# MODELING OF PANPHONICS G1 FLAT LOUDSPEAKER FOR ACTIVE ACOUSTIC BARRIER CONTROL (AABC)

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## 1 INTRODUCTION

In this paper some special modeling questions regarding to the Panphonics G1 flat loudspeaker will be described. The aim is to develop a sound insulation package that can increase the sound transmission loss of the car steel structure – sound package combination and thus attenuate the interior noise field by active means. The concept is called "Active Acoustic Barrier Control (AABC)". The main component of this package is a special active device: the EMA - *Elastic Mass Actuator* based on the Panphonics' G1 panel loudspeaker element. The EMA sound package (at this stage) consists of two G1 flat panel: one is the actuator and another one is the sensor. The work has been done in the framework of the European Union funded project *Intelligent Materials for Noise Reduction* (InMAR).

## 2 G1 LOUDSPEAKER MODEL

### 2.1 Modeling of the membrane movement

Consider a G1 loudspeaker panel situated on $xy$ plane (Fig. 1.). The structure of the panel is homogeneous toward to the direction of $y$, hence a two dimensional model could be appropriate as a first modeling phase. In the first step the model of the panel membrane was created. This membrane was modeled as a two dimensional thin bar with the length of $L$ and the height of $h$ (Fig. 2.). A pretension $T$ was also taken into account at both of the ends of it. Thus, a partial differential equation of this thin bar [1] with pretension was used as a very basic of our model as:

$$\frac{\partial^2 u(x,t)}{\partial x^2}\left[EI(x)\frac{\partial^2 u(x,t)}{\partial x^2}\right] - \frac{\partial u(x,t)}{\partial x}\left[T(x)\frac{\partial u(x,t)}{\partial x}\right] + m(x)\frac{\mathrm{d}^2 u(x,t)}{\mathrm{d}t^2} = f(x,t)\ . \tag{1}$$

In this case $E(x)$, $T(x)$, $I(x)$ and $m(x) = h\rho$ (where $\rho$ is the density of the bar material)

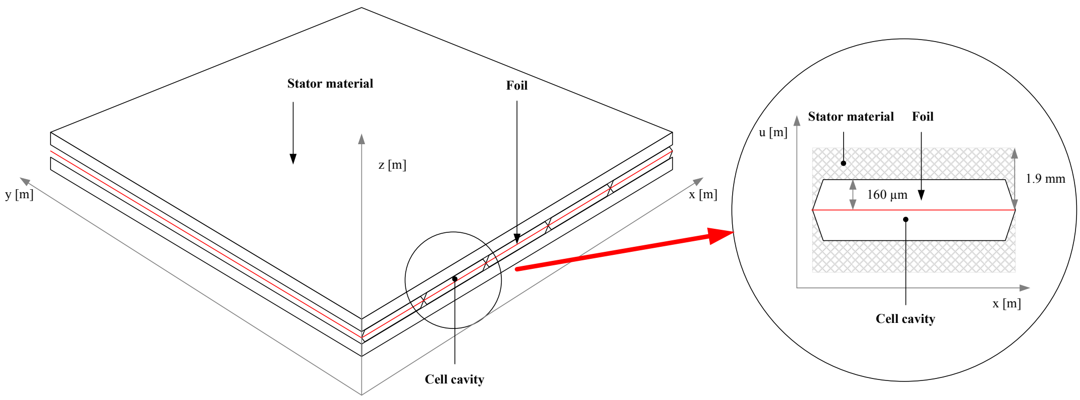


Figure 1: *Global structure of the G1 panel*

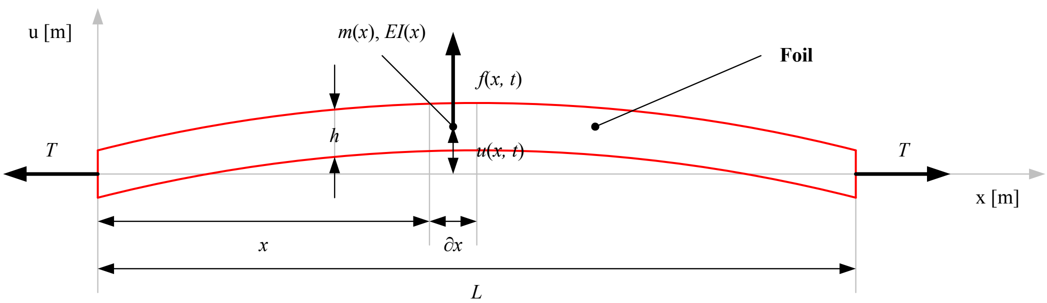


Figure 2: *Mechanical model of the membrane*

are independent from its position ($x$), thus we can substituting them with constant values respectively. Assuming only harmonic functions as the resulted displacement ($u(x,t) = u_0(x)e^{j\omega t}$) and rewrite the Eq. 1. as:

$$EI\frac{\partial^4 u(x,t)}{\partial x^4} - T\frac{\partial^2 u(x,t)}{\partial x^2} - \omega^2 h\rho u(x,t) = f(x,t)\ . \tag{2}$$

Instead of solving this equation analytically, a finite estimation method was used [1].

[1] Let us consider $f(x)$ as a normal function can be derivate in a finite interval. Divide this interval to $N$ point. The value of the derivate function of $f(x)$ at the point $x_i$ can be calculated as follows:

$$\left.\frac{\mathrm{d}f(x)}{\mathrm{d}x}\right|_{x=x_i} \approx \frac{f(x_i) - f(x_{i-1})}{\Delta x}\ .$$

The accuracy of this estimated derivate value is dependent on the length of $\Delta x$, thus the higher value of $N$ lead us to higher precision. This way the higher order derivate value of the $f(x)$ also can be expressed.

Divide the whole bar to $N$ segment. Let $u_i$ denote the displacement of the $i$-th segment measured from the middle line of the rod and $F_i$ denote the transversal force acting on that piece due to the electrical field of the panel. Length of the $i$-th segment is $\Delta x_i$. Rewrite the Eq. 2. for each element in as:

$$EI\frac{\Delta u_i^4}{\Delta x_i^4} - T\frac{\Delta u_i^2}{\Delta x_i^2} - \omega^2 h\rho u_i = \frac{F_i}{\Delta x_i} \ . \tag{3}$$

Let us define the $\mathbf{u}$ and $\mathbf{F}$ vectors as:

$$\mathbf{u}_i = [u_i \ u_{i-1} \ u_{i-2} \ u_{i-3} \ u_{i-4}]^\mathsf{T} \ , \tag{4a}$$

$$\mathbf{F}_i = [F_i \ F_{i-1} \ F_{i-2} \ F_{i-3} \ F_{i-4}]^\mathsf{T} \ , \tag{4b}$$

and rewrite the Eq. 3:

$$EI\frac{1}{\Delta x^3}[1 \ -4 \ 6 \ -4 \ 1]\mathbf{u}_i - T\frac{1}{\Delta x}[0 \ 1 \ -2 \ 1 \ 0]\mathbf{u}_i - \omega^2 h\rho\Delta x[0 \ 0 \ 1 \ 0 \ 0]\mathbf{u}_i = \mathbf{F}_i \ . \tag{5}$$

After using the appropriate mechanical terms and define some new matrices let us expand Eq. 5 for the whole bar:

$$[\mathbf{K}_s - \mathbf{K}_t - \omega^2\mathbf{M}]\mathbf{u} = \mathbf{F} \ . \tag{6}$$

where [2]:

$$\mathbf{K}_s = E\frac{h^3}{12L^3}N^3\mathbf{B}_4 \ , \tag{7a}$$

$$\mathbf{K}_t = T\frac{1}{L}N\mathbf{B}_2 \ , \tag{7b}$$

$$\mathbf{M} = \rho\frac{hL}{N}\mathbf{I}_N \ , \tag{7c}$$

$$\mathbf{u} = [u_1 \ u_2 \ \ldots \ u_{i-N+1}]^\mathsf{T} \ , \tag{7d}$$

$$\mathbf{F} = [F_1 \ F_2 \ \ldots \ F_{i-N+1}]^\mathsf{T} \ . \tag{7e}$$

### 2.2 Calculation method

The force acting on the membrane is depending on the displacement of the membrane. If we calculate the displacement based on the initial force with solving the equation of motion the result will be incorrect due to the spatial dependency of the force. In order

[2] In these matrices $\mathbf{B}_4$ and $\mathbf{B}_2$ are a special matrices constructed based on the appropriate coefficients mentioned above and $\mathbf{I}_N$ is the identity matrix of order $N$.

to find a proper solution an iteration method was used. In the first step the displacement based on the initial force is calculated:

$$\mathbf{u}_0 = [\mathbf{K} - \omega^2 \mathbf{M}]^{-1} \mathbf{F}_0 \ . \tag{8}$$

Afterward this displacement is rescaled and based on its value the actual force is calculated:

$$\mathbf{F}_1 = F(s\mathbf{u}_0) \tag{9}$$

where $F$ is the force function and $s$ is the scaling coefficient. This iteration procedure has to be repeated until the difference between the elements of the displacement sequence is small enough e.g. the solution is convergent.

## 3 WHOLE G1 MODEL

In order to model the mechanical behavior of the whole G1 panel, expand our model with two additional thin bar as the model of the stator material on the both side of the foil. The resulted model can be seen on the Fig. 3. In order to model the real situation the corner elements of each cell connected to the appropriate stator elements with rigid sticks (indicated with black vertical lines), e.g. these elements must be moved together. The forces acting on the foil elements due to the electrical force indicated with red color.

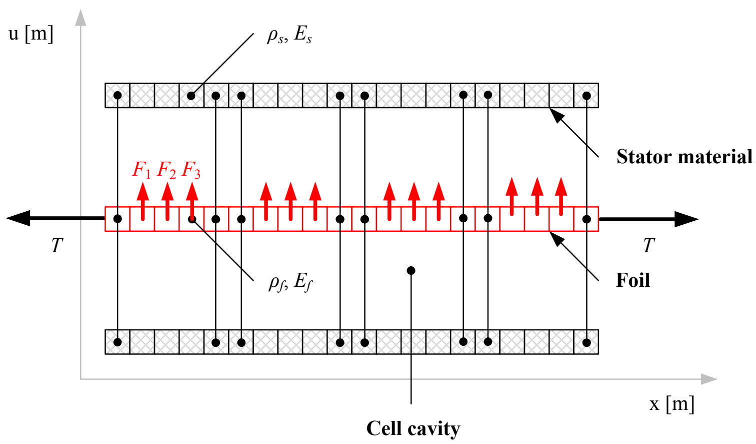


Figure 3: *The mechanical model of the G1 panel*

## 4 RESULTED MOVEMENT

Based on the previously described considerations, now we can rewrite the equation of motion and solve it. The movement of the foil and the cell cavities can be seen on the Fig. 4. As one can see the whole structure is moving not only the membranes. A detailed preliminary measurement have done by the University of Stockholm (KTH) and they found quite similar results related to the local movement of the membrane and the global movement of the panel.

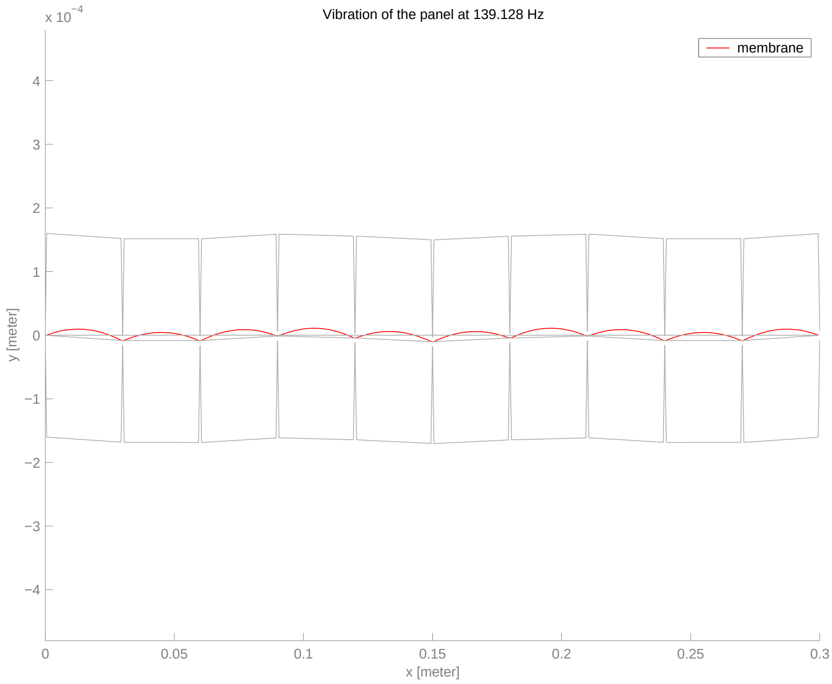


Figure 4: *Movement of the foil and the cells at 139 Hz*

## 5 RADIATED SOUND

Let us investigate the sound radiation of the panel based on the sound radiation of one cell. For the calculation an acoustic transmission line model was used. In this case the transmission line consist of the cavity of the cell, the stator material (as an acoustical resistance) and the impedance of the air. Calculation of the air load based on the appropriate Bessel and Struve functions. For the impedance of the stator material the *Delany-Bazley* formula was used. Based on transmission line model the pressure was calculated and it is qualitative similar to the earlier measurement results.

## 6 EMA INTEGRATION IN CAR SOUND PACKAGE

In order to model the whole car sound insulation package, an additional bar (model of the steel plate) added to the the original panel model as depicted on Fig. 5. The springs will be model the effect of the thin air gap. One of the future task will be to evaluate this new model, verify and optimize it.

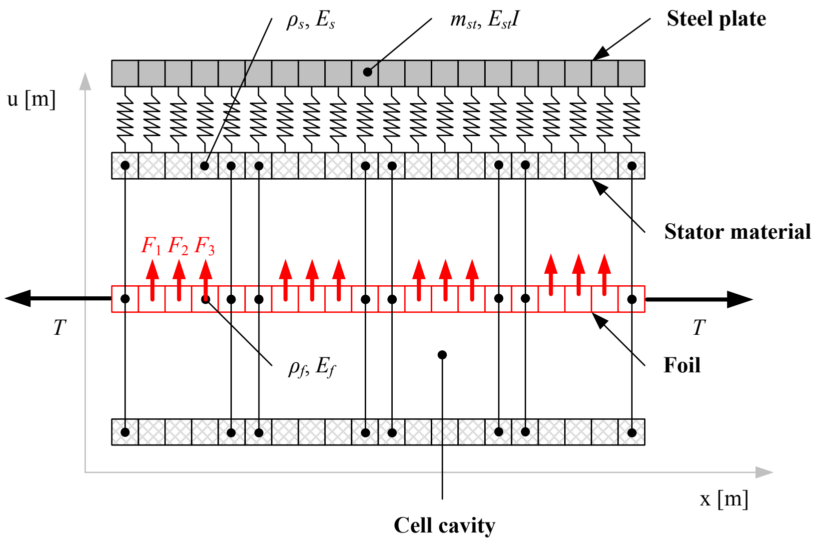


Figure 5: *Model of the car sound insulation package*

## 7 CONCLUSION AND FUTURE WORK

Due to the scattering of the initial parameter of the G1 (material parameters), the results are also various. In order to get more reliable results a detailed verification measurement will be done. Based on this verification the G1 panel and also the car sound package optimization will be done. In the meantime several preliminary measurements have been done with the existing EMA in the laboratory of VTT Tampere. In order to optimize this sound insulation package, different control systems (and control system strategies) have been tried and evaluated.

## REFERENCES


[1] MEIROVITCH L, *Elements of Vibration Analysis*, McGraw-Hill Inc., New York, 1975.