure are measured simultaneously using a digital oscilloscope. The bending plate has a height of $h_{pl} = 0.5$ mm and a radius of $r_{pl} = 20$ mm excluding the clamped area. The piezoelectric ceramic disk (PIC155) has a radius of $r_{piezo} = 10$ mm and a height of $h_{piezo} = 1$ mm. The enclosed air cavities on both sides of the plate have a length of $l_{cav} = 10$ mm.

Results

Figure 9 shows the magnitude of the measured velocity compared to the network simulated values over frequency. Up to the first resonance frequency, the measured

Figure 9. Normalized velocity response in the middle of the bending plate measured and simulated with network methods (Steinert, 2012).

values are nearly identical to those of the simulation. Because of the invalidity of the network model at higher frequencies, the simulation data deviate for frequencies higher than the resonance frequency from the measurements. Additionally, the measured resonance frequency is slightly lower than simulated. It is assumed that the difference is caused by the non-ideal clamping. A higher compliance at the modeled mounting would reduce the resonance frequency in the simulation, too.

Figure 10 shows the consistency of the simulated and measured frequency response of the pressure. The first significant deviation occurs at the first mechanical resonance frequency. The further deviations are due to the low deflection of the plate at high frequencies. The previous experiments confirm that the network model assures the coupling of the electrical, mechanical, and acoustic behaviors of the stabilization unit up to the first resonance frequency with high accuracy.

Simulation of the stabilization performance

In this section, the stabilization performance of an exemplary plate transducer is investigated using the

Figure 10. Normalized acoustic pressure response divided by the velocity magnitude measured and simulated with network methods (Steinert, 2012).

Figure 11. Amplitude response of the load impedance attached to the capsule.

previously developed network approach. Especially, the influence of the mass of the capsule and the additional mass on the dynamic behavior of the system is of interest. The capsule is attached to the complex mechanical impedance $\underline{z}_L$ (Figure 1). The amplitude response of the load impedance is shown in Figure 11.

For the investigations of the mechanical performance of the transducer, the network model shown in Figures 2 and 3 can be reduced without a significant influence on the dynamic behavior of the transducer. The reduced model is still suitable for the simulation of the transducer. A reduction of the network elements in Figure 2 can be realized by

- Short-circuiting of the acoustical domain. All acoustical elements as well as the mechanical–acoustical and the electroacoustic transducer can be removed from the network. This can be done because of the acoustic transparency of the perforated plate transducer.
- Removing the mass elements m_m and m_c representing the vibrating mass of the plate transducer because the regarded frequency working range

Figure 12. Mechanical network of the capsulated plate transducer.

Figure 13. Normalized displacement response of the capsule for different additional masses.

of the transducer is below the first mechanical resonance.

- Disregarding negligible damping effects by removing the damping parameters h_{bp} and Z_{ac} .

The remaining mechanical network of the capsulated plate transducer is shown in Figure 12.

Based on the reduced network model, the performance of the plate transducer has been simulated by harmonic network simulations. The exemplary dimensions of the micro-electromechanical plate transducer are as follows: the carrier plate made of titanium alloy has a diameter of $d_{pl} = 4$ mm and a height of $h_{pl} = 40$ μ m. The piezoelectric element has a diameter of $d_{piezo} = 3.5$ mm and a height of $h_{piezo} = 100$ μ m. For the piezoelectric element, the electromechanical properties of PZN-8%PT according to the work of Zhang et al. (2002) are assumed. The initial mass of the capsule is $m_c = 90$ mg. The displacement ξ_c at the capsule has been chosen as an indicator for the performance of the plate transducer. This is suitable because the displacement of the capsule is proportional to the force induced into the capsule by the transducer

$$\xi_c \sim F_c \quad (5)$$

Following the changes of the displacement magnitudes for different parameters, the mass of the capsule or the mass of the additional mass has been investigated. Figure 13 shows the normalized displacement

Figure 14. Normalized displacement response of the capsule for different capsule masses.

ξ_c/V_{SU} at the capsule for different additional masses. The simulation results show that the displacement at the capsule increases for higher masses m_{add} . Furthermore, the resonance frequency of the magnitude displacement decreases for increasing the additional mass. This resonance frequency f_0 depends on the mechanical compliance of the transducer and the additional mass by

$$f_0 = \frac{1}{\sqrt{n_K \cdot m_{add}}} \quad (6)$$

In another simulation experiment, the mass of the capsule has been varied. The weight of the additional mass is set to $m_{add} = 10$ mg. Figure 14 shows the progress of the displacement ξ_c/V_{SU} for different weights of the capsule.

The diagram shows that the influence of the plate transducer on the capsule increases for lower capsule masses. For low frequencies, the load impedance dominates the dynamic behavior of the system. In addition, the performance of the transducer is independent from the capsule mass. For vibration damping applications, the mass of the capsule has to be chosen as light as possible to increase the performance of the plate transducer.

In the previous simulation experiments, only one parameter at a time was varied and the dependency of the achievable displacement at the capsule was discussed. In a next simulation experiment, both parameters m_c and m_{add} have been varied simultaneously. The masses have been chosen such that their sum is $m_c + m_{add} = 100$ mg. Figure 15 shows the normalized displacement at the capsule for different mass configurations.

Compared to the initial values of $m_c = 90$ mg and $m_{add} = 10$ mg, the performance of the transducer can be increased by the factor of 4 for a capsule mass $m_c = 50$ mg and an additional mass $m_{add} = 50$ mg. Since the mass $m_c + m_{add}$ is nearly constant, this configuration is preferable. A higher ratio $m_{add}/m_c = 75$ mg/25 mg, as shown in the diagram, does not significantly increase the performance of the transducer. This is because for the given load impedance, a capsule mass

Figure 15. Normalized displacement response of the capsule for different capsule masses and the additional masses.

Figure 16. Experimental model for the analysis of the capsule stabilization (left: loudspeaker for excitation of the housing, right: capsule with stabilization unit placed inside).

$m_c = 25$ mg has a minor influence compared to the load impedance. Furthermore, the mechanical resonance frequency f_0 decreases into the working range.

In conclusion, the masses of the capsule have been chosen in a way that the resonance frequency f_0 is out of the working range and the performance of the transducer achieves the required actuating or vibration damping specifications. By using network methods, the dynamic behavior of the system has been analyzed very efficiently. The simulation results provide important design rules for the development of the micro-electromechanical vibration damper.

Vibration reduction experiments using a macroscopic capsule

Setup

The performance of the piezoelectric bending plate for position stabilization has been investigated on a macroscopic laboratory capsule. Figure 16 shows the laboratory setup. It consists of a loudspeaker and a rotational

symmetric capsule. The stabilization unit is placed inside the capsule.

The loudspeaker of type Visaton SC ND4.7 is used to excite the capsule. A mass glued to the middle of the speaker connects it to the capsule. A series resistance $R = 470 \Omega$ ensures that the force produced by the electromagnetic transducer is independent from the excitation frequency. The chosen excitation current of $i_{LS} = 5 \text{ mA}$ at the speaker generates a force of $F = 6 \text{ mN}$.

The rotational symmetric capsule is made of aluminum. It has an outer radius of $r_{o,pl} = 27.5 \text{ mm}$ and an inner radius of $r_{i,pl} = 20 \text{ mm}$. Its height is $h_{pl} = 30 \text{ mm}$. The endings of the capsule are hermetically sealed by plates. The weight of the capsule is $m_c = 95 \text{ g}$. The safety wires shown in Figure 16 only hold the capsule in position and do not influence its dynamic behavior.

The stabilization unit placed inside the capsule is shown in Figure 17. The bending plate made of aluminum has a height of $h_{pl} = 0.5 \text{ mm}$ and a radius of $r_{pl} = 25 \text{ mm}$. The outer ring of holes shown in this figure is used to screw the plate down to the capsule. The inner holes allow an air flow between the air chambers inside the capsule. For the chosen configuration of holes with $N_h = 10$ and $r_h = 0.25 \text{ mm}$, the stabilization unit is acoustically transparent. The piezoelectric disk on the bending plate is made of PIC155. The element has a radius of $r_{piezo} = 12.5 \text{ mm}$ and a height of $h_{piezo} = 0.5 \text{ mm}$. The additional mass of $m_{add} = 10 \text{ g}$ is connected to the middle of the bending plate.

Figure 17. Stabilization unit used in the macroscopic setup. It consists of holes to guarantee acoustic transparency and an additional mass to increase its performance.

The stabilization unit is controlled by the acceleration $\underline{a}_c$ of the capsule. An accelerometer of type Analog Devices ADXL103 with a negligible mass was used to measure the capsule vibration. The cutoff frequency of the accelerometer was set to $f_c = 500 \text{ Hz}$.

Results

To stabilize the capsule, its velocity $\underline{v}_p$ has to be reduced. The achievable amount of velocity reduction by the stabilization unit depends on the stability of the control circuit. A flowchart of the control circuit is shown in Figure 18.

Input value of the circuit is the acceleration $\underline{a}_p$ at the capsule. The electromechanical behavior of the accelerometer is described by the transfer function $\underline{H}_{acc}(f) = \underline{V}_{ADXL}/\underline{a}_p$. The signal of the accelerometer is adjusted by the control transfer function $\underline{H}_c(f)$ and the control gain g . The resulting voltage $\underline{V}_{SU}$ excites the stabilization unit. The electromechanical transfer factor of the piezoelectric bending plate $\underline{Y}_{SU}$ determines the relation between the excitation voltage and the excitation force $\underline{F}_{SU}$. The transfer function $\underline{H}_{p,SU}(f) = \underline{E}_{p,SU}/\underline{F}_{SU}$ describes the ratio of the force $\underline{E}_{p,SU}$ induced into the capsule. Output of the control circuit is the velocity $\underline{v}_p$ at the capsule (see Figure 16).

An indication for the stability of the control circuit is its open-loop transfer function, which can be described by the modified flowchart of the circuit shown in Figure 19.

The measured and simulated open-loop transfer function $\underline{V}'_{SU}/\underline{V}_{SU}$ is shown in Figure 20. The clipped resonance frequency of the measurement is caused by the limited output voltage $\underline{V}_{ADXL} = \pm 5 \text{ V}$ of the accelerometer.

The sensitivity of the control circuit increases over the frequency until the mechanical resonance frequency f_0 of the stabilization unit. It is known that under the condition

$$\left| \frac{\underline{V}'_{SU}}{\underline{V}_{SU}} \right| > 1 \text{ and } \arg\left(\frac{\underline{V}'_{SU}}{\underline{V}_{SU}}\right) = -2n \cdot 180^\circ \text{ for } n \in \mathbb{Z} \quad (7)$$

the control circuit becomes unstable. To increase its stability, an analog controller has been implemented into

Figure 18. Control circuit for position stabilization.

investigated based on simulations to optimize its function. The elements of the network model can be determined by static and modal FE simulation experiments. It has been shown that the mechanical and acoustic aspects of the network model agree very well with the experiments up to the first resonance. Depending on the attached mass, the whole capsule can be accelerated to act as shaker, which can be used to stabilize it. The performance of the piezoelectric bending plate has been analyzed on a laboratory model. A reduction of the capsule vibration up to 6 dB could be measured. Future work concerns the miniaturization of the stabilization unit for medical application and the analysis of further control mechanisms to increase the gain of the control circuit.

Declaration of conflicting interests

The authors declare that there is no conflict of interest.

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