THE EFFECT OF SOUND AND VIBRATION CHARACTERISTICS OF CARBON FIBER REINFORCED PLASTIC COMPOSITE PLATE ON INTERIOR NOISE BY CARBON FIBER LAMINATION ANGLE

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Recently, research on the weight lightening of the vehicle body for increasing the fuel efficiency of automobile have been actively carried out. One way to reduce the weight is to change the material applied to the car body. In recent years, CFRP, which has lower density and a similar strength to conventional metals, is increasingly applied to the car body. However, since CFRP has different sound and vibration characteristics compared with conventional metal materials, it needs research and improvement. This study discusses the effect of sound and vibration characteristics of CFRP plates according to the fiber lamination angle on Interior noise. Through this study, it is aimed to derive the angle of carbon fiber arrangement suitable for vehicle application by grasping sound and vibration characteristics according to fiber lamination angle.
Keywords: CFRP, Sound and vibration, Interior noise, Lamination angle

1. Introduction

Recently, CFRP(Carbon Fiber Reinforced Plastic) has been used in many transportation sectors such as aerospace, automobiles and ships. CFRP is a suitable material for the weight lightening because it has similar strength and lower density than metal materials. And CFRP has different characteristics according to the carbon fiber lamination angle because it has anisotropic properties. However, the weight lightening using CFRP causes some noise and vibration problems. So it is important that the research about the characteristics of noise and vibration of CFRP plates, recently. In the previous study, we observed the basic characteristics of noise and vibration of baffled CFRP plates according to the carbon fiber lamination angle [1]. On this study, we try to understand the influence of interior noise through numerical solution, experiment and analytic method [2,3] by impact testing based on the characteristics of noise and vibration according to the lamination angles of CFRP. This study can be the basis of identifying vehicle NVH characteristics when CFRP plates are applied to vehicles.

2. Theory

2.1 Cavity mode

The acoustic resonant frequency of a rigid-walled rectangular cavity is given by:

$$f_{l,m,n} = \left(\frac{c_0}{2}\right)\sqrt{\left(\frac{l}{L_x}\right)^2 + \left(\frac{m}{L_y}\right)^2 + \left(\frac{n}{L_z}\right)^2} \quad . \tag{1}$$

where c_0 is the speed of sound in air, *l*, *m* and *n* are integers and L_x, L_x and L_x are the dimensions of the rectangular cavity in the *x*, *y* and *z* coordinate directions.

2.2 Vibration of plates

The equation of motion for the vibration of CFRP plate is given by [4]:

$$D_{11}\frac{\partial^4 w}{\partial x^4}+4D_{16}\frac{\partial^4 w}{\partial x^3\partial y}+2(D_{12}+2D_{66})\frac{\partial^4 w}{\partial x^2\partial y^2}+4D_{26}\frac{\partial^4 w}{\partial x\,\partial y^3}+D_{22}\frac{\partial^4 w}{\partial y^4}+C_d\frac{\partial w}{\partial t}+\rho h\frac{\partial^2 w}{\partial t^2}=f(x,y,t)\,. \quad (2)$$

The natural frequency of the plate corresponding to the mode shape is expressed as follows:

$$f_{mn}=\frac{1}{2\pi}\sqrt{\frac{\left(D_{11}I_1I_2+4D_{16}I_3I_4+2(D_{12}+2D_{66})I_5I_6+4D_{26}I_7I_8+D_{22}I_9I_{10}\right)}{\rho h I_2 I_9}}\,. \quad (3)$$

where *m* and *n* mean *x* and *y* directional modal lines respectively.

2.3 Structural-acoustic coupled system

The analytic method of acoustic pressure in structural-acoustic coupled system is given by [2,3]. The acoustic pressure and the structural vibration are described by the summation of *N* and *M* modes, respectively. Hence, both the acoustic pressure ***p*** at **x** inside the enclosure and the structural vibration velocity ***u*** at **y** are given by:

$$p(\mathrm{x},\omega)=\sum_{n=1}^{N}\psi_n(\mathrm{x})a_n(\omega)=\Psi^{\mathrm{T}}\mathrm{a}\,. \quad (4)$$

$$u(\mathrm{y},\omega)=\sum_{m=1}^{M}\phi_m(\mathrm{y})b_m(\omega)=\Phi^{\mathrm{T}}\mathrm{b}\,. \quad (5)$$

where the *N* length column vectors **Ψ** and **a** consist of the array of uncoupled acoustic mode shape functions $\psi_n(\mathrm{x})$ and the complex amplitude of the acoustic pressure modes $a_n(\omega)$. Likewise, the M length column vectors **Φ** and **b** consist of the array of uncoupled vibration mode shape functions $\phi_m(\mathrm{y})$ and he complex amplitude of the vibration velocity modes $b_n(\omega)$, respectively. The superscript *T* denotes the transpose. The modal acoustic pressure vector **a** and the modal vibration amplitude vector **b** can be expressed as:

$$\mathrm{a}=(\mathrm{I}+\mathrm{Z_a C Y_s C^T})^{-1}\mathrm{Z_a}(\mathrm{C Y_s g})\,. \quad (8)$$

$$\mathrm{b}=(\mathrm{I}+\mathrm{Y_s C^T Z_a C})^{-1}\mathrm{Y_s}(\mathrm{g})\,. \quad (9)$$

where the ($N \times M$) matrix **C** is the structural-acoustic mode shape coupling matrix. $\mathbf{Z}_a=\mathbf{A}\rho_0 c_0/V$ is an ($N \times N$) matrix defined as the uncoupled acoustic modal impedance matrix. The uncoupled modal impedance matrix is diagonal because of the orthogonal property of uncoupled modes. The matrix **A** is a ($N \times N$) diagonal matrix in which each (*n*, *n*) diagonal term consists of $A_n(\omega)$. **g** is the generalized modal force vector due to the external force distribution $f(\mathrm{y},\omega)$. $\mathbf{Y}_s=\mathbf{B}/\rho_s h S_f$ is the ($M \times M$) diagonal matrix defined as the uncoupled structural modal mobility matrix. The matrix **B** is a ($M \times M$) size diagonal matrix in which each (*m*, *m*) diagonal term consists of B_m, $\mathbf{C}^{\mathrm{T}}$ is the transpose matrix of **C**, and the *M* length vector **g** is the generalized modal force vector due to the external force distribution $f(\mathrm{y},\omega)$.

3. Numerical solution and experiment

3.1 Numerical solution

The mode shape, natural frequencies, interior sound pressure of the CFRP plates were calculated by numerical simulation. The FE models of the plate and acoustic cavity were created and the mode shape and natural frequency were calculated by inputting properties of CFRP. Based on the generated data, the acoustic response of the point force on the structural plate was obtained by coupling structural plate with the acoustic cavity. All of these processes use Finite Element Method(FEM) and the software used is Siemens' Virtual Lab.

3.2 Experiment

The plates used in experiment are six rectangular CFRP plates with each carbon fiber lamination angles [-15°,15°], [-30°,30°], [-45°,45°], [-60°,60°], [-75°,75°], [0°,90°]. The dimensions of the plate are 0.7×0.5 m. Four sides of the plate are fixed on one side of a rectangular enclosure and a microphone is installed at an internal point 200mm away from the center of the plate. The dimensions of the rectangular cavity are $0.7 \times 0.5 \times 1.175$ m and the thickness is 25mm.

4. Results

4.1 Mode shape & natural frequency of CFRP

The results of the FEM numerical solution of the mod shapes and natural frequencies where the structure plate and the acoustic cavity are coupled were compared. The 1st mode and the 2nd mode were generated the same (1,1) mode shape and (2,1) mode shape at all lamination angles, respectively. However, [±15°], [±30°], [±45°] and [0°/90°] were generated (1,2) mode shape and [±60°] and [±75°] were generated (3,1) mode shape at the 3rd mode. Therefore, it can be seen that natural frequency and mode shape are different according to the lamination angle.

4.2 Cavity mode

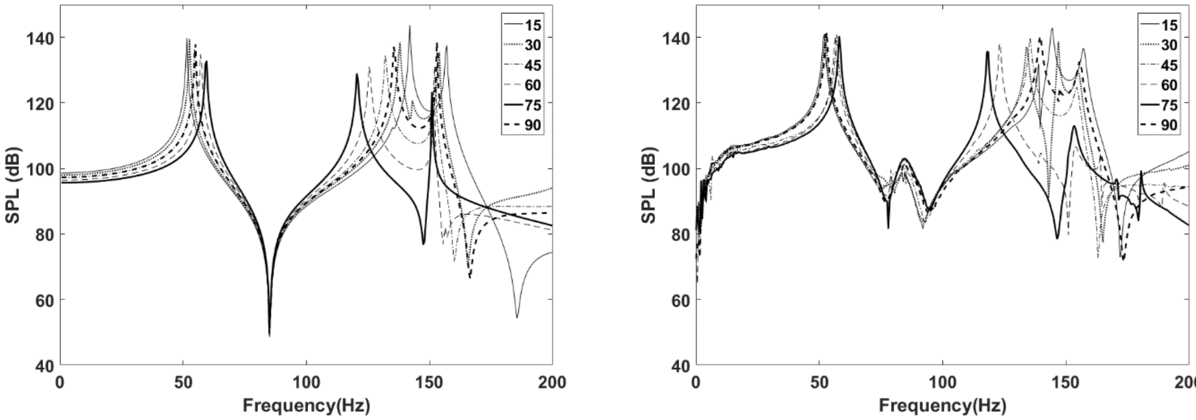
The cavity mode of the rectangular enclosure occurs at 144.68 Hz by Eq. (1). However, the cavity mode in which the structure plate and acoustic cavity are coupled is shifted by the impedance of the structural plate and occurs above 150 Hz as shown in Table 1. The frequency shifting occurs large as the lamination angle increased and the frequency shifting occurs small as the lamination angle decreased.

Table 1: Frequency of cavity mode according to the fiber lamination angle

Cavity mode	±15°	±30°	±45°	±60°	±75°	0°/90°
FEM (Hz)	156.835	153.831	152.164	151.869	151.082	153.072
EXP (Hz)	157	155	154.5	154	153.5	155.5

4.3 Pressure

The results of numerical, experimental and analytic solutions of the interior noise of CFRP according to fiber lamination angle were compared as shown in Fig. 1. The acoustic peaks were observed in non-symmetric modes [(1,1) and (3,1)] and cavity mode. Because the sound pressure was measured at a point vertically offset from the center of the plate by impacting the center of the plate, the peaks of the symmetric mode did not appear. Since the sound pressure measured in experiment did not completely satisfy the above conditions, the peaks of the symmetric mode were appeared small. The difference in sound pressure level at (1,1) mode, which is the first peak near 50 Hz, is not significant. At the (3,1) mode, which is the second peak, the frequency difference of the peak is large and the sound pressure level was measured low as the lamination angle increased. In other words, the low sound pressure level was measured in [±75°] and [±60°] relatively. Likewise, the third peak, cavity mode, showed similar tendency to (3,1) mode. Larger lamination angle occurred at low frequency and was measured low sound pressure level relatively. Overall, low sound pressure level was measured as the lamination angle increased.



(a) (b)

Figure 1: The interior sound pressure level(dB) comparison of CFRP; (a) Numerical solution (FEM), (b) Experiment

5. Conclusion

This study compared the numerical, experimental and analytic methods of the interior noise of the CFRP according to the carbon fiber lamination angle. In the case of CFRP, the difference in natural frequencies and mode shapes according to the fiber lamination angle. This is due to the difference in CFRP properties depending on the carbon fiber lamination angle. These differences caused some differences in interior noise in the rectangular enclosure. Because of the differences both natural frequencies and mode shapes, the peaks of the sound pressure occurred at different frequencies. And the overall sound pressure level values were measured lower as the lamination angle was larger. Therefore, by changing the lamination angle, the resonant phenomenon at a specific frequency can be avoided and the interior noise can be predicted.

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