Applied Acoustics 120 (2017) 34–44

Contents lists available at ScienceDirect

Applied Acoustics

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

Sound insulation analysis and optimization of anti-symmetrical carbon fiber reinforced polymer composite materials

Zhe Zhang [1], Yu Du *

School of Automotive Engineering, Dalian University of Technology, Dalian 116024, China

ARTICLE INFO

Article history:
Received 8 March 2016
Received in revised form 29 November 2016
Accepted 4 January 2017

Keywords:
CFRP
Anti-symmetrical
Sound insulation
Optimization
Lightweight

ABSTRACT

Due to the advantages of high strength and low density, carbon fiber reinforced polymer (CFRP) composite material has become more and more attractive given the increasing demand of lightweight design requirement in the automobile industry. However, as density reduces, the sound insulation ability of CFRP panels is usually inferior compared with traditional metal materials such as steel and aluminum, which may lead to deteriorated noise insulation performance inside the vehicle cabin when the CFRP material is used as body panels. For the sake of investigating the sound insulation ability of CFRP, two optimization methods are proposed, and the optimal ply angles that lead to the largest sound transmission loss in two different frequency ranges are obtained accordingly. On the basis of optimization, the sound insulation capability of CFRP and two metal materials are thoroughly compared. It is found that, when optimally designed, the CFRP panel can achieve a transmission loss level comparable to its metal counterparts at low frequencies (e.g., <300 Hz) with substantially less weight.

1. Introduction

Due to the unique mechanical property of exceptionally high strength-to-weight ratio, carbon fiber reinforced polymer (CFRP) has found extensive applications in structures where light weight and high mechanical strength are required simultaneously. Typical applications of CFRP include military and commercial aircrafts, satellites and high-end race cars, etc. When properly designed, parts made of continuous CFRP are about 75% and 40% lighter than those made of steel and aluminum materials, respectively. In the automotive industry, as weight reduction directly leads to improved fuel economy and reduced emission, the high strength-to-weight ratio feature is a significant motivation for automakers to increase the use level of CFRP in vehicle designs. However, from the standpoint of improving the noise, vibration and harshness (NVH) performance, the lightweight nature of the CFRP material may limit its application in vehicle structures as the noise isolation capability or sound transmission loss (TL) is typically proportional to the density of the material [1]. It is therefore important to investigate engineering approaches to optimize the sound isolation performance of structures (e.g., panels) made of CFRP while still maintain the lightweight feature.

The interest in understanding the acoustical properties of CFRP panels has attracted many research efforts since nearly half a century ago, when its exceptional mechanical properties were first noticed. Lin and Wang established a sound transmission loss model applicable for composite laminated panels (CLPs) made of orthotropic materials with the application of transfer matrix [2]. The influence of thickness and ply angle of different laminates upon the sound insulation ability of CLPs was investigated. Lu deducted the expression of the sound transmission loss for CLPs by taking advantage of its difference in mechanical impedance from isotropic panels [3]. Using Mindlin panel theory and Rayleigh surface integral equation, Li and Zhao established a sound transmission loss model of CLPs embedded in an infinite baffle with plane waves obliquely impinging upon its surface [4]. Based on this model, a series of studies were conducted to investigate the influences of various design factors of CPLs on the TL. They concluded that the sound insulation ability of CPLs varies with different ply angles.

In addition to the single layer composite panel, the sound transmission characteristics of other type of composite panels have also been extensively studied. For example, Shen et al. developed models for theoretically calculating the sound transmission loss of single laminated composite plate reinforced by orthogonal stiffeners [5,6]. To further increase the sound isolation performance of lightweight structure, double-panel structures are commonly used. Xin and Lu presented some analytical and experimental results showing the boundary condition influences on the transmission loss of

* Corresponding author.
E-mail address: yuduvt@gmail.com (Y. Du).
[1] Current address: Roslagstullsbacken 5, room 1102, Stockholm, Sweden.

http://dx.doi.org/10.1016/j.apacoust.2017.01.003

double panels separated by an air cavity [7]. Zhou et al. investigated the sound transmission performance of laminated composite double panels lined with poroelastic material [8,9]. In some recent studies, Shen, Xin and Lu gave some results on the sound transmission across composite laminate double-panel structures stiffened by ribs [10–13]. For either the single-panel or the double-panel structure, existing studies mentioned above demonstrated that the ply angle of the composite laminate layers play an important role in the sound transmission property, particularly in the stiffness controlled region. However, to the best of the authors' knowledge there seems to be a lack of discussions focusing on the direct comparison of the sound insulation performance of panels made of CFRP and metal materials, especially in light of the background of lightweight applications.

In this study, the sound TL of CFRP panels with anti-symmetrical laminate structures is studied and compared with its metal counterparts. For this purpose, a numerical model applicable for simply-supported CFRP panels is firstly established and verified. Then, two optimization strategies are proposed to calculate optimal ply angles to maximize the TL of anti-symmetrical CFRP panels in different frequency ranges. Based on the proposed optimization methods, the sound insulation ability of anti-symmetrical CFRP panels under different design strategies are compared with panels made of two common metal materials - aluminum and steel. It is demonstrated that the CFRP panel, when optimally designed, can achieve a sound TL level comparable to its metal counterparts at low frequencies (e.g., <300 Hz) with substantially less weight. However, as well-known, because of the low mass density, sound TL values of CFRP panels are in general inferior to those of metal panels at mid-high frequencies where the mass law starts dominating.

2. Model development

In this section, a finite element (FE) model for calculating the sound TL of simply supported panels is developed and verified experimentally. This model is subsequently used in Section 3 to justify the analytical TL model based on which the optimal ply angle designs are conducted for CFRP panels in various frequency ranges.

2.1. Finite element model for calculating the sound transmission loss

A FE model of a CFRP panel with the size of $800 \times 700 \times 0.8$ mm mounted on a rectangular acoustic test box (ATB) made of medium-density fiberboard (MDF) is established in Virtual.Lab. As illustrated in Fig. 1, the dimension of the ATB is $1000 \times 800 \times 700$ mm; five of its six faces (marked in blue) of the ATB are 18 mm thick MDF while the last face (green color) is sealed with a 0.8 mm thick simply supported CFRP panel. In the numerical model, both the CFRP panel and the MDF faces are meshed with 2D quadrilateral shell elements with a size of 5×5 mm. In Virtual.Lab, the automatically matched layer (AML) is applied on the outer surface of the air volume (a hollow cubic mesh of air space enclosing the entire ATB, not shown in the figure). The use of the AML makes it possible for the air volume not necessarily to include the sound source, which significantly improves the computation efficiency [14]. The "no reflection" boundary condition is applied on the inner surfaces of the five MDF faces so that the internal acoustic field of the ATB can be considered as anechoic while the field outside the ATB is assumed to be reverberant in the model. As shown in Fig. 1, a monopole sound source is positioned 2500 mm away and directly in front of the center of the CFRP panel. The sound pressure levels at two measurement points, one at the center of the ATB chamber and the other at a point 50 mm away from the center of the CFRP panel outside the chamber are calculated. The sound transmission loss of the CFRP panel can then be derived. The frequency range considered in the model is from 0 to 3000 Hz.

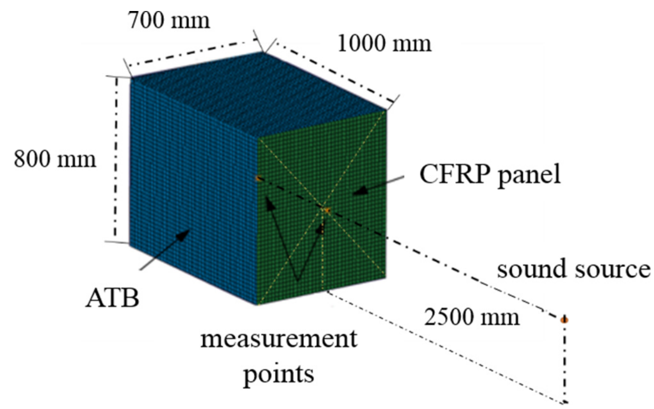

Fig. 1. Finite element model of the acoustic test box (ATB).

2.2. Model validation and simplification

To verify the feasibility of the numerical FE model, a sound insulation test is carried out. As illustrated in Fig. 2, the test setup is prepared based on the configuration of the numerical model described in Section 2.1. A thin flat CFRP panel is mounted on a specially designed rectangular shaped acoustic test box (ATB), whose dimensions are $1000 \times 800 \times 700$ mm. Five of its six faces are made of MDF boards with a thickness of 18 mm. These MDF panels are fairly massive and can be considered rigid in practice. To minimize the sound reflection inside the cavity, the inner surfaces of the five MDF panels are lined with 5 mm thick acoustic foam. The sixth face is sealed with the simply-supported panel under test (PUT). As previously described in Fig. 1, the sound insulation capability of the PUT can be evaluated using an external sound source and two measurement microphones positioned outside and inside of the ATB. By definition, the difference in sound pressure levels measured by the two microphones is the sound TL of the PUT.

Using the test setup shown in Fig. 2, the TL of a 4-layer CFRP panel, a steel panel and an aluminum panel are measured. All three panels under test are 800 mm long and 700 mm wide with a thickness of 0.8 mm. Their material properties are listed in Table 1.

Fig. 3 compares the sound transmission loss of the CFRP PUT in the frequency range between 100 and 3000 Hz obtained from the experiment and numerical simulation (the blue and green curves in the plot, respectively). It can be seen that the two curves match with each other fairly well except that there is a peak predicted by the numerical curve around 1200 Hz. The experimental versus numerical comparison is shown in Fig. 4 for the steel and aluminum panels. For the two metal panels, it is seen that simulation results predicted by the FE model match the experimental curves fairly reasonably at all frequencies up to 3000 Hz.

One reason that could contribute to the discrepancy between the experimental and the numerical results around 1200 Hz in Fig. 3 is the lack of the knowledge on design details of the experimental CFRP panel. The CFRP panel used in the experiment was a commercial off-the-shelf panel whose dimensions were measured but other design features (e.g., anti-symmetrical vs. symmetrical, ply angle of each laminate layer) were not known. On the other hand, the CFRP panel in the numerical model is assumed to be of the anti-symmetrical type with four laminate layers of 0.2 mm thick each and a ply angle arrangement of $45°/-45°/45°/-45°$. In