Numerical simulation of a viscoelastic sound absorbent coating with a doubly periodic array of cavities

Sayed Hamid Sohrabi and Mohammad Javad Ketabdari

Abstract: Nowadays, research on underwater sound absorbing coatings is important on marine combats. Due to high cost of acoustic experimental tests, determination of an accurate simulation method for assessment of the coating characteristics is an important goal for scientists. In this paper, a simulation-based code in ANSYS was developed. Embedding cavities in coating mediums is a way to improve their acoustic performance. The behavior of a sound absorption coating dramatically depends on the medium features, layer thickness, the size and distribution of cavities. Due to cavities symmetrical distribution and calculating time reduction and cost, only one isolated unit cell of a periodic sound absorption coating was studied. In this study, two fluid mediums in front and back of coating are modeled. The objective is to study the acoustic performance of the absorbent coating due to normal incident plane wave. The present model results were compared with measurements test data. The good agreement in this comparison illustrates the accuracy and validity of the model. The model was then utilized to investigate the effects of cavity geometry on the behavior of sound absorption coatings. The results of the comparison between a cone frustum hole and a...
cylindrical hole demonstrate larger transmission coefficients in cone frustum holes. Echo reduction results show no major difference between these coatings with the above mentioned types of holes. Furthermore, it is evident from the results that in general, in low-frequency ranges under 10 kHz, material characteristics play the most important role regarding sound absorption and dissipation properties.

**Keywords:** underwater sound absorbent coating; anechoic coating; double periodic scatterers; effective impedance; viscoelastic; numerical method

### 1. Introduction

In order to reduce the reflections of sonar impinging sound wave energy in underwater structures, acoustic anechoic coatings are used. The primary research on absorbent coatings was carried out with a focus on pulsation oscillations of cavities in a rubber medium (Meyer, Brendel, & Tamm, 1958) and a one-dimensional model proposed for sound absorbent properties of a viscoelastic material which included short cylindrical cavities (Gaunaurd, 1977). Investigation on sound transmission and reflection of an elastic plate immersed in water without absorbent coating was performed using resonance theory and results were compared with exact calculations (Fiorito, Madigosky, & Uberall, 1979).

Some researchers have theoretically studied the oscillatory behavior of perforated viscoelastic layers as well as the deformation modes of cavities (Gaunaurd, Callen, & Barlow, 1984; Gaunaurd, 1985). In order to attain a high standard sound absorption performance, it is necessary that coating and water have very close characteristic impedances. In such a situation, minor induced reflected waves occur. In addition, it is recommended that the part of sound energy which enters the layer be dissipated to the desired extent. In this way, the result will be the prevention of more reflections at back plate, which eventually will return to water (Easwaran & Munjal, 1992; Gaunaurd, 1985; Meng, Wen, Zhao, Lv, & Wen, 2012). A number of designs, including both materials and structures, are proposed for the attenuation of sound in air and water. Among materials, viscoelastic polymers are extensively used in sound attenuation coatings due to their exceptional mechanical features (Jarzynski, 1990). Many sound absorbent coatings are proposed to attenuate the reflection of sound from the structure. The most important types of these designs include the absorbent coating with a multilayered gradual transmission, the absorbent coating using sound redirection and mode conversion. Acoustical absorbent coatings can be applied in the outer surface of underwater vehicle hulls. Coatings exposed to an acoustic wave incidence cause a reduction in the sound reflection and a more internal scattering in the coatings medium (Kinsler, Frey, & Coppens, 1999).

In order to create a better impedance matching, rubber-like materials are utilized. These materials have a close characteristic impedance to water and as a result are suitable in making underwater anechoic layers. However, their compressional wave absorption is not as efficient as their shear wave absorption. Therefore, in order to increase sound absorption, scatterers are usually embedded in rubber layers. The result would be the induction of mode conversion and scattering in the sound wave, leading to an enhancement in sound absorption (Panigrahi, Jog, & Munjal, 2008). Mode conversion at boundaries, scattering by homogeneities, redirection and intrinsic absorption of viscoelastic material, are among the most important mechanisms used for sound attenuation in lossy materials.

Finite element method (FEM) has been proposed as a widely used method for evaluating different absorbent coatings. A 2D FEM approach is applied to analyze the scattering of an infinite, uniform plane grating of compliant tubes (Hennion, Bossut, & Decarpigny, 1990). In addition, Hennion and Decarpigny (1991) developed a 3D FEM approach to analyze the Alberich
coating, which can be regarded as a double-periodic elastic structure. They made a comparison between this structure and previous analytical and experimental results in order to validate the accuracy of the proposed model. Easwaran and Munjal (1992) have also proposed a 3D FEM to investigate the reflection behavior of resonant sound absorbents. They replaced the modal summation mechanism used in the fluid domain with particular conditions related to impedance boundary. A modified hybrid type FEM was used by Panigrahi, Jog and Munjal (2008) to analyze different types of underwater acoustic coatings. Cai, Hung & khan (2006) studied the optimization of linings base on FEM analysis. Moreover, Ivansson (2008) used a technique usually applied in electron scattering and band-gap computations of photonic crystals. He studied the viscoelastic sound absorption coatings with periodic spherical and super ellipsoidal cavities. FEM can be applied to irregular cavities. However, it is a time-consuming approach and optimization may become a challenge. An analytical approach can be applied to predict the acoustical performance of simple models. Tang, He and Fan (2005) proposed a 2D analytical model in order to calculate the absorption performance of a viscoelastic coating which contained cylindrical holes. In this research, the unit cell of sound absorption coating was estimated to be a finite cylindrical tube.

Study on local inhomogeneity effects in absorption and scattering of a coated plate under impinging on plane sound wave was investigated by Zhang and Pan (2013) and Zhang, Huang, Zhang and Pan (2015).

Work on FEM modeling of anechoic coating included scatterer cavities with backed stiffened plates using the Bloch-periodic boundary conditions which were performed by Fu, Jin, Yin and Liu (2015). Some absorbent coating use porous material with new representative models, such as considering a periodic arrangement of rectangular cross-section pores (Groby, Pommier, & Auregan, 2016). However, it is not possible to describe more complicated cavities directly by using analytical functions.

Therefore, there is a need to develop both simplified and approximated analytical methods. The most usual acoustic anechoic coatings have a resonant cavity design as depicted in Figures 1 and 2. As can be seen, the absorbent coatings are made of rubber layers containing some cavities. In this study, a simulation-based method which employed FEM is suggested in order for the design synthesis of the acoustic absorbent coating in which cylindrical holes are located. The schematic of an Alberich anechoic coating is illustrated in Figure 2. It includes three layers: an inner viscoelastic rubber layer which contains holes itself, a back layer and a cover layer contiguous with water. Lower reflectivity of the sound waves is possible using these absorber designs. These designs have a low thickness relative to the sound wavelength in water. This thickness of anechoic coatings usually leads to a small reflection over a narrow frequency range (Lane, 1981).

Figure 1. A representative model of underwater sound anechoic coating.
The mechanism used in this study for sound absorption and reflection is a direct method. In this method, the specific impedance at any section in the upper duct is used as the input specific impedance of the absorbent coating at its front surface. Regarding echo reduction (ER) and transmission loss (TL) performance of these coatings, different types of designs are proposed for varied and contradictory requirements.

2. Theoretical background

To present the theoretical background of absorbent or anechoic coatings, a plane harmonic wave as one of the simplest and most fundamental types of sound signals is considered. The acoustic pressure $p_a$ which is a function of time and location is calculated as follows (Hopkins, 2012):

$$p_a = p_0 \cos(\omega t - kx + \phi_0)$$

where, $p_0$ stands for amplitude and $\phi_0$ is the initial phase of the wave. The total phase $\phi$ at position $x$ and time $t$ equals to $\phi = (\omega t - kx + \phi_0)$. As the wave fronts are planes perpendicular to the $x$ axis, the wave itself is also plane. The wave is harmonic because it has a single associated frequency $f$. The harmonic wave is a basic signal because any transient signal can be defined as a superposition of harmonic waves with various frequencies. Using complex form representation for sound waves, in many cases, will suffice (Easwaran & Munjal, 1992). The harmonic wave is depicted by the following complex exponential (Eq. (2)):

$$p = p_0 e^{i(\omega t - kx + \phi_0)}$$

Here, the physical wave (Eq. (1)) is the real part of the complex quantity which stands for the sound wave. Due to the properties of the undamped medium, sound waves propagate with a constant speed and basic relation between sound speed, wavelength and frequency will be established. The study of sound propagation in lossy media has been restricted to linear cases as the relationship between stress and strain is linear in such materials. In such cases, a plane harmonic sound wave experiences an exponential decay with distance as follow:

$$p_a = p_0 e^{-\alpha x} e^{i(\omega t - kx)}$$
In this equation, \( p_a \) represents the acoustic pressure at distance \( x \) from the reference point, \( k' = 2\pi/\lambda \) is the real part of the wavenumber, \( \lambda \) is the wavelength and \( \alpha \) is the sound attenuation coefficient in \( 1/\text{m} \). It should be noted that the attenuation coefficient \( \alpha \), reflection and transmission coefficients determine the performance of an anechoic coating. The simplest situation is when there is a plane surface as the boundary between medium and coating and the angle of incident of the sound is normal. The anechoic performance of an acoustic absorbent coating is calculated based on its sound reflection (Eq. (4)) and transmission (Eq. (5)) coefficients defined as follows (Kuttruff, 2007; Vigran, 2008):

\[
R = \frac{p_r}{p_i} = \frac{\gamma - \gamma_m}{\gamma + \gamma_m} 
\]

\[
T = \frac{p_t}{p_i} 
\]

where \( p_i \) and \( p_r \) represent the incident and reflected waves, respectively, on incident surface and \( p_t \) shows the transmitted wave on transmitted side of the absorbent coating (see Figure 3). They are dynamic because of the dynamic definition of incident, reflected and transmitted pressure wave as

\[
p_i = p_i e^{i(\omega t - kx + \phi_i)}
\]

\[
p_r = p_r e^{i(\omega t + kx + \phi_r)}
\]

\[
p_t = p_t e^{i(\omega t - kx + \phi_t)}
\]

Meanwhile, \( \gamma_m = \rho_m c_m \) is the characteristic-specific impedance of the fluid medium in which \( \rho_m \) and \( c_m \) are the density and sound speed in the medium, respectively. \( \gamma \) which is expressed by Eq. (6) is the input specific impedance at incident surface and can be described as the ratio between the acoustic pressure and the particle velocity in this surface.

\[
\gamma = \frac{p_r + p_i}{s} = \frac{p_a}{j\omega U} 
\]

Figure 3. Schematic of acoustic wave terms propagating through different media.
In this equation, \( s \) and \( u \) are the structural normal velocity and displacement of the absorbent coating, respectively. Moreover, \( p_a \) shows the complex acoustic pressure including both incident and reflected acoustic pressures. \( \omega \) represents the angular frequency and \( j \) is the imaginary symbol. Considering Eq. 4, it is clear that a perfect match between the impedances of the absorbent coating and medium will abolish any sound reflection from the absorbent coating. In general, some part of the incident sound wave would be reflected inevitably as a result of the mismatch between \( \gamma_m \) and \( \gamma \) impedances. The remaining parts will be transmitted into the absorbent coating through its incident surface and continue their way inside the absorbent coating. Accordingly in lossy media, the wave energy will be attenuated resulting in a total reduction in the radiation.

3. Boundary conditions

In the present model, the following boundary conditions were specified for the unit cell.

According to Figure 1, the uniform structural displacement \( U_z \) is used at the top of the upper acoustic duct to produce the plane acoustic wave impinging on the absorbent coating. The full absorbing boundary condition was set on the bottom of the lower acoustic duct to represent an anechoic end. This guarantees that in the lower acoustic duct, there exists only transmitted acoustic wave. On the other four side surfaces of FEM model, the following periodic boundary conditions are expressed (Easwaran & Munjal, 1992; Hennion & Decarpigny, 1991):

\[
\begin{align*}
    u_x(x, b, z) &= u_x(x, -b, z) \\
    u_y(b, y, z) &= u_x(-b, y, z) = 0 \\
    u_y(b, y, z) &= u_y(-b, y, z) \\
    u_y(x, b, z) &= u_y(x, -b, z) = 0
\end{align*}
\]

(7)

4. Transmission and reflection equations

In the upper acoustic duct, the standing acoustic pressure \( p_{st} \) includes two pressures. These pressures are related to the two opposing incident and reflected acoustic waves. When the reflection coefficient \( R \) is calculated, the incident acoustic pressure in the upper duct can be computed as

\[
p_i = \frac{p_{st}}{1 + R}
\]

(8)

The acoustic pressure in the lower acoustic duct consists of only the pressure which is related to the transmitted pressure. In other words, \( p_t \), as the lower end of the bottom acoustic duct, is nonreflective. Therefore, the transmitted coefficient can be achieved from Eq. (9) as

\[
T = 20\log\frac{p_t}{p_i}
\]

(9)

Consequently, in order to achieve the ER or TL characteristic of any design under water closure conditions, it is necessary that an acoustic pressure wave of certain magnitude and frequency impinge on the layers. In this case, the relative magnitude of the reflected and transmitted waves can be calculated to specify the desired characteristics. However, it is important to consider the distribution of the air cavities while modeling the FE domain in this analysis. As the unit cell has a symmetric nature, the displacements normal to all side surfaces are limited. The analysis of these viscoelastic coatings follows the certain steps described by Panigrahi et al. (2008).

Taking into account the symmetry of the perforation distribution, we selected a portion of the coating which can represent the total domain when considered in the appropriate boundary conditions. This unit cell domain is meshed with eight-noded elements for analysis. The formulation of the FE follows this discussion. The mass, stiffness and damping matrices are extracted from the discretized differential equations of motion in the weak form while taking into account the appropriate material features and their frequency dependence. Proper displacement boundary
conditions are applied considering the symmetry of the domain. Impedance boundary condition was applied on the water closure side. Applying this impedance boundary condition, it was possible to simulate the semi-infinite water domain on the downstream side.

In order to simulate the impinging sound pressure, a unit pressure loading was applied on the outer coating surface. The actual amplitude of this pressure is not important. The reason is that the ratio of the reflected and transmitted amplitudes to that of the incident wave is used in computing ER or TL. The complex nodal displacement values were obtained by solving the equation of motion. The corresponding complex nodal pressure values and particle velocities were evaluated using the above results. All the values obtained correspond to those of the standing waves occurred in the domain. To obtain the mean particle velocity, the normal particle velocities of the wall in the wave propagation direction were averaged. Therefore, the acoustic impedance in the wave propagation direction at the impinging surface is obtained by

\[ \gamma_c = \frac{P_{st}}{S_x} \]  

where \( P_{st} \) shows the measured standing pressure in front surface of coating and \( S_x \) is the mean particle velocity along the direction of wave propagation. It should be noted that the \( x \)-axis of the global coordinate system was considered in the wave propagation direction. Through the calculation of the reflection coefficient of the coating, the two corresponding parameters, i.e., ER and TL, can be obtained as follows:

\[ ER = 20 \log_{10} \left| \frac{1}{1 + R} \right| = 20 \log_{10} \left| \frac{P_i}{P_r} \right| \]  

\[ TL = 20 \log_{10} \left| \frac{1}{T} \right| = 20 \log_{10} \left| \frac{P_i}{P_t} \right| = 20 \log_{10} \left| \frac{P_{st}}{(1 + R)P_t} \right| \]  

5. FEM validation

In order to validate the suggested simulation-based method, the reflection and transmission of an absorbent coating structure was evaluated. The selected coating structure contains an array of cylindrical holes. The properties of the upper and lower acoustic ducts in the surrounding medium and the absorbent coatings are presented in Table 1. The geometrical dimensions of the absorbent coating and holes are depicted in Figure 4.

In Table 1, \( E \) stands for the Young’s modulus of elasticity, \( \rho \) represents the density, \( \nu \) is the Poisson’s ratio, \( C \) is the sound speed and \( \eta \) is the loss factor. Using ANSYS code, the FE model for the coating structure was created. This model includes two acoustic fluids for upper and lower fluid ducts as well as the structural models for three layers (cover, lining and backing).

### 5.1. Model calibration

The acoustical performance of the structure was analyzed over a frequency range of 1–12 kHz using the FE model. We also performed a mesh sensitivity analysis to represent the mesh dependency of the current problem. Comparing the transmission coefficient values shown in Figure 5, it is evident that the mesh size has a significant effect on the obtained transmission coefficient values. The effect of mesh size was found to be more important at higher frequencies.

| Materials | \( E \) (Pa) | \( \rho \) (kg/m\(^3\)) | \( \nu \) | \( C \) (m/s) | \( \eta \) |
|-----------|-------------|----------------|-----|-------------|---------|
| Silicone  | \( 1.8 \times 10^6 \) | 1,000           | 0.49976 | –           | 0.15    |
| Fluid (water) | – | 1,000 | – | 1,489 | – |

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Based on trend obtained from the mesh sensitivity analysis, a frequency-dependent mesh model was developed. Table 2 presents the element sizes used in this model related to the frequency of excitation.

**Table 2. Element size distribution with respect to the smallest acoustic wave length**

| Frequency (Hz) | Mesh size (EPW) |
|---------------|-----------------|
| 1,000–2,000   | EPW = 10        |
| 2,000–6,000   | EPW = 20        |
| 6,000–8,000   | EPW = 30        |
| 8,000–9,000   | EPW = 40        |
| 9,000–10,000  | EPW = 50        |
| 10,000–12,000 | EPW = 60        |
The frequency-dependent mesh was modeled to represent additional array parameters using FORTRAN commands within the ANSYS Parametric Design Language (APDL). These arrays are referred to as meshing parameters related to individual frequencies. In this way, the mesh is modified continuously all through the simulation regarding the frequency of iteration. The process continues in a loop until the final iteration, which refers to a frequency of 12 kHz, is reached.

5.2. Validation of the model

Figure 6 represents the transmission coefficient obtained from FEA for the frequency range of 1–12 kHz. Comparison of the FE results and those of experimental test data shows that there is an average difference of 1.7 dB between results. This difference means that, for most of the frequencies, the transmission coefficient values obtained from FE model agree well with the experimental data.

Accordingly, the ER performance of that coating was computed using the validated model as shown in Figure 7. According to Figure 8, the ER has small values in a range of frequencies lower than 8 kHz. However, for frequencies higher than 8 kHz, it increases significantly.

Figure 6. The transmission coefficient of silicone absorbent coating: dashed line is experiment and solid line is present FE model.

Figure 7. The ER performance obtained from FE model, for coating immersed fully in water.
For better comprehended evaluation, a simple absorbent rubber coating immersed in water was considered. The ER and TL of this homogeneous absorbent coating were evaluated. For comparison, the analytical relations of reflected and transmitted coefficients (Easwaran & Munjal, 1992) of this rubber coating as a viscoelastic layer was introduced (Eqs. (13) and (14)).

\[
R = \frac{c_1 c_2 - \tau^2}{c_2 + c_1} \left( \frac{1}{c_2 + j\tau} + \frac{1}{c_1 - j\tau} \right)
\]  

(13)

\[
T = -j\tau \left( \frac{1}{c_2 + j\tau} + \frac{1}{c_1 - j\tau} \right)
\]  

(14)

where \(c_1 = \tan(\delta)\), \(c_2 = \cot(\delta)\), \(\tau = \rho_0 c_0 / \rho c\) and \(\delta = \omega d / 2c_d\) and \(c_d\) is the dilatational sound speed in coating material. Replacing Eqs. (13) and (14) into Eqs. (11) and (12) in turn, ER and TL of absorbent coating can be calculated. The properties of adjoining environment i.e. water medium and the absorbent coating are presented in Table 3. The dimensions of the unit cell of absorbent coating are demonstrated in Figure 8.

In Table 3, \(E\) stands for the Young’s modulus of elasticity, \(\rho\) represents the density, \(\nu\) is the Poisson’s ratio, \(C\) is the sound speed and \(\eta\) is the loss factor. Using ANSYS software, the finite element model for the rubber coating was created. This model comprises two acoustic fluids as well as the structural models for absorbent coating.

It is evident in Figures 9 and 10 that the offered FEM is valid as there is a good agreement between analytical and model results.

In addition, studying the acoustic performance of an Alberich type coating or any acoustic coating that included voids is complicated. These complexities consist of meshing the models, model validation with experimental data, considering material dependency to frequency, true viscoelastic martial models and accurate post processing. Calculation time is another problem, because we need to check further number of frequency points to get all peaks in TL and ER curves. Modeling is needed in wide range of frequencies to discover better vision in subject. But, accounting wider frequency ranges needs for smaller elements size and results in more computer time, modern hardware and harder meshing.

| Materials          | \(E\) (Mpa) | \(\rho\) (kg/m³) | \(\nu\) | \(C\) (m/s) | \(\eta\) |
|--------------------|-------------|------------------|---------|-------------|---------|
| Rubber coating     | 1.4         | 1,100            | 0.49    | –           | 0.23    |
| Fluid (water)      | –           | 1,000            | –       | 1,489       | –       |
Furthermore, considering coating with complex cavity shapes, lead to more complexity. For optimization goals, another complexity is the inter-scattering between cavities. Then scale analysis of these coatings will needed.

5.3. Input noise sensitivity

In this study, the ER and TL are the frequency responses of rubber absorbent coating and are depend on the nature of system. While the nature of the system has no change, if we change the input incident wave, the response must be unchanged accordingly. The changes include noisy amplitude or for better sense the doubled amplitude of incident wave. In some other analysis, the variation in material property was checked. Below, there are some analysis results performed to show this fact.

(a) ER and TL of the original model with unit amplitude incident wave were compared with model results with double and noisy incident wave amplitudes. As can be seen in Figures 11 and 12, there is no change in the model response for various wave inputs.

(b) Because the system is linear, when the input (here incident wave amplitude) changes, variables such as input pressure or normal velocity of front surface of rubber coating in any point on it will change too. But trend of these graphs do not change and vary linearly with the input magnitude. It means that if we double the input amplitude, the pressure and normal velocity over a frequency range have the same trend with original model with unit amplitude, but with doubled values. Furthermore, if the input incident wave amplitude be
noisy, the value of pressure and normal velocity on any point on front surface of rubber coating in a frequency range turn to noisy similar to input amplitude, while overall trend of pressure or normal velocity will not change. Figures 13 and 14 show these results for real pressure and normal velocity over frequency range of 0.2–12 kHz.

Because of large magnitude of real part of normal velocity in front surface of coating, it is difficult to show the fluctuating behavior of signals for frequencies over 2 kHz. So, for better sense, the normal velocity (real part) versus frequency over 2 kHz, the exaggerated noisy behavior of signal, is illustrated in Figure 15.

(c) Changes in material properties such as density, elasticity modulus and loss factor change the nature of the model. Then model responses will be altered, too. Nonetheless in sensitivity analysis if the input variables are noisy, the model responses will not change significantly. Therefore, the evaluated model represents a constant value. Here to examine this fact, three models with various elasticity modulus are considered. The elasticity modules are differing in values as $E$, $0.95E$ and $1.05E$ where $E = 1.8e6$ Mpa. Figures 16 and 17 show that for such a change, the model response will not be altered significantly and this means that the considered model is working properly.
Figure 13. The effect of noisy input incident wave on pressure real parts.

Figure 14. The noisy input effect on particle normal velocity real parts.

Figure 15. Noisy behavior of particle normal velocity real part.
As can be seen in these figures, the considered model was not sensitive to noisy changes in its variables.

6. Results and discussion
In this section, the ER and TL performance of several coatings as well as the hole size and geometry effects on coating performance are presented.
6.1. Cylindrical hole size effect

Figure 18 shows the results of the transmission coefficient obtained from FE modeling for various cylindrical section radii. Simulation was performed with a coating included a cylindrical cavity with radius $R$. To study the effect of cylindrical cavity, the cavities length was fixed. Comparison of the results in Figure 18 shows that increasing the inclusion section radius provides a better level of the transmission coefficient in the desired frequency range. It is obvious that the TL of a coating containing a hole with a radius of $3R$ is larger than that of for a cavity with $2R$ radius. This trend remains valid for smaller cavity radius. It is evident from this figure that in general, larger cavity size leads to more TL magnitude. The reason is that the reduction in stiffness causes the impedance reduction of the coating. In such conditions, with lossy coating material, the sound wave energy will dissipate and convert to heat. Thus, the coating will show a better absorption.

The ER results of the geometry obtained from FE model are presented in Figure 19. Comparing ER of different hole sizes shows that increasing the radius of sections worsens the level of ER in desired frequency range due to a reduction in the stiffness of the coating.
6.2. Hole geometry effect
In this section, the hole geometry effects on transmission coefficient and ER are investigated. Figures 20–24 present the results of the transmission coefficient and ER obtained from the FE model for the cone frustum inclusion. Two cone frustum holes were considered for two cases: one with small cone base toward the impinging sound waves (case 1) and the other with larger base (case 2).

For case 1, the transmission coefficient and ER are shown in Figures 21 and 22. As can be seen in Figure 21, the cone frustum, as compared to cylindrical hole, shows a larger transmission coefficient.

The largest TL value for coating included frustum cavity case 1 and cylindrical cavity case 2 are 36.7 and 29 dB respectively in a frequency of 1,500 Hz. It means that if the material media is lossy enough, a much greater portion of the wave energy will be attenuated. This phenomenon may
happen due to the mode shapes of frustum or the profile of the cavity in scattering of incoming waves inside the material. It seems that the cone shape better redistributes the imposing signals, demonstrating more efficient TL. Figure 22 shows the ER results of coatings with cylindrical and cone frustum holes. Results indicate that there is no major difference between these two coatings for frequency ranges up to 8 kHz. However, in ranges over 8 kHz, discrepancy starts to rise and the layer containing cylindrical hole shows a better ER performance.

Discrepancy on TL results between coatings with frustum (case 1) and cylindrical cavity in average is 10.5 dB over frequency range of 1–12 kHz.

In case 2 for reverse cone frustum hole (Figure 23), transmission coefficient and ER are considered. In this case as shown in Figure 24, the TL performance of reverse cone frustum hole coating is better than that of coating containing cylindrical hole. In this case, the maxima value of
about 34 dB happens in a frequency of 1,500 Hz. Comparing these three cases reveals that coating with frustum cavities is more efficient in TL.

In addition, the ER characteristics of this coating, similar to what was observed in case 1, are worse than that of cylindrical hole coating (see Figure 25). Nevertheless, the ER characteristic of this coating (case 2) is close to that of case 1.

A comparison between the results of coatings in cases 1 and 2 (Figure 26) shows that case 1 has a slightly better transmission performance than that of coating of case 2. The differences between two frustum cavity cases over 1–12 kHz are about 1.2 dB in average. Furthermore, regarding the ER performance (see Figure 27), the two cases are very close to each other while coating in case 1 behaves slightly better.

7. Conclusions
This paper is an investigation into an Alberich type acoustic absorbent coating. We developed a combined FEM–ADM model to analyze the TL and ER performance of an absorbent coating. This model has some advantages over other models, including uniform measured pressure data in upper and lower ducts, direct modeling of acoustic fluid and using FSI elements between
acoustic fluid and absorbent coating. The use of FSI elements guarantees degrees of freedom (DOF) matching between fluid and solid elements, resulting in more accurate results.

The relationship between ER and TL regarding impedances of fluid and incident surfaces was introduced. Based on the presented model and this relationship and by using APDL of ANSYS software, a code was developed to study the acoustic performance of the coating. The proposed model was validated based on experimental data. Meanwhile, the cylindrical hole radius size effects were studied. Results show that increasing the hole section radius improves the transmission coefficient but worsens the level of ER in an almost desired frequency range. In addition, the hole geometry effects were also investigated. Two individual cones frustum in direct and reverse positions (cases 1 and 2) were considered. The comparison of these two types of cone frustum with cylindrical hole indicates that cone frustum holes show larger transmission coefficients. In other words, in lossy media, a main part of wave energy will be attenuated in coatings with cone frustum holes. ER results indicate that there are no major differences between the two types of coatings (cone frustum and cylindrical holes) for frequency ranges below 8 kHz. However, in frequency ranges over 8 kHz, discrepancy in ER starts to rise and the coating with cylindrical holes demonstrates a better ER performance. Finally, a comparison was made between the two cone frustum holes embedded in individual coatings but in reverse directions (cases 1 and 2). The comparison of these two cases shows that case 1 coating creates a slightly better transmission performance than
the coating used in case 2. Regarding the ER performance, the two cases were very close to each other but case 1 coating had a slightly better performance. Overall, it can be concluded that in low-frequency ranges below 10 kHz, material plays the most important role regarding sound absorption and dissipation properties.

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Author details
Sayed Hamid Sohrabi1
E-mail: s.h.sohrabi@aut.ac.ir
Mohammad Javad Ketabdari1
E-mail: ketabdari@aut.ac.ir
1 Department of Marine Technology, Amirkabir University of Technology, Tehran, Iran.

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