Applied Acoustics 113 (2016) 34–38

Contents lists available at ScienceDirect

Applied Acoustics

journal homepage: www.elsevier.com/locate/apacoust

Technical note

Prediction of noise inside an acoustic cavity of elongated shape using statistical energy analyses with spatial decay consideration

Cheng Yang *, Li Cheng

Department of Mechanical Engineering, The Hong Kong Polytechnic University, Hung Hom, Hong Kong Special Administrative Region

ARTICLE INFO

Article history:
Received 5 May 2016
Received in revised form 2 June 2016
Accepted 6 June 2016

Keywords:
Statistical Energy Analysis
Interior noise prediction
Long acoustic room

ABSTRACT

In this paper, Statistical Energy Analysis (SEA) is used to predict the interior noise of an acoustic cavity of elongated shape. The disadvantage of the conventional SEA method, which quantifies the response in terms of the energy averaged over each subsystem, is overcome by introducing a one-dimensional spatial decay relation, through which information about the acoustic energy variation in the elongated direction is taken into account. The modified SEA is experimentally validated using a 1:5 scaled space station prototype, having the longitudinal dimension much larger than the cross-sectional dimension. It is also compared with a model reported in the literature. It is shown that, in the region where the acoustic pressure level decays at a constant rate, the two models agree well with each other and are capable of estimating the acoustic pressure variation along the space station cabin. However, near the end walls where the decay rate of the acoustic pressure level is not constant, the proposed model provides better accuracy.

1. Introduction

The overwhelming noise to which astronauts are exposed in space station becomes a growing concern in long-term space travel missions. Excessive noise interferes with communication, disrupts concentration and reduces operation performance, impacting on the well-being of the crew on board. A systematic acoustic control strategy for the quietness of the space station should therefore be developed in the design stage to avoid remedial actions after launch.

Prediction of the acoustic pressure level as well as its distribution inside the space station is an indispensable part of a successful implementation of the acoustic control strategy. There are various prediction tools available for interior noise problems: Boundary Element Method [1] and Finite Element Method [2] are suitable for low frequency noise prediction; SEA [3] is appropriate for high frequencies; and some hybrid methods [4–7], which bridge the gap between low and high frequency ranges, are developed for mid-frequency problems. It is the objective of this paper to predict the interior noise of a space station at high frequencies where explicit prediction tools like BEM or FEM become computationally expensive.

Standard SEA formulation has been well documented in many textbooks, for example Lyon [8] and Craik [9]. The condition that allows the implementation of SEA is that, at high frequencies, energy is distributed uniformly within each subsystem. In connection with this assumption, it is fairly acceptable to use a single quantity, usually the mean energy, to quantify a subsystem response. This ensemble representation, however, becomes an obstacle to observing the spatial variation in the subsystem as the difference in energy level could only occur between different subsystems. An example in which such a need is required would be the design of a space station, where the information about the acoustic energy distribution in the compartment would be crucial for designers to plan a less noisy sleeping area for astronauts. A space station compartment constitutes intrinsically an air volume and thus it is a straightforward practice to subdivide this air volume into a number of SEA subsystems. However, considerable debate arises over the validation of this practice. Fahy [10] made a comment on the validity of subdividing an air volume into SEA subsystems. In that note, one major concern is that the contiguous subsystems, defined by the artificially introduced interfaces, are not weakly coupled because these interfaces represent no geometry or material discontinuities. The subsystems, in this case, vibrate in a global manner and the system behaviour should not be interpreted in terms of coupled 'local' subsystems. It is suggested that the practice of subdivision may only be justified within limited frequency band.

Apart from subdividing a system, solution to the spatial information could also be pursued through post-processing the SEA result. This, in turn, relies on a priori knowledge of the acoustic characteristics of the subsystems. This supplementary information,

* Corresponding author at: Department of Mechanical and Aerospace Engineering, The Hong Kong University of Science and Technology, Clear Water Bay, Kowloon, Hong Kong Special Administrative Region.

E-mail address: chengyang@ust.hk (C. Yang).

http://dx.doi.org/10.1016/j.apacoust.2016.06.004
0003-682X/

in fact, could be acquired from the subsystem itself. For instance, the interior noise problem of a vehicle in which the main body features an acoustic cavity of elongated shape could benefit from acoustic knowledge on corridors. It is found, in room acoustics, that the acoustic pressure in a room that has one dimension dominant over other dimensions exhibits a certain decay pattern, starting from the point where the source is placed [11]. Forssén et al. [12] made use of this information and introduced a constant decay relation, derived by Redmore [13] for a corridor, to the standard SEA model to predict the acoustic pressure level in a railway train. Validation on a scale model showed that the modified SEA model could be able to visualize the acoustic variation across the train compartments. Despite the improved accuracy, discrepancy was still identified at the terminating boundaries where the effect of end walls of the enclosure is substantial. Indeed, Kang [14] demonstrated that in long enclosures the reverberation time increases sharply to a maximum and then decreases slightly with the increase of source-to-receiver distance. This variation eventually results in a longitudinal acoustic pressure decay expression different from that derived by Redmore. Picaut et al. [15] then developed an expression, based on a diffusion model, to quantify the variation of the acoustic energy in a long room, with experimental result showing that the model allows for the spatial variation to be predicted with a better precision.

In this paper, an improved SEA model would be developed by introducing the spatial sound pressure decay relation [15] into the standard SEA framework for predicting the interior noise of a space station of elongated shape. Inheriting the appealing feature of the SEA, the modified approach allows an efficient handling of the complex system whilst providing sound pressure variation within the main cavity. The paper is structured as follows: Section 2 gives a description of the prototype of the space station for scale model validation; a modified SEA model to which the spatial variation relation of the acoustic pressure is introduced would be developed in Section 3; in Section 4, the model is validated in the scale model and conclusions are drawn in Section 5.

2. Description of the space station model

A 1:5 scale model of a space station is depicted in Fig. 1. The major part of the prototype consists of two rectangular air cavities aligned in a row, representing the crew compartment in a space station. In view of the similar cross sections of the two air cavities, it is assumed that they constitute a single long air cavity, with its length being the sum of the lengths of the two cavities and the cross-sectional area being the average of those of the two cavities. The long air cavity is coupled, through a circular opening with a radius of 0.07 m, to an irregular-shaped cavity on top inside which a loudspeaker is placed to represent the noise emitted by the air ventilation system, a major noise generator in space stations [16]. Except the arc part, which is made of a 10 mm thick aluminum structure, the walls enclosing the air volume are made of 30 mm thick acrylic panels. A Cartesian coordinate system is used with the origin set at the middle of the far end wall of the long cavity. The geometry information about the two air volumes is given in Table 1.

Table 1
Geometry information of the two air volumes.

	Volume	Surface area
Long air cavity	$1.8 \times 0.35 \times 0.4\ m^3$	$2.98\ m^2$
Irregular air cavity	$0.027\ m^3$	$0.584\ m^2$

3. Formulation of the problem

For calculating the system response using SEA, the whole prototype is divided into two subsystems, being respectively, the irregular air cavity (subsystem 1) and the long air cavity (subsystem 2). The former, hosting a noise source inside, has an irregular shape, while the latter is a rectangular cavity having one dimension much larger than the others. The two subsystems are coupled through an acoustic opening that represents the passage for air ventilation, and it is assumed that the acoustic energy is by no means transmitted through other paths but this opening. The objective of this work is to predict the acoustic pressure level in the cabin and its spatial variation in the longitudinal direction. This is accomplished by, first solving the standard SEA model for subsystem energy of the cabin, and then introducing a spatial variation relation, based on the expression of the acoustic energy decay in a long cavity developed by Picaut et al. [15], to the subsystem.

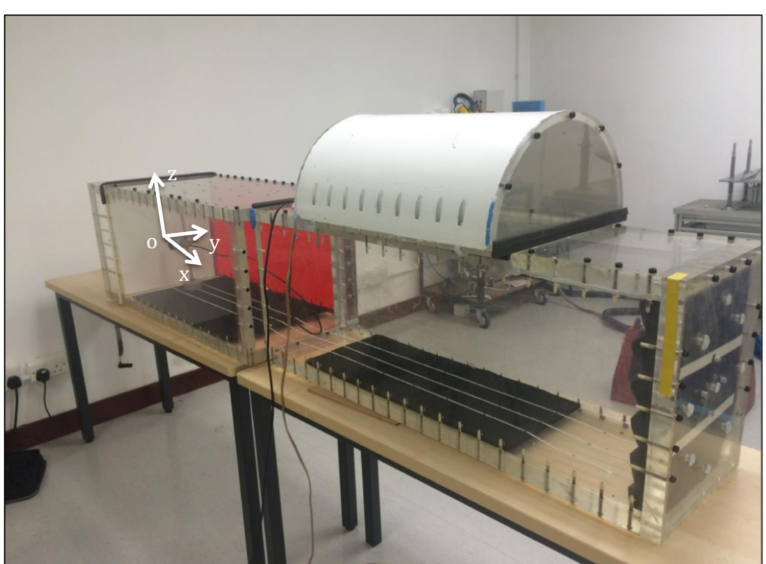

Fig. 1. The 1:5 scale model of the space station prototype.

3.1. SEA formulation

According to the standard SEA procedure [8,9], the power balance equations for the subsystems are written as

$$W_1 + \omega\eta_{21}E_2 = \omega\eta_{1d}E_1 + \omega\eta_{12}E_1 \quad (1a)$$
$$\omega\eta_{12}E_1 = \omega\eta_{2d}E_2 + \omega\eta_{21}E_2 \quad (1b)$$

where W_1 is the source power input into subsystem 1, E_i is the acoustic energy of subsystem i, η_{id} is the internal loss factor of subsystem i, and η_{ij} is the coupling loss factor between subsystems i and j. The terms on the left hand side of each equation represent the power injected into the subsystem and those on the right hand side represent the power exiting from the subsystem.

The internal loss in the subsystem is attributed to the energy dissipation at the boundaries, thus η_{id} could be obtained from standard room acoustics [11], such that

$$\eta_{id} = \frac{13.7 S_i \overline{\alpha_i}}{f V_i} \quad (2)$$

where S_i and V_i are the total surface area and the volume, respectively, of subsystem i, and $\overline{\alpha_i}$ is the average absorption coefficient in subsystem i.

The coupling loss factor η_{ij} that accounts for the fraction of energy transmitted from subsystem i to subsystem j in one radiation cycle could be expressed as

$$\eta_{ij} = \frac{cS'}{8\pi f V_i} \quad (3)$$

where S' is the area of the opening. The coupling loss factors η_{12} and η_{21} satisfy the consistency relation [9]

$$n_1\eta_{12} = n_2\eta_{21} \quad (4)$$

where n_i is the modal density of subsystem i, which can be obtained as [17]

$$n_i = \frac{4\pi f^2 V_i}{c^3} + \frac{\pi f S_i}{2c^2} + \frac{L_i}{8c} \quad (5)$$

In Eq. (5), L_i is the total length of the edges of subsystem i. Rewriting Eqs. (1a) and (1b) in a matrix form yields

$$\begin{bmatrix} \eta_{1d}+\eta_{12} & -\eta_{21} \\ -\eta_{12} & \eta_{2d}+\eta_{21} \end{bmatrix} \begin{bmatrix} E_1 \\ E_2 \end{bmatrix} = \begin{bmatrix} W_1/2\pi f \\ 0 \end{bmatrix} \quad (6)$$

Eq. (6) can be solved for the acoustic response of each subsystem in terms of the acoustic energy in an ensemble sense. This quantity, however, provides no information about the spatial change in acoustic energy within the subsystem. Hence, additional effort has to be made to recover the spatial variation information. In what follows, a description is given of the modification made to the SEA subsystem response for attaining the spatial variation information.

3.2. Acoustic decay in a long cavity

Picaut and his coworkers [15] used the diffusion model to develop an expression for predicting the spatial variation of the acoustic energy in a room that has its length much larger than its width and height. In connection with this particular geometry, assumption is made that the variation of acoustic energy is significant in the longitudinal direction but is negligible over the cross section. In a stationary state, the acoustic energy density $W(x)$ of the diffuse field in the cavity is written as

$$W(x) = \sum_{n=1}^{\infty} \frac{A_n}{u_n} \frac{u_n\cos(u_n x/l_x) + B_2\sin(u_n x/l_x)}{\frac{2h(l_y+l_z)}{l_y l_z} + \left(\frac{u_n}{l_x}\right)^2 D} \quad (7)$$

where

$$\frac{A_n}{u_n} = \frac{W_0[u_n\cos(u_n x_0/l_x) + B_2\sin(u_n x_0/l_x)]}{\frac{l_x}{2}\left[u_n^2 + B_2 + B_2^2 + \cos^2\left[u_n\left[\frac{(B_1+B_2)(u_n^2-B_2)}{u_n^2-B_1B_2} - 2B_2^2\right]\right]\right]} \quad (8)$$

In the above expressions, l_x, l_y and l_z are the lengths of the rectangular cabin in the x, y and z directions, respectively; x_0 is the location of the acoustic source in the x direction; and W_0 is a constant dependent on the power strength of the acoustic source. B_1 and B_2 are called Biot's numbers and are defined as

$$B_1 = \frac{h_1 l_x}{D} \text{ and } B_2 = \frac{h_2 l_x}{D} \quad (9)$$

And

$$h = \frac{c\alpha}{4}, h_1 = \frac{c\alpha_1}{4} \text{ and } h_2 = \frac{c\alpha_2}{4} \quad (10)$$

where α is the absorption coefficient of the side walls, and α_1 and α_2 are those of the two end walls. The diffusion coefficient D is

$$D = \frac{\lambda c}{3} \quad (11)$$

where $\lambda = 4V/S$ is the mean free path in subsystem 2. Eq. (7) describes the attenuation of the acoustic energy, from the position at which the acoustic source is located, in a long rectangular room of finite length. It shows that the extent to which the attenuation occurs in the subsystem depends on the absorption at the boundaries as well as its geometry.

3.3. Spatial variation of the acoustic energy in the cabin

Integrating Eq. (7) with respect to x gives the total acoustic energy in subsystem 2

$$E_2 = \int_0^{l_x} W(x)dx \quad (12)$$

The above equation could be solved for the constant W_0. By substituting W_0 back into Eq. (7), the acoustic energy variation in subsystem 2 could be known. The acoustic pressure level relative to the average pressure in the source cavity (subsystem 1) is

$$L_p(x) = 10\log_{10}\left(\frac{V_1 W(x)}{l_y l_z E_1}\right) \quad (13)$$

4. Experimental results

Before model validation, efforts are first made to assess the absorption of the two subsystems. This is accomplished by performing reverberation time measurement in each subsystem. For the result to be valid, the acoustic opening that connects the two air volumes is sealed to create an acoustic chamber. A Larson Davidson model 831 sound level meter is used to record the decay time, with 6 combinations of loudspeaker and microphone positions in subsystem 1 and 12 combinations in subsystem 2, to obtain the average decay times. The data with 20 dB drop in acoustic energy is extrapolated to T_{60}. By substituting the data into Sabine formula, i.e. $\overline{\alpha_i} = 0.16V_i/T_{60}S_i$, the averaged absorption coefficient in each subsystem could be known and is given in Fig. 2. Those data are to be used to calculate the internal loss factor of the SEA subsystems. In the meantime, the average absorption coefficient in subsystem 2 is substituted into Eq. (10) to get the spatial decay relation. For simplicity, the absorption coefficients of the side walls and the two end walls of the long air cavity are assumed to be identical and equal to that measured in the reverberation room test.

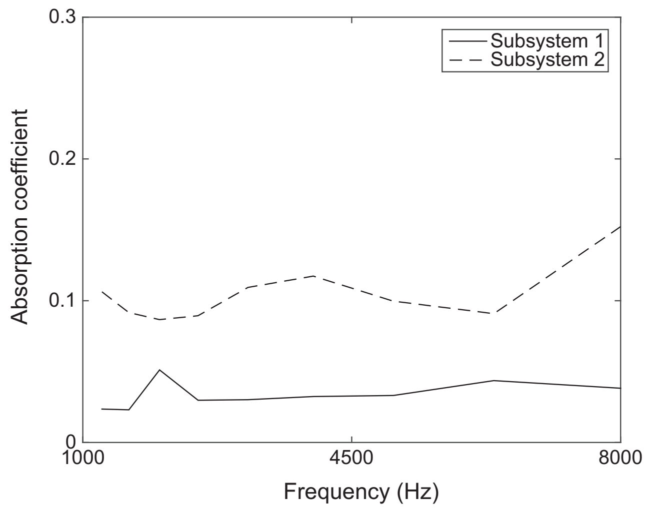

Fig. 2. Measured average acoustic absorption coefficient of each subsystem.

The frequency band of interest in a full-scale space station is from 250 Hz to 1600 Hz, a region which is important for the speech intelligence of the astronauts on board [16]. This corresponds to a frequency band from 1250 Hz to 8000 Hz in the 1:5 scale model. A Brüel & Kjær 4942 ½″ microphone is used to measure the one-third octave band acoustic pressure level in the air volumes. To visualize the acoustic variation in the longitudinal direction of the long air cavity, a total of 9 sampling points, equally spaced at an interval of 0.2 m, are chosen in the lengthwise dimension, with the first point being located at $x = 0.05$ m. For the acoustic pressure level at each sampling point, a mean value is obtained by averaging over 5 points on a cross section perpendicular to the x axis. The coordinates of the 5 points in the y–z plane are, respectively, (0.13, 0.13), (0.13, −0.13), (−0.13, 0.13), (−0.13, −0.13) and (0, 0).

In the work of Forssén et al. [12], a decay relation is introduced to the SEA subsystem to get the spatial variation information. The relation, derived by Redmore [13], is expressed as

$$\Delta L_p = \frac{10}{\ln 10} \frac{\pi}{8} \frac{U\alpha}{S} \tag{14}$$

where U and S are the perimeter and the area of the cross section of a long room. Eq. (14) indicates that the acoustic pressure level varies at a constant rate, the extent of which depends on the geometry of the cross section and the absorption of the room.

The relative acoustic pressure level, L_p, measured inside the prototype at different frequency bands is shown in Fig. 3. At 1250 Hz and 1575 Hz, the decay pattern is less obvious, as evidenced by the moderate fluctuation across the long air cavity, due to the domination of the modes that results in a relatively strong modal feature in the acoustic field. When frequency increases, the decay pattern becomes obvious, initiating from the position close to the opening ($x = 1.16$ m). The L_p is also predicted, respectively, by the current model using Eq. (12) and the Forssén model using Eq. (13), and is depicted in the figures. It can be seen that, around the opening, the Forssén model yields an acoustic pressure level higher than that of the current model. Away from the opening position, the two curves approach gradually to each other and undergo similar decay rate. It is in this region that L_p decreases with respect to the distance at a constant decay rate. However, discrepancy is noticed in the region close to the end wall of the long air cavity, where the L_p calculated by the current model decreases with a smaller decay rate. This phenomenon can be ascribed to the property involved in the decay expression given by Picaut et al. [15]. In contrary, the Forssén model results in a value smaller than that predicted by the current model due to the constant decay rate used in his model. Generally speaking, a good agreement between the measurement result and the results predicted by both models is found in the region where the acoustic pressure level decays at a constant rate. Near the end walls where the reflection effect is substantial, both models underestimate the acoustic pressure level, but the current model yields less discrepancy.

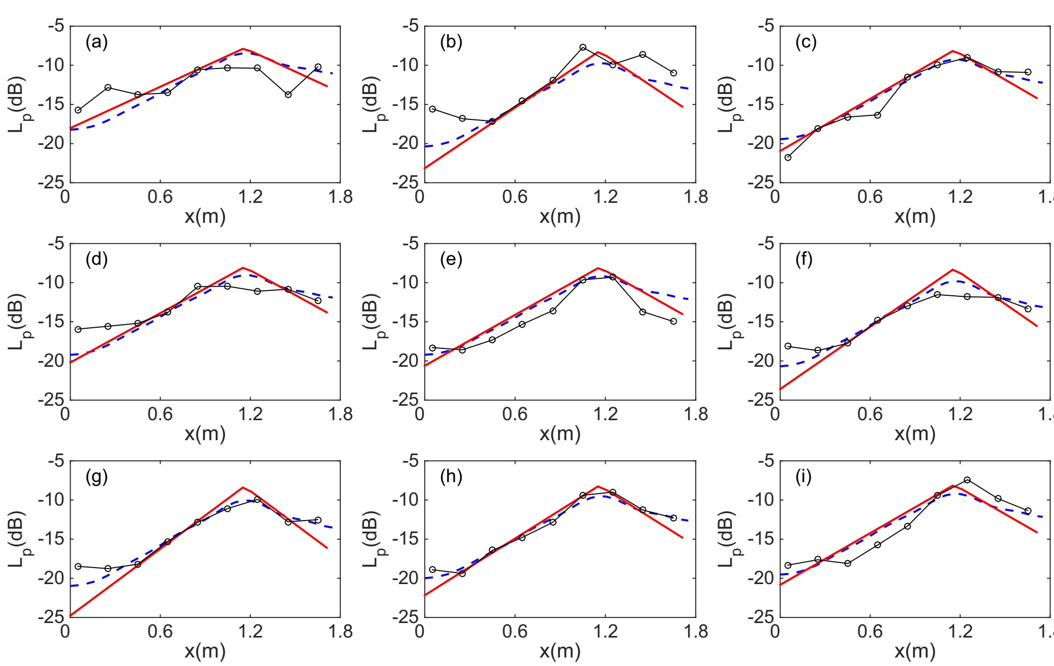

Fig. 3. Relative acoustic pressure level, L_p, measured in the scale model (circle) and predicted by the proposed model (dash) and the Forssén model (solid). (a) 1250 Hz; (b) 1575 Hz; (c) 2000 Hz; (d) 2500 Hz; (e) 3150 Hz; (f) 4000 Hz; (g) 5000 Hz; (h) 6250 Hz; (i) 8000 Hz.

5. Conclusions

A modified SEA model is developed to estimate the interior noise of an acoustic cavity of elongated shape. The model offers a convenient extension to standard SEA model that fails in providing spatial distribution information within individual SEA subsystems. Validation on a scale space station model shows a good agreement between the experimental and predicted results, indicating the potential of the proposed method in visualizing the variation of the acoustic energy across a long air cavity, enriching the result obtained from standard SEA model. Comparing with the model in the literature, the proposed model truthfully depicts the decay pattern in the region near end walls, in agreement with the measurement. However, due to the use of the global absorption coefficient of the subsystem in the calculation, the decay rates at the end walls are not estimated accurately. Further effort should be made to take into account the absorption coefficient at the end walls to improve the prediction accuracy.

Acknowledgement

A research fund from China Academy of Space Technology (CAST) is acknowledged. The authors would like to extend thanks to Mr. Xinming Li, from CAST, for coordinating the fabrication and the logistics of the prototype.

References

[1] Bernhard RJ, Gardner BK, Mollo CG, Kipp CR. Prediction of sound fields in cavities using boundary-element methods. AIAA J 1987;25(9):1176–83.
[2] Richards TL, Jha SK. A simplified finite element method for studying acoustic characteristics inside a car cavity. J Sound Vib 1979;63(1):61–72.
[3] Miller VR, Faulkner LL. Prediction of aircraft interior noise using the statistical energy analysis method. J Vib Acoust Stress Reliab 1983;105(4):512–8.
[4] Maxit L, Yang C, Cheng L, Guyader JL. Modeling of micro-perforated panels in a complex vibro-acoustic environment using patch transfer function approach. J Acoust Soc Am 2012;131(3):2118–30.
[5] Riou H, Ladevèze P, Kovalevsky L. The variational theory of complex rays: an answer to the resolution of mid-frequency 3D engineering problems. J Sound Vib 2013;332(8):1947–60.
[6] Reynders E, Langley RS, Dijckmans A, Vermeir G. A hybrid finite element–statistical energy analysis approach to robust sound transmission modeling. J Sound Vib 2014;333(19):4621–36.
[7] Gerges Y, Hwang HD, Ege K, Maxit L, Sandier C. Vibroacoustic modeling of a trimmed truck cab in the mid frequency range. InterNoise 2015, San Francisco.
[8] Lyon RH. Statistical energy analysis of dynamical systems: theory and applications. MIT Press; 2003.
[9] Craik RJ. Sound transmission through buildings: using statistical energy analysis. Gower Publishing Company; 1996.
[10] Fahy FJ. A note on the subdivision of a volume of air in a vehicle enclosure into sea subsystems. J Sound Vib 2004;271(3):1170–4.
[11] Kuttruff H. Room acoustics. 5th ed. CRC Press; 2009.
[12] Forssén J, Tober S, Corakci AC, Frid A, Kropp W. Modelling the interior sound field of a railway vehicle using statistical energy analysis. Appl Acoust 2012;73(4):307–11.
[13] Redmore TL. A theoretical analysis and experimental study of the behaviour of sound in corridors. Appl Acoust 1982;15(3):161–70.
[14] Kang J. Reverberation in rectangular long enclosures with geometrically reflecting boundaries. Acta Acust United Ac 1996;82(3):509–16.
[15] Picaut J, Simon L, Polack JD. Sound field in long rooms with diffusely reflecting boundaries. Appl Acoust 1999;56(4):217–40.
[16] Zhang F, Yang J, Feng YQ. Current state of art and suggestions about noise evaluation and control for the space station. Space station Environment Engineering; 2014.
[17] Morse PM, Bolt RH. Sound waves in rooms. Rev Mod Phys 1944;16(2):69–150.