Control of Structural Acoustic Radiation Based on Topography Optimization

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***Abstract*-Structural acoustic radiation controlling is very important for noise reduction. To minimize the acoustic radiation of folded plates, a topography optimization method is proposed in this paper. In the proposed method, the structural vibration characteristics of a folded plate structure are analyzed by using of FEM. Then the structural acoustic radiation is analyzed by using Helmholtz integral method. The natural frequency of the first mode shape of the folded plate structure is taken as the objective function. The average value of acoustic radiation power in the analysis frequency band can be minimized by maximizing the natural frequency of the first mode of the folded plate structure. The optimization software Altair OptiStruct is used to optimize the design of folded plate structures. Numerical results show that the acoustic radiation power of folded plate structures can be significantly reduced by topography optimization.**

***Index Terms*—finite element method (FEM); boundary element method (BEM); structural acoustic radiation; topography optimization; folded plate structures.**

I. Introdution

In the latest several decades, NVH (Noise-Vibration-Harshness) has become an important indicator of quality and comfort of cars. The structural acoustic performance of a car becomes an important issue in the design process. In car structures, the vibration of plates is not self-generated, but passed from the frame structure. Thus, the frame structure of a car is the important source of vibration and noise.

The purpose of this paper is to show the feasibility of the folded plate design optimization to minimize the car's structural acoustic radiation power. Many numerical methods, such as the finite element method (FEM) [1], [2], the boundary element method (BEM) [3], [4], the statistical energy analysis (SEA) [5], [6] and the energy flow analysis (EFA) [7], have been developed to simulate the structural acoustic performance of a car. Different methods must be used based on the design objective. For example, FEM and BEM can be used for simulation in the low-frequency range, while SEA and EFA can be used for simulation in the high-frequency range. In this study, FEM and BEM are used to calculate the structural acoustic radiation power of folded plate structures. After building the model mesh, a finite element code MSC/NASTRAN [8] is used to analyze the frequency response of a folded plate and a boundary element code SYSNOISE [9] is used to calculate the structural acoustic radiation power and sound pressure level.

Many optimization methods have been proposed to reduce the structural acoustic radiated power. As one of the optimization methods, topology optimization has been extensively applied to a wide variety of structures. The topology optimization problem can be treated as a material redistribution problem to minimize/maximize the certain objective functions. In other words, to achieve the optimization goal bounded by various constraints, the structural material is redistributed. The efficiency of this method was deeply discussed in literatures [10]-[12]. Akl *et al*. [13] have discussed the fluid–structure interaction problem in which a vibrating flexible plate is coupled to a closed acoustic cavity by using the topology optimization and the moving asymptote method. Unlike topology optimization, topography optimization only makes structural shape change to meet the demand of design without making material distribution change. Topology optimization, topography optimization and the combination of both techniques are used to optimize the hard disk drive suspensions structure [14]. The combination of topology and topography optimization techniques can create a product with good shape and stiffness. However, these studies are not applied to the acoustic radiation problem yet. The topography optimization of a transaxle based on the β method was carried out by Dai, Y. and Ramnath, D. to minimize the radiated noise [15].

The folded plates which consist of flat plates exist in the car body, ship hulls, buildings, and box girder bridges widely. Recently, the frequency optimization of the one and two-fold folded laminated plates has been investigated by many researches [16]-[19]. These issues focused on the free vibration, the dynamic behavior and

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bending characteristics of the folded plates. However, the optimization of folded plates to minimize the structural acoustic radiation power has not been investigated yet.

In this paper, a method for the topography optimization of folded plates based on the vibration analysis, FEM, BEM and modal analysis was presented. The objective of topography optimization is to minimize the structural acoustic radiation power of a folded plate structure. In the proposed method, the structural vibration characteristics of a folded plate structure are analyzed by using of FEM. Then, the structural acoustic radiation is analyzed by using Helmholtz integral method. The natural frequency of the first mode shape of the folded plate structure is taken as the objective function. The average value of acoustic radiation power in the analysis frequency band can be minimized by maximizing the natural frequency of the first mode of the folded plate structure. In topography optimization method, only the structural shape is changed to meet the demand of design, while the material distribution of the structure remains unchanged. The shape of the folded plate structure is defined by the finite element nodes. Thus, the nodes in the design region are taken as the design variables. The sequential quadratic programming algorithm is used to minimize the acoustic radiation of the folded plate with different folding angles. Numerical results of the topography optimization of a car body structure model show that the radiated power of plate structures can be significantly reduced after optimization.

II. The Calculation of Structural Acoustic Radiation

Vibration of a structure under an exciting force vector $\left\{F(x,t)\right\}$ with angular frequency ω can be written as

$$[M]\{\ddot{u}\}+[C]\{\dot{u}\}+[K]\{u\}=\left\{F(x,t)\right\} \qquad x\in\Omega^S \quad t>0 \tag{1}$$

where Ω^S is the domain of structure; $\left\{u\right\}$ is the node displacement vector; $\left[M\right]$ is the mass matrix; $\left[C\right]$ is the viscous damping matrix; $\left[K\right]$ is the stiffness matrix; $\left\{F(x,t)\right\}$ is the external exciting force.

Taking the Fourier transform of both sides of the Eq. (1), we can get

$$(-\omega^2[M]+i\omega[C]+[K])\left\{u(\omega)\right\}=\left\{f(\omega)\right\} \tag{2}$$

where $\left\{f(\omega)\right\}$ is the vector of magnitude of harmonic force; $\left\{u(\omega)\right\}$ is the vector of node displacement; ω is the angular frequency of the external exciting force; $i=\sqrt{-1}$ is an imaginary number.

The vector of node velocity $\left\{v(\omega)\right\}$ can be expressed as

$$\left\{v(\omega)\right\}=i\omega\left\{u(\omega)\right\} \tag{3}$$

Multiplying each side of Eq. (3) by the transformation matrix $\left[T\right]$, we can get the normal velocity components on the surface of the acoustic boundary element model

$$\left\{v_n\right\}=i\omega\left[T\right](-\omega^2[M]+i\omega[C]+[K])^{-1}\left\{f(\omega)\right\} \tag{4}$$

where $\left\{v_n\right\}$ is the normal velocity vector on the surface of the structure; $\left[T\right]$ is the function of geometries of the structural model, the acoustic model and the interface between the structural model and the acoustic model.

The acoustic pressure must satisfy the Neumann boundary condition on the surface of the structure

$$\frac{\partial\{p\}}{\partial n}=-i\omega\rho\{v_n\} \tag{5}$$

where ρ is the density of fluid; n is the outer-normal units vector of the structure surface.

When the vibration boundary condition is input to the plate structure, the acoustic pressure at a specified field point B can be calculated by the pressure and normal velocity distribution on the surface, as expressed by the Helmholtz surface integral

$$\{p(B)\}=\iint\limits_S(G(A,B)\frac{\partial\{p(B)\}}{\partial n}-\{p(A)\}\frac{\partial G(A,B)}{\partial n})dS(A) \tag{6}$$

where $\{p(A)\}$ is the sound pressure of point A; S is the structure's surface; G is the free-space Green's function (Fundamental solution of the Helmholtz equation for a point source), which is given as

$$G(A,B)=\frac{1}{4\pi R}e^{-ikR} \tag{7}$$

where $k=\frac{\omega}{c}$ denotes the wave number; ω and c are the circular frequency and speed of sound, respectively; $R=|B-A|$ is the distance between the points A and B; A is a accidental point on the surface of structure; B is any point in space.

Substituting Eq. (5) into Eq. (6), one may get

$$\{p(B)\}=-\iint\limits_S(i\omega\rho\{v_n\}G(A,B)+\{p(A)\}\frac{\partial G(A,B)}{\partial n})dS(A) \tag{8}$$

The evaluation of Eq. (8) was performed using isoperimetric element and numerical integration. For a isoperimetric element, interpolation of the pressure and velocity at element nodes determine the pressure and velocity distributions over entire element

$$\begin{cases} p=\sum_{l=1}^{4}N_l p^l \\ v_n=\sum_{l=1}^{4}N_l v_n^l \end{cases} \tag{9}$$

where $\{p^l\} = [p^1, p^2, p^3, p^4]^T$, $\{v_n^l\} = [v_n^1, v_n^2, v_n^3, v_n^4]^T$; $\{p^l\}$ and $\{v_n^l\}$ represent the pressure vector and normal velocity vector of nodes of a element, respectively.

$N_l = \frac{1}{4}(1+\xi_1\xi)(1+\eta_1\eta)$ is the interpolation shape function, $l = 1, 2, 3, 4$; ξ and η are local coordinates.

Substituting Eq. (9) into Eq. (8) for the nodes of each element, we can obtain the algebraic equation of the structural-acoustic system as following

$$[H_H]\{p\} = [G_H]\{v_n\} \tag{10}$$

where $\{p\} = [p_1, p_2, ..., p_{N_d}]^T$, $\{v_n\} = [v_{n1}, v_{n2}, ..., v_{nN_d}]^T$; N_d is the number of nodes on the surface of structure; $[H_H]$ and $[G_H]$ are the acoustics coefficient matrices.

Based on Eq. (10), the vector of sound pressures can be expressed as

$$\{p\} = [Z]\{v_n\} \tag{11}$$

where $[Z] = [H_H]^{-1}[G_H]$ is the impedance matrix of the structural-acoustic system. Note that, the impedance matrix $[Z]$ is symmetric, namely $[Z] = [Z]^T$ .The structural acoustic radiation power of the ith element can be calculated by using the following formula

$$\begin{aligned} \mathrm{W}_i &= \frac{1}{2}\mathrm{Re}\left(\left\{v_n\right\}^T [Z]_i^T \left(\int_S \left(N^T N\right) dS\right)\left\{v_{in}^*\right\}\right) \\ &= \frac{1}{2}\mathrm{Re}\left(\left\{v_n\right\}^T \left[R_i\right]\left\{v_{in}^*\right\}\right) \end{aligned} \tag{12}$$

where W_i is the structural acoustic radiation power of ith element; $\{v_{in}\}$ is the normal velocity of ith element; N is the interpolation shape function of ith element; $\left[R_i\right]$ is the structural acoustic radiation resistance matrix, which is given by

$$\left[R_i\right] = [Z]_i^T \int_{S_i} \left(N^T N\right) dS \tag{13}$$

The acoustic radiated power from the vibrating panel can be rewritten as

$$\mathrm{W} = \frac{1}{2}\int_S \mathrm{Re}\left\{p v_n^*\right\} dS = \frac{1}{2}\mathrm{Re}\left(\left\{v_n\right\}^T \left[R\right]\left\{v_n\right\}\right) \tag{14}$$

where $\left[R\right]$ is the assembly matrix of $\left[R_i\right]$. $\left[R\right]$ can be written as follows [20]

$$\left[R\right] = \frac{\omega^2 \rho_0 A_e^2}{2\pi c}\begin{bmatrix} 1 & \frac{\sin(kR_{12})}{kR_{12}} & \cdots & \frac{\sin(kR_{12})}{kR_{1r}} \\ \frac{\sin(kR_{21})}{kR_{21}} & 1 & & \\ \cdots & \cdots & \cdots & \cdots \\ \frac{\sin(kR_{r1})}{kR_{r1}} & & & 1 \end{bmatrix} \tag{15}$$

where ρ_0 is the density of fluid, A_e is the area of each element, respectively; R_{ij} is the distance between the centers of ith and jth elements; r is the number of elements of the structure.

The acoustic radiation power from the structure surface is a function of frequency. The external excitation frequency varies over a frequency band, which may include resonant frequencies of the structure. The averaged acoustic radiation power over this frequency band can be obtained by integrating $W(f)$ over the frequency band [21]

$$\overline{W} = \frac{1}{f_{\max} - f_{\min}} \int_{f_{\min}}^{f_{\max}} \mathrm{W}(f) df \tag{16}$$

where $f_{\min}$ is the lower limit frequency of the frequency band; $f_{\max}$ is the upper limit frequency of the frequency band; $\overline{W}$ is the averaged acoustic radiation power over this frequency band.

From Eq. (14), it can be seen that the normal vibration velocity $\left\{v_n\right\}$ and the acoustic radiation resistance matrix of the structure's surface $\left[R\right]$ are the two parameters controlling the structural acoustic radiation power. From Eq. (3) and Eq. (14), it can be seen that these two parameters are influenced by the stiffness and surface shape of structure.

III. Topograghy Optimization Analysis

A. Structure Topography Optimization

Usually, the natural frequency of the first mode has the largest contribution to the dynamical characteristics of a plate structure. The first-order mode of vibration is the one of primary interest. Maximizing the natural frequency of the first mode shape will also increase the natural frequency of higher modes and the stiffness of a structure. In this study, the natural frequency of the first mode shape of a folded plate structure is taken as the objective function. By introducing beads or swages to the bracket, the natural frequency of the first mode shape of a folded plate structure is maximized. The optimization software Altair OptiStruct is used to optimize the design of folded plate structures. The shape of folded plate structures is defined by the finite element nodes. The nodes in the design region are taken as the design variables. The finite element mesh is generated by MSC/NASTRAN automatically. The mesh generation parameters are input by the designer. The selection of the nodes to move is automatic. As shown in Fig. 1, the new position of the nodes and elements are determined through the parameters selected by the designer as minimum width (m), draw angle (β), draw height (h) and manufacturing constraints. The process of topography optimization is to move nodes in the direction of the normal vector of an element. The topography optimization had been done iteratively and automatically until the convergence conditions were met.

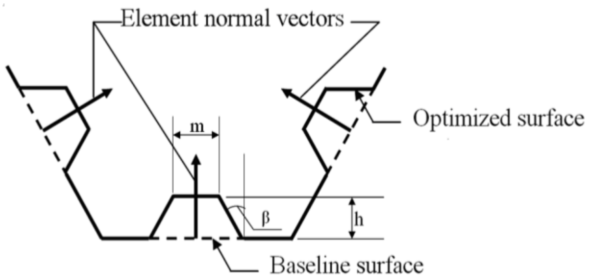

Figure 1. Beads created using the element normal vectors

B. *The Structural-Acoustic Radiation Power Optimization*

In this section, we will investigate the problem of structural acoustic radiation power optimization based on topography optimization. The objective function of optimization is to maximize the natural frequency of the first mode shape. Usually, the bending stiffness of a plate can be increased by maximizing the natural frequency of the first mode. The first-order natural frequency of a clamped rectangle plate can be calculated by using the following formula [22]

$$f_1 = \frac{C_1^2}{2\pi\delta^2}\sqrt{\frac{D}{\rho h}} = \frac{C_1^2}{2\pi\delta^2}\sqrt{\frac{Eh^2}{12\rho\left(1-\nu^2\right)}} \tag{17}$$

where f_1 is the natural frequency associated with the first-order normal mode; C_1 is the factor of the first natural frequency of the plate; δ is the size parameter of the plate; ρ is the mass density of the plate; D is the bending stiffness of the plate; E is the Young's modulus of the plate; h *is* the thickness and ν is the Poisson's ratio of the plate, respectively. It is assumed that the plate has homogenous material properties in all directions.

Note that the normal velocity of an element of a plate structure under the exciting force $\{f(\omega)\}$ can be written as [23]

$$\{v_n\} = \frac{\{f(\omega)\}}{2\pi D(k_n^4 - k_f^4)} \tag{18}$$

where k_n is the normal wave number of the plate; $k_f = \left(\frac{\rho h\omega^2}{D}\right)^{1/4}$ is the free wave number of the plate.

Eq. (17) shows that the first-order natural frequency f_1 is proportional to the bending stiffness D of the plate. Eq. (18) shows that the normal velocity $\{v_n\}$ of an element is inversely proportional to the bending stiffness D of the plate. Thus, when the natural frequency of the first mode is increased, the normal velocity vector $\{v_n\}$ will be reduced. Therefore, we can see from Eq. (14) that the structural acoustic radiation power can be reduced by increasing the natural frequency of the first mode through topography optimization.

Based on the above analysis, the topography optimization method is employed in this paper for the structural acoustic radiation optimization. Topography optimization is a form of generalized shape optimization with automatic shape variable generation. The normal velocity on the surface of structure and the structural acoustic radiation power can be reduced by maximizing the natural frequency of the first mode shape. The goal of maximizing the natural frequency of the first mode shape can be achieved by increasing the bending stiffness of folded plates. In topography optimization, stiffening of a flat plate surface is carried out by the optimum placement of stiffness elements, such as beads, embosses and so on, the shape variation will be under the logical manufacturing constraints.

C. *Optimization Software Altair OptiStruct*

Altair OptiStruct is a linear finite-element based structural optimization software that can be used to design and optimize structures. The objective and constraint functions of Altair OptiStruct are compliance, frequency, volume, mass, displacements, weighted compliance, combined weighted compliance and weighted frequencies of structures etc. For thin-walled structures, beads are often used to reinforce the structures. For a given allowable bead dimensions, OptiStruct's topography optimization technology will generate innovative design proposals with the optimal bead pattern for reinforcement [24].

Topography optimization in software Altair OptiStruct can be considered as a shape optimization method. The process of topography optimization is to move nodes in the direction of the normal vector of an element. By creating protrusions or “corrugations” in the direction of normal vectors of the elements, the moment of inertia becomes larger and the stiffness of the structure is increased.

IV. Numerical Example

A. *A Car Body Structure Model*

A car body structure consists of metal plates. In this car, the frame structure and the form channel structure can be modeled as the folded plate. The acoustic radiation control of folded plates is very important to optimize the noise, vibration, and harshness (NVH) characteristics of a car.

The folded plate is a symmetry structure with crank angle α as shown in Fig. 2. The length of the folded plate is 0.3 m, the width is 0.2 m and the thickness is 0.001 m. The density of this folded plate structure is 7800 kg/m^3 and Poisson's ratio of the folded plate structure is 0.3. The acoustic velocity of the air is 343 m/s and the air density is 1.21 kg/m^3. After optimization, the above structural model is changed from a plate to a shell.

The plate is divided by the four-node quadrilateral shell elements. The finite element model of the folded plate consists of 2400 elements and 2500 nodes. An external harmonic exciting force with the amplitude of 1N is imposed at the node (0, 0, 0) m. The analysis frequency range is 15 to 300 Hz. That is the acoustic frequency band mainly exists inside the car. All edges of the plate are clamped (200 nodes).

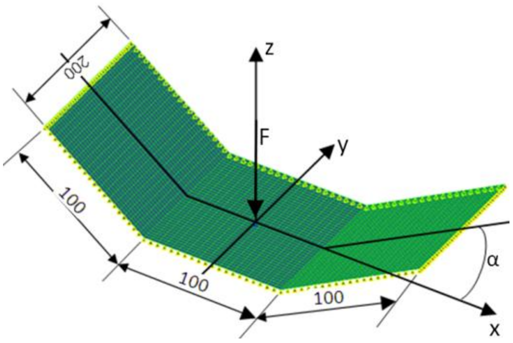

Figure 2. The finite element model of the folded plate structure

B. *Numerical Results*

Frequency response analysis of the folded plate is carried out on MSC/NASTRAN. Code LMS/SYSNOISE is employed to obtain the structural acoustic radiation power on the basis of the results obtained by MSC/NASTRAN. The acoustic radiation power of the folded plate structure with folding angles α = 0^0 and 60^0 at the frequency range from 15 Hz to 300 Hz is plotted in Fig. 3. It can be seen from Fig. 3 that the acoustic radiation power of folded plate structure (α = 60^0) is lower than that of the flat plate structure (α = 0^0). The average value of the structural acoustic radiation power of the folded plate (α = 60^0) is less than that of the flat plate structure (α = 0^0) about 14.9 dB. It shows that the peak value of acoustic radiation has been decreased approximate 48 dB. The comparison of the structural acoustic radiation power of the flat plate (α = 0^0) and folded plate (α = 60^0) demonstrates the superiority of the folded plate structure not only in the structural dynamic characteristics, but also in the reduction of acoustic radiation. The calculated natural frequency of the first mode of the flat plate (α = 0^0) is f_1 = 168.5 Hz. It can be seen from Fig. 3 that the greatest peak value of the acoustic radiation power of the flat plate (α = 0^0) appears at the frequency of 168 Hz and the corresponding acoustic radiation power is W_{max} = 236dB. Therefore, the resonance phenomenon has occurred at the frequency of 168 Hz

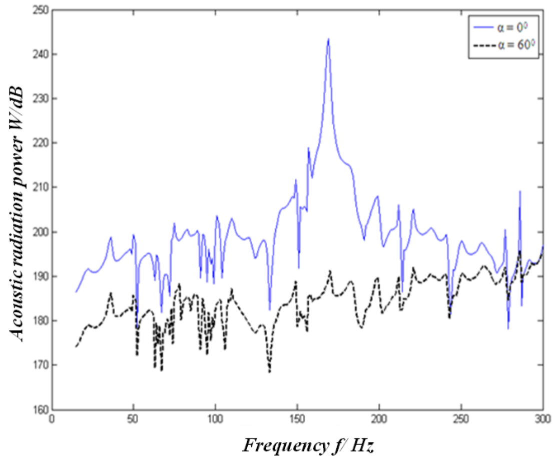

Figure 3. Comparison of the acoustic radiation power between the folded plate and the flat plate

The acoustic radiation power of the proposed model with folding angles α = 30^0, 60^0 and 90^0 are shown in Fig. 4. The analysis frequency range is taken from 15 Hz to 300 Hz.

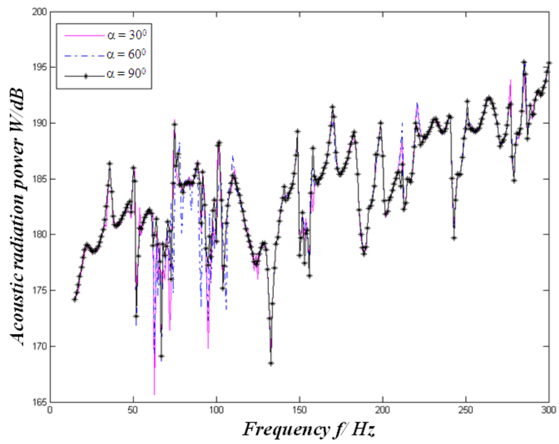

Figure 4. The acoustic radiation power of the folded plate with different α angles

From Fig. 4, we can obtain that when the value of folding angles α is changed, the structural acoustic radiation is almost not changed. It proves that if the theoretical model is not optimized, the variation of folding angle α does not change the structural acoustic radiation power obviously.

Eq. (14) represents that the structural acoustic radiated power depends on the normal velocity vector and the acoustic radiation resistance matrix of structure. Topography optimization method has been applied to the folded plate model, different folding angles α are considered. Each folding angle α represents a model shape. In order to obtain a practical optimal shape, some geometrical and topographical constraints are required in optimization. In this optimization, the geometric parameters to control the shape of beads, namely the drawn height (h), the minimum width (m) and the drawn angle (β) will be specified. The drawn bead height (h) is specified as 5mm according to the material characteristics of steel plate structures. The minimum drawn bead width (m) is 10mm according to the size of the plate structure. The drawn angle (β) is defined as 15° according to the actual drawn angle in forming process. The results after optimization have been shown in Fig. 5.

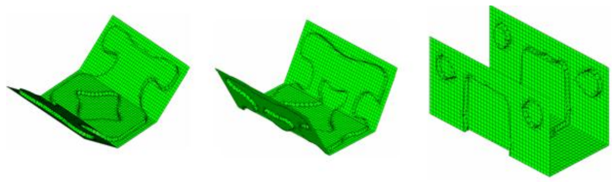

Figure 5. Topography model of the folded plate after optimization with folding angles $\alpha = 30^0$, 60^0 and 90^0

The structural acoustic radiation power depends on the common interface between the structural and the acoustic model. Fig. 6, Fig. 7 and Fig. 8 show that the folded plate models with folding angles $\alpha = 30^o$, 60^o and 90^o have lower acoustic radiation power after optimization compared with the models before optimization. In other words, structural acoustic radiation is reduced by using the topography optimization technique. To study the effect of the folding angles on the structural acoustic radiation power, the structural acoustic radiation powers of the optimized folded plate with folding angles $\alpha = 30^o$, 60^0 and 90^0 are shown in Fig. 9.

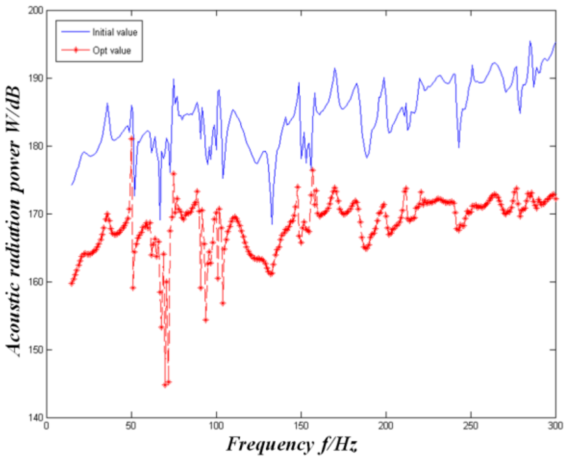

Figure 6. Acoustic radiation power before and after optimization for folding angle $\alpha = 30^0$

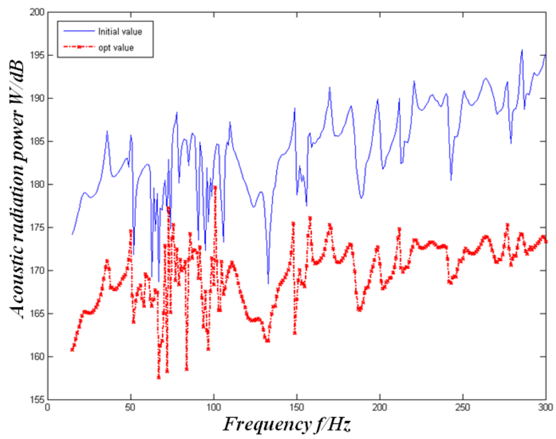

Figure 7. Acoustic radiation power before and after optimization for folding angle $\alpha = 60^0$

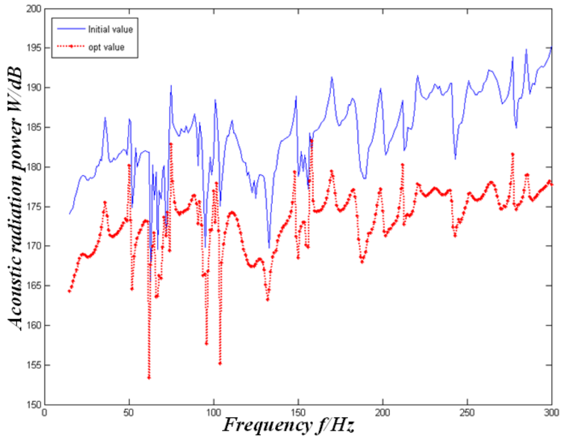

Figure 8. Acoustic radiation power before and after optimization for folding angle $\alpha = 90^0$

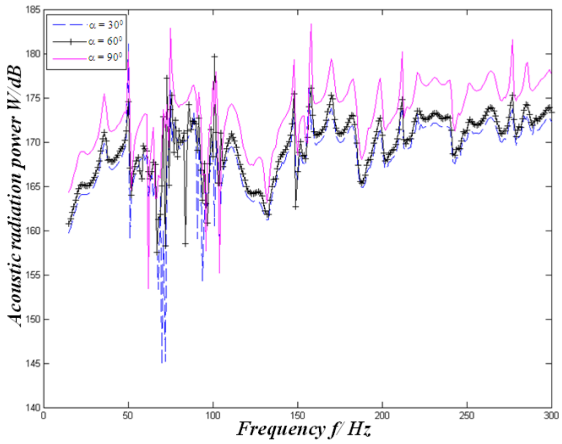

Figure 9. The acoustic radiation power of optimized folded plates under different folding angles α

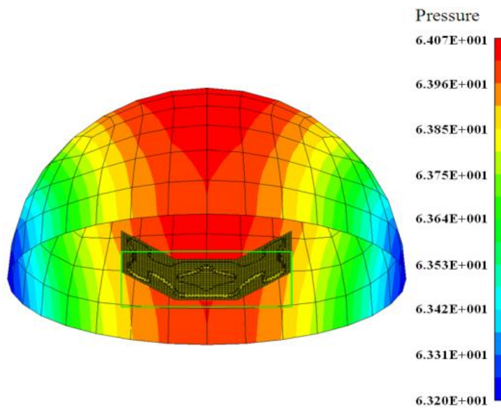

Figure 10. The fringe of the sound pressure of the semi-sphere surface at 154Hz ($\alpha = 30^0$)

It can be seen from Fig. 9 that the structural acoustic radiation power of the optimized plates are different when α has different values. This difference helps the designer to select the optimal folding angle α in the early stage of design. The above results have shown that the structural acoustic radiation power of folded plate structures is decreased by using topography optimization method. In a closed space, the sound pressure may be high even if the structural acoustic radiation power is not high at all. Therefore, both the structural acoustic radiation power and the sound pressure affecting the ears of the people working in the sound environment should be reduced by the topography optimization of folded plate structures. In this study, we define a semi-sphere surface with the radial of 0.32 m around the folded plate structure. The nodal velocity of the folded plate obtained by FEM is transformed into boundary element model. Then, the acoustic radiation pressure of the model can be calculated by boundary element method. Fig. 10 shows the sound pressure of ambient semi-sphere with a folded plate, the folding angle of folded plate is $\alpha = 30^{o}$. The analysis frequency is 154 Hz.

Fig. 11 shows the sound pressure level at a specified node (node 200) on the semi-sphere surface before and after optimization when the folding angle of folded plate is $\alpha = 30^{o}$. It can be seen from Fig. 11 that the average value of the sound pressure level is significantly reduced after optimization, and the comfort of the car driver is improved.

From the discussions above, we can come to a conclusion that both the structural acoustic radiation power and the sound pressure can be reduced by using the topography optimization technique.

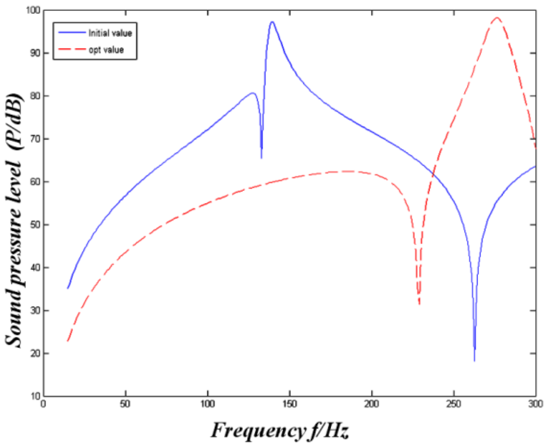

Figure 11. The sound pressure level before and after the optimization ($\alpha = 30^{0}$)

V. CONCLUSION

In this paper, the problem of the structural acoustic radiation power controlling of folded plates by using the topography optimization technique is investigated. The finite element model of folded plates is established and the structural acoustic radiation power analysis is implemented by using CAE software. Altair OptiStruct is used for the topography optimization of folded plates with the objection function of maximizing the frequency of the first mode shape. The numerical analysis results of a folded plate show that the acoustic radiation power and sound pressure level of the folded plate are obviously improved after optimization. When the folding angle of a folded plate is changed, the acoustic radiation power and the sound pressure level of the folded plate will be changed. One can chose the best folding angle according to the acoustic radiation power and the sound pressure level after optimization. The presented approach can be regarded as an effective design tool to control the structural acoustic radiation power of folded plates.

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