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Acoustic design optimization for vibration of cylinder container coupled with interior acoustic field

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ABSTRACT

Cylinder container is a thin cylindrical shell sealed by a top and bottom plate. The internal noise reduction of the cylinder container is very important for vehicle, ship and other projects. A finite element model of the cylinder container vibration coupled with internal acoustic field is set up and the acoustic response feature is determined. Then, two optimization models are adopted: (1) the acoustic response is taken as the object function and the structure weight is taken as constraints; (2) the structure weight is taken as the object function and the acoustic response is taken as constraints. The design variables are thickness distribution for shell, bottom plate and top plate. The optimization problem is solved by the feasible direction method and a 30 dB reduction of the pressure level is achieved for optimum designs. The results show the effectiveness of the method.

1. Introduction

Cylinder container is widely applied in many industries, such as the cab of the automobile, the cabin of the ship and that of the aeroplane. Vibration of this kind of structure is coupled with interior acoustic field and becomes a major noise source of the traffic facility. The structural vibration produces noise and the noise generates a reaction to the structure because of the acoustic pressure. Finite element method is a major method of solving this kind of problem. The acoustic numerical value method of the coupled structure-acoustic system based on FEM is a foundation of the design of the noise reduction. It can forecast the vibration and noise at an early design stage and be widely used in the noise reduction research of car and vehicle [1–3]. The traditional design of noise reduction requires time-consuming and tedious processes of successive analyses involving sequential design modifications until the desired criteria are satisfied. However, when the design sensitivity and optimization is used in design process, engineers can design the best suitable structure for the noise reduction without both tedious and time-consuming sequential design modifications. The size optimization design of the structure-acoustic system is to minimize the acoustic pressure or acoustic power by modification structure parameter. Three steps must be done in this optimization process: the first one is to carry out the acoustic response, the second is to derive the acoustic sensitivity with respect to the structure parameter, and the last step is to conduct the acoustic optimization.

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Several papers that deal with structural-acoustic optimization indicate significant improvements of the acoustic properties inside or outside the structure. Noise reductions of 10–50 dB have been reported. In optimization process, the sensitivity calculations of acoustic pressure or acoustic power with respect to the structure parameters are necessary when classical basing-gradient methods are used. Ma and Hagiwara [4,5] performed the sensitivity analysis of both modal and response acoustic pressure of coupled acoustic-structure system with respect to the structure parameters using modal method and the direct method, and the pressure sensitivities had been successfully used in the design of reducing noise inside the vehicle. Hambric [6,7] reported the reduction of the emitted sound power of a ribbed cylindrical shell with hemispherical ends over a large frequency range by about 10 dB using location of ribs, shell thickness and loss factors as parameters. Similarly, Lamancusa [8,9] indicated 8–15 dB reduction in a frequency range of 100–1000 Hz only by optimizing the local distribution of plate thickness at a constant mass. Pal and Hagiwara [10] performed a optimization of the structure response and the sound pressure within a box with the structure synthesis method and the pseudo-inverse method and a reduction of about 50 dB is achieved. S. Marburg and H.-J. Beer et al. [11] researched a steel box of with an external beam structure welded on three surface plates, their investigation included experimental modal analysis and experimental measurements of certain noise transfer functions (sound pressure at points inside the box due to force excitation at beam structure), and the results showed validity of the acoustic optimization. Jianghui Luo and Hae Chang Gea [12] used modal method to evaluate the interior sound level for coupled structural-acoustic system, then performed a topology optimization of stiffener with Microstructure-based Design Domain Method (MDDM) and a pressure level reduction of about 20 dB is achieved. Erik Bangtsson et al. [13] optimized the shape of an acoustic horn to provide impedance matching to the surrounding air using a BFGS quasi-Newton algorithm and the resulting horn has good impedance properties throughout the entire frequency band of interest. Renato Barbieri and Nilson Barbieri [14] presented a shape optimization of muffler design with the transmission loss being maximized in the frequency range of interest. Eddie Wadbro and Martin Berggren [15] presented a method for topology optimization of an acoustic horn with the aim of radiating sound as efficiently as possible, the Helmholtz equation modeling the wave propagation is solved using the finite element method. In the above mentioned references, the optimization objects are all the acoustic parameters such as acoustic pressure, acoustic power and acoustic feature coefficient, etc, the structure weight have not been taken as the object.

In general, the weight reduction will induces a noise addition, reversely, a weight addition will result to the noise reduction, the acoustic characteristic has a significant relation with the weight of structure. In this paper, the acoustic response and the acoustic sensitivity have been carried out by the use of the direct FEM, the acoustic optimization have been performed by the feasible direction method. The two optimization model have been set up: one is that the response acoustic pressure of nodes is taken as object function and the structure weight is taken as the restriction, the other one is that the structure weight is taken as object function and the response acoustic pressure of nodes is taken as restriction. The acoustic optimization of the cylinder container have been performed as an example. The example results show that the design optimization of cylinder container is an effective methods for simultaneously reducing response acoustic pressure level and structure weight.

2. The acoustic response and its sensitivity of the coupled structure-acoustic system

The radiated steady-state acoustic pressure P of structure when acted a harmonic load satisfy Helmholtz differential equation

$$\overline{\nabla}^2 P + k^2 P = 0 \tag{1}$$

where P is the acoustic pressure amplitude, $k=\omega/c$ is the wavenumber, c is the speed of sound in the medium, ω is the frequency;

$$\nabla^2 = \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2}$$

is the Laplacian operator. The acoustic pressure amplitude P is independent of time, but relates directly to the structure characteristic, acoustic field position and ω.

The Eq. (1) indicates the distribution of the acoustic pressure P in space. The acoustic pressure P can be solved if the boundary condition is knew well. If the vibration boundary condition of a structure is a small vibration amplitude on the flexibility surface, it can be expressed as follows

$$\frac{\partial P}{\partial n} = n\rho\omega^2 U_S = -jn\rho\omega V_S \tag{2}$$

where n is the unit normal vector on the structure surface; U_S, V_S is the normal displacement amplitude and normal velocity amplitude of the structure vibration, respectively, ρ is the density of the acoustic medium, and $j = \sqrt{-1}$.

The Eq. (1) having the boundary condition described by Eq. (2) can be solved using the numerical method such as FEM (finite element method) and BEM (boundary element method), the FEM is more applied to the interior acoustic field enveloped by the thin shell structure.

When the FEM is used to solve the acoustic problem radiated by a structure vibration, The acoustic field space described by Eq. (1) with the boundary condition of Eq. (2) is partitioned into the finite elements while the structure should also be partitioned into the finite elements, the acoustic pressure vector of acoustic media nodes is expressed as $\{\mathbf{P}\}$, the vibration displacement vector of the structure nodes is expressed as $\{\mathbf{U_s}\}$. Combining the Eq. (1) with the Eq. (2) and using the weighted residual formulation and the Green's function, the FEM equation used to solve the acoustic pressure $\{P\}$ can be formulated as follows

$$([\mathbf{K_a}] - \omega^2[\mathbf{M_a}])\{\mathbf{P}\} = \omega^2[\mathbf{S}]\{\mathbf{U_s}\} \tag{3}$$

where $[\mathbf{K_a}]$, $[\mathbf{M_a}]$ and $[\mathbf{S}]$ are, respectively, stiffness, mass of acoustic field and coupled matrices between the acoustic and the structure, which are directly related to the property of acoustic media, the geometry shape of the acoustic field, the stimulated frequency and the coupling characteristic. If only the acoustic pressure $\{\mathbf{P}\}$ is required to solve, then $\{\mathbf{U}_s\}$ should be knew well. If the reaction on the structure loading by the acoustic pressure $\{\mathbf{P}\}$ can not be neglected, then this kind of problem is regarded as the problem of the coupled acoustic-structure. In the problem of the coupled acoustic-structure, the integrative FEM equation of the acoustic field and the structure is common established, then $\{\mathbf{P}\}$ and $\{\mathbf{U_s}\}$ can be solved simultaneously.

If the reaction force on the structure loading by the acoustic pressure $\{P\}$ is considered, then the vibration FEM equation of the non-damping structure can be expressed as follows:

$$([\mathbf{K_S}] - \omega^2[\mathbf{M_S}])\{\mathbf{U_s}\} = \{\mathbf{F_a}\} + \{\mathbf{F_S}\} \tag{4}$$

where $\{\mathbf{F_a}\} = [\mathbf{S}]^\mathrm{T}\{\mathbf{P}\}$ is the reaction force on the structure loading by the acoustic pressure $\{\mathbf{P}\}$, $[\mathbf{K_S}]$, $[\mathbf{M_S}]$ are the stiffness and mass matrices of the structure, respectively, $\{\mathbf{F_S}\}$ is the outer stimulating load acting on the structure.

By a combination of Eq. (3) and (4), the coupled integrative FEM equation can be formulated as follows

$$[\mathbf{Z}]\{\mathbf{U}\} = \{\mathbf{F}\} \tag{5}$$

$$[\mathbf{Z}] = \begin{bmatrix} [\mathbf{K_S}] - \omega^2[\mathbf{M_S}] & -[\mathbf{S}]^\mathrm{T} \\ \omega^2[\mathbf{S}] & [\mathbf{K_a}] - \omega^2[\mathbf{M_a}] \end{bmatrix} \tag{6}$$

where $[\mathbf{Z}]$ is called as impedance matrix, $\{\mathbf{U}\} = \{\{\mathbf{U_s}\}\ \{\mathbf{P}\}\}^\mathrm{T}$ is unknown vector including the vibration displacement amplitude vector of structure node and response acoustic pressure amplitude vector of fluid node, $\{\mathbf{F}\} = \{\{\mathbf{F_S}\}\ 0\}^\mathrm{T}$ is the outer load vector. If outer load $\{\mathbf{F}\}$ acting on the structure is knew well, then $\{\mathbf{U}\}$ can be solved from the Eq. (5).

The dynamics response of coupled structure-acoustic system can be obtained from Eq. (5) using the direct method

$$\{\mathbf{U}\} = [\mathbf{Z}]^{-1}\{\mathbf{F}\} \tag{7}$$

We denote the design variable as $\mathbf{X} = \{x_1, x_2, \ldots, x_n\}^\mathrm{T}$, differentiate Eq. (5) with respect to x_j $(j = 1, 2, \ldots, n)$, we can have the dynamics response sensitivity $\{\mathbf{U}\}'$ with respect to x_j, which include the structure displacement sensitivity $\{\mathbf{U_s}\}'$ and the acoustic pressure sensitivity $\{\mathbf{P}\}'$, this sensitivity indicate the direction of the structure modification for the design optimization, and it is major part of the reduction noise design. The outer load $\{\mathbf{F}\}$ is not related to the design variables, then differentiating Eq. (5) with respect to design variable $\mathbf{X}$, we have

$$[\mathbf{Z}]'\{\mathbf{U}\} + [\mathbf{Z}]\{\mathbf{U}\}' = 0 \tag{8}$$

from Eq. (8) we can get the sensitivity $\{\mathbf{U}\}'$ as follows

$$[\mathbf{U}]' = -[\mathbf{Z}]^{-1}[\mathbf{Z}]'[\mathbf{U}] \tag{9}$$

where $[\mathbf{Z}]'$ is the sensitivity of the impedance matrix $[\mathbf{Z}]$.

According to Eq. (7) the acoustic pressure $\{\mathbf{P}\}$ can be carried out, then the pressure level can be expressed as follows:

$$\{\mathbf{L}\} = 20\log_{10}(\{\mathbf{p}\}/U_0) \tag{10}$$

where U_0=2×10^{-5} pa is the reference acoustic pressure.

From Eq. (10) the sensitivity of the pressure level can be formulated as follows:

$$\{\mathbf{L}\} = 20\{\mathbf{p}\}'(\{\mathbf{p}\}\ln 10) \tag{11}$$

3. The optimization design model

The structure-acoustic optimization depends on both the dynamics response and the sensitivity of the coupled structure-acoustic system, the shell thickness is taken as the design variable in this research, the optimization model is formulated as follows

(1) Minimizing the acoustic response
The object function

$$\min y = [\sum_{k=1}^{l}\sum_{j=1}^{m} p_{k,j}^2(\mathbf{x},\omega_j)]^{1/2}, j = 1,2,\ldots,m \tag{12}$$

The constraining condition

$$\begin{cases} G_1 < G(\mathbf{X}) < G_2 \\ x_{id} < x_i < x_{iu}, i = 1,2,\ldots n \end{cases} \tag{13}$$

(2) Minimizing the structure weight
The object function

$$\min G = \sum_{i=1}^{n} x_i A_i \rho, i = 1,2,\ldots,n \tag{14}$$

The constraining condition

$$\begin{cases} p_{k,j}(\mathbf{X},\omega_j) \le p_u, & j = 1,2,\ldots,m \\ x_{id} < x_i < x_{iu}, & i = 1,2,\ldots n \end{cases} \tag{15}$$

In Eqs. (12)–(15), the design variable $\mathbf{X}$ is the local shell thickness; G, G_1 and G_2 is the structure weight, the lower bounds and the upper bounds of the structure weight, respectively; x_{id}, x_{iu} is the lower bounds and upper bounds of the design variable x_i, respectively; m is the number of the interest frequency considered; n is the number of the design variables; l is the number of the considered interest fluid nodes; ω_j is the jth simulating frequency; $p_{k,j}$ is the response acoustic pressure of the node k in the frequency ω_j.

The above optimization models belong to the constrained optimum problems, the feasible direction method is commonly used and is an effective and main method for solving this kind of problems. The fundamental principle is that a feasible preliminary point $\mathbf{X}^0$ is selected firstly, then a feasible direction $\mathbf{d}$ is established by the sensitivity information of the object function with respect to the design variables and a step length α is figured out with one dimension search method, and the iterative formulation can be written as follows

$$\mathbf{X}^{k+1} = \mathbf{X}^k + \alpha\mathbf{d}(k = 1,2,\ldots,) \tag{16}$$

According to Eq. (16), we will achieve an optimum design variable $\mathbf{X}$ that it make the object function have a least value. In optimization process, the sensitivities of both the object functions and the constrain conditions with respect to the design variables are required, this sensitivities requires that the acoustic pressure sensitivities with respect to the design variables are calculated.

4. Numerical results and the validation of the method

4.1. The FEM model and its acoustic pressure response

The FEM model of the cylinder container coupled with interior acoustic field is shown in Fig. 1 as an example for validating the mentioned method. The cylinder height is 1.5 m and the diameter is 1 m, the interior is filled with air, the nodes on circumference of lower circle plane are hinged, an excitation force with amplitude 50 N and the frequency range from 1 to 200 Hz is applied to center point on upper circle plane, which pointed to the $-z$ direction. The cylinder shell is meshed into 394 elements with both 294 quadratic shell elements, 100 triangle shell elements and 496 nodes, the interior acoustic field is meshed into 600 hexahedron solid elements and 200 wedge-shaped solid elements. The shell material property is follows: the elasticity modulus E=2×10^{11} Pa/m^2, Poisson ratio μ=0.3, the mass density ρ_s=7800 kg/m^3; the acoustic feature of air is follows: the bulk modulus B=2.1×10^5 Pa/m^3, the density ρ_a=1.25 kg/m^3, the acoustic speed C=340 m/s.

The node number on the axis of the cylinder acoustic field are shown in Fig. 2, the numbers in the bracket are the coordinate parameters. The acoustic response of the 8 nodes on the center line of the cylinder acoustic field are shown in Fig. 3, the node 8 is the upper end and the node 1 is the lower end, the 8 nodes are distributed uniformly on the center line of the cylinder acoustic field from node 1 to node 8. Fig. 3 shows that the acoustic pressure peak are produced at the frequency of 45, 135 and 175 Hz, the maximal pressure value is produced at node 1 at 175 Hz, the value is 251 Pa.

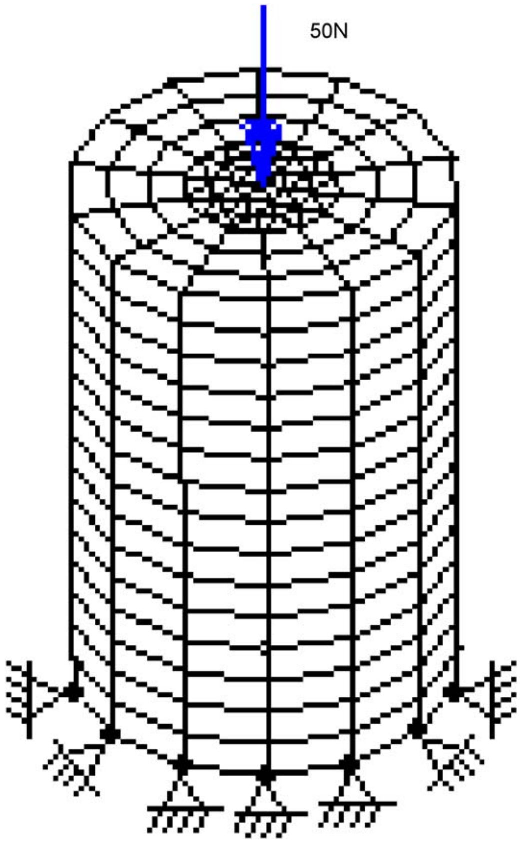

Fig. 1. The FEM model of the cylinder coupled with the interior acoustic field.

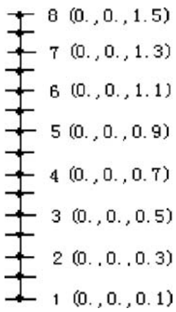

Fig. 2. The interesting optimization design nodes.

The pressure distribution contour of the cylinder acoustic field at frequency of the peak pressure value are shown in Figs. 4–6, the Figs. 4 and 5 show that the acoustic pressure in cylinder acoustic field only change along the axis direction at both 45 and 135 Hz. Fig. 6 shows that the acoustic pressure in cylinder acoustic field change not only along the axis direction but also along the radial direction at 175 Hz, the change along the radial direction can be found on the down end plane of acoustic field, it shows that its central region pressure is maximal, the pressure value reaches to the 424.8 Pa.

4.2. The acoustic pressure sensitivity

The shell of cylinder container is partitioned into 13 parts, the thickness of each part is taken as design variables, and it is shown in Fig. 7. The upper circle plane and the lower circle plane are partitioned into 4 parts, respectively. Both part 1 and part 13 are a circle plane with the diameter 0.4 m on the lower and upper circle plane, and their thicknesses is denoted as the design variable 1 and 13, respectively; the part 2 and 12, 3 and 11, 4 and 10 are the circle ring with the outer diameter

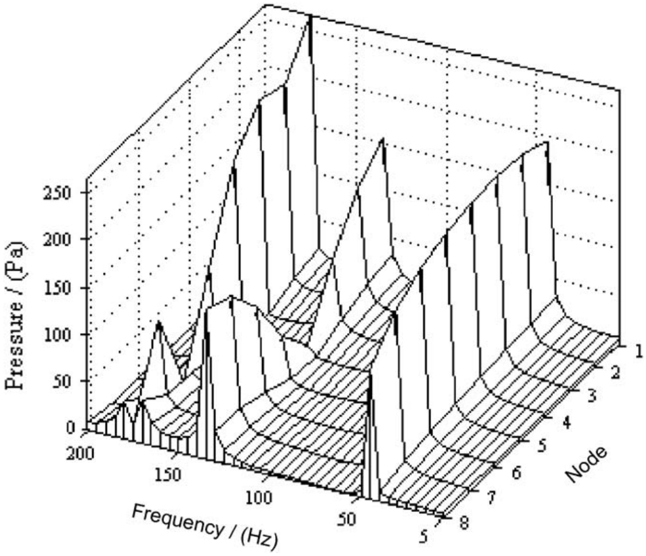

Fig. 3. The response acoustic pressure before optimization.

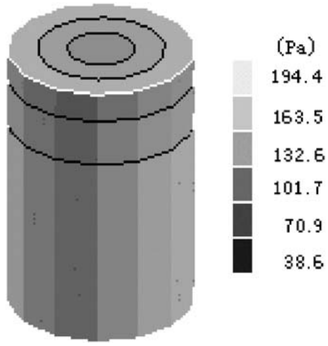

Fig. 4. The pressure contour at 45 Hz before optimization.

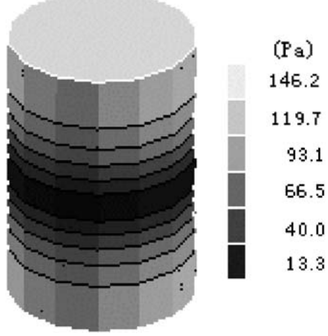

Fig. 5. The pressure contour at 135 Hz before optimization.

0.6, 0.8, and 1.0 m, respectively, and their thicknesses are denoted as the design variable 2 and 12, 3 and 11, 4 and 10, respectively; the cylinder surface is partitioned into 5 parts with the uniform height 0.3 m, which are noted as 5, 6, 7, 8 and 9, and their thicknesses are denoted as the design variable 5, 6, 7, 8 and 9. The acoustic pressure level (SPL) sensitivity of node 1–8 in Fig. 2 at 100 Hz for all design variables are shown in Fig. 8. The pressure level sensitivity of node 4 (1612) at each frequency are shown in Fig. 9. Figs. 8 and 9 show that the frequencies of the peak of SPL sensitivities are same as the frequencies of the peak of the response acoustic pressures.

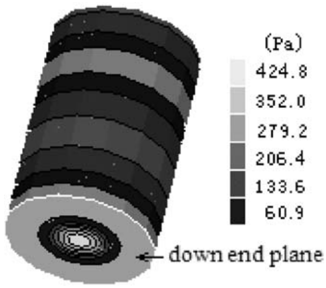

Fig. 6. The pressure contour at 175 Hz before optimization.

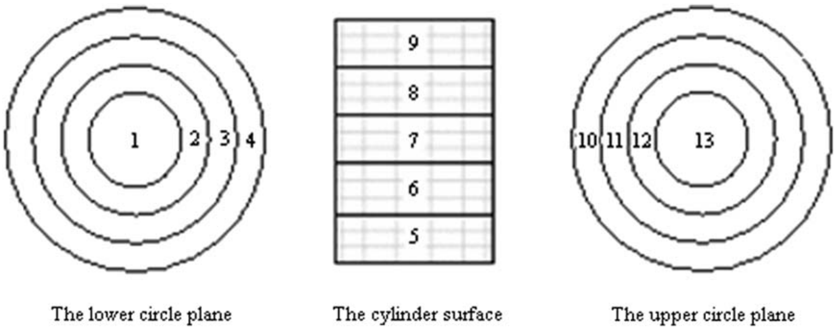

Fig. 7. The design variable distribution.

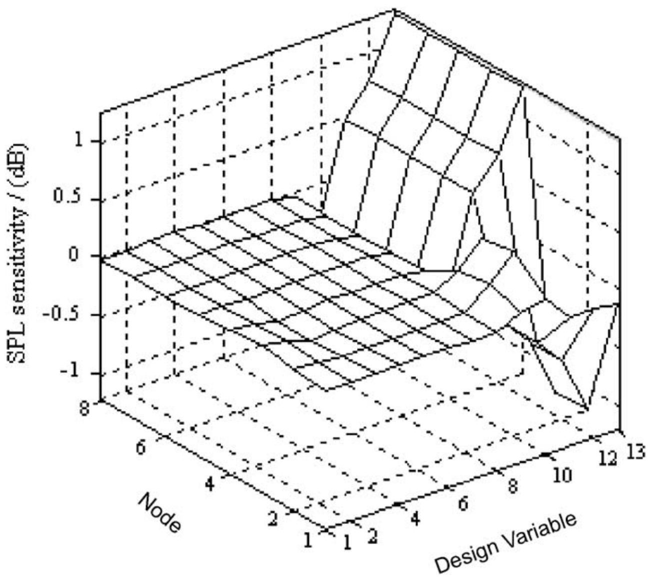

Fig. 8. The pressure level sensitivity of all nodes at 100 Hz.

4.3. *The structure-acoustic optimization design*

4.3.1. *Minimizing acoustic pressure with the restriction of weight*

The optimization design of minimizing acoustic pressure is performed according to the optimization model (1) formulated by Eqs. (12) and (13). The acoustic fluid nodes 1–8 in Fig. 2 are chosen as the design nodes; the design variables are allowed to change from 3 to 7 mm; the structure weight is allowed to change from 240.92 to 242.92 kg, the initial weight is 241.92 kg, which indicates that the structure weight is relatively non-changed in the optimization process. The optimization frequencies are a series of values with an interval of 5 Hz starting from 5 Hz in 5–200 Hz. The actual

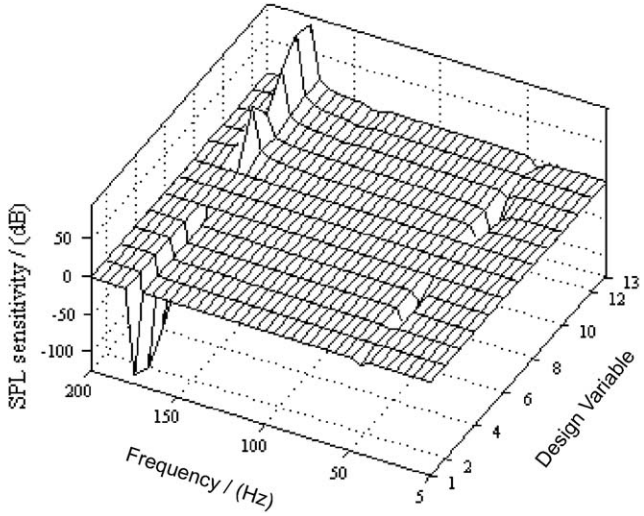

Fig. 9. The pressure level sensitivity of node 4 (1612) at each frequency.

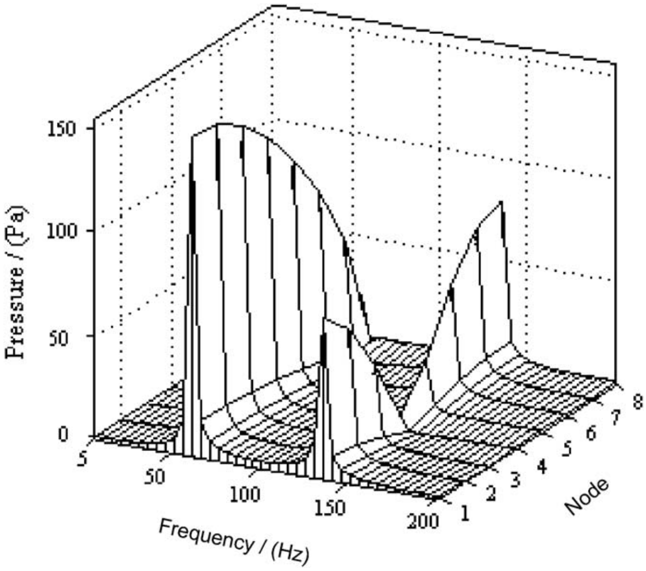

Fig. 10. The optimized acoustic pressure by minimizing acoustic pressure.

Table 1
The optimized design variable values by minimizing acoustic pressure.

Design variable	1	2	3	4	
Variable value (mm)	7.0	7.0	3.0	7.0	
Design variable	5	6	7	8	9
Variable value (mm)	7.0	3.0	3.0	3.0	7.0
Design variable	10	11	12	13	
Variable value (mm)	7.0	7.0	7.0	7.0	

optimization model is follows

$$\begin{cases} \min y = [\sum_{k=1}^{8}\sum_{j=1}^{20} p_{k,j}^2(\mathbf{X},\omega_j)]^{1/2}, \qquad \omega_j = 5\,\text{Hz}, 10\,\text{Hz}, \ldots, 200\,\text{Hz}, \;\; k = 1, 2, \ldots, 8 \\ 240.92\,\text{kg} < G(\mathbf{X}) < 242.92\,\text{kg} \\ 3\,\text{mm} < x_i < 7\,\text{mm}, \;\; i = 1, 2, \ldots 13 \end{cases} \tag{17}$$

The optimized acoustic pressures of nodes 1–8 are shown in Fig. 10. Obvious reductions of the response acoustic pressures can be seen in Fig. 10 compared with the Fig. 3, and the peak acoustic pressure at 175 Hz has been reduced to a little value; the peak of acoustic pressure at 135 Hz has been also reduced from 150 to 80 Pa, and the peak of acoustic pressure at 45 Hz has been reduced from 200 to 151 Pa at 60 Hz, this may be thought that the peak frequency is shifted from 45 to 60 Hz and the pressure peak is decreased. The optimized design variable values are shown in Table 1, and the optimized design variable values are all gotten the maximal value 7 mm or the least value 3 mm. The optimized structure weight is 242.6 kg.

The acoustic pressure will get more reduction if the structure weight is allowed to add ΔG kg in optimization process, then the structure weight will achieve a maximal value $\Delta G+G$ kg for reducing the noise after optimization because of optimization effect, and the optimization program will search for a optimum addition weight location. If ΔG is equal to 20 kg, then the upper restriction bounds of structure weight is $G \leq 241.92+20$ kg (241.92 is the original weight), which is that upper bound of the weight restriction is 261.92 kg. According to the optimization model (1), the optimization has been performed again, the acoustic pressures of design nodes have been gotten the more reductions, the optimized response acoustic pressures are shown in Fig. 11, the optimized design variable values are listed in Table 2. The Table 2 indicates that the thickness of down circle plane of cylinder container not changed, the thickness of middle part of cylinder surface (design variable 6, 7 and 8) is reduced to 4.7 mm while the thickness of upper part of cylinder surface (design variable 9) is increased to 7 mm, and the thickness of upper circle plane is all increased to 7 mm that is the upper bounds value of design variables. These clarify that an added weight of 20 kg is appended to the upside of the close cylinder container shell. Fig. 11 indicates that the response acoustic pressures are farther decreased from 151 to 120 Pa at 45 Hz when compared with Fig. 10, but it can be found that the acoustic pressure at 135 Hz has been rather increased from 80 to 110 Pa, this fact shows that an addition of 20 kg weight is chiefly used to reduce the lower frequency noise.

4.3.2. Minimizing the structure weight with the restriction of acoustic pressure

According to the optimization model (2) formulated with Eqs. (14) and (15), the optimization of minimizing weight has been achieved with the restriction of the acoustic pressures at the nodes of the cylinder container center, the restriction

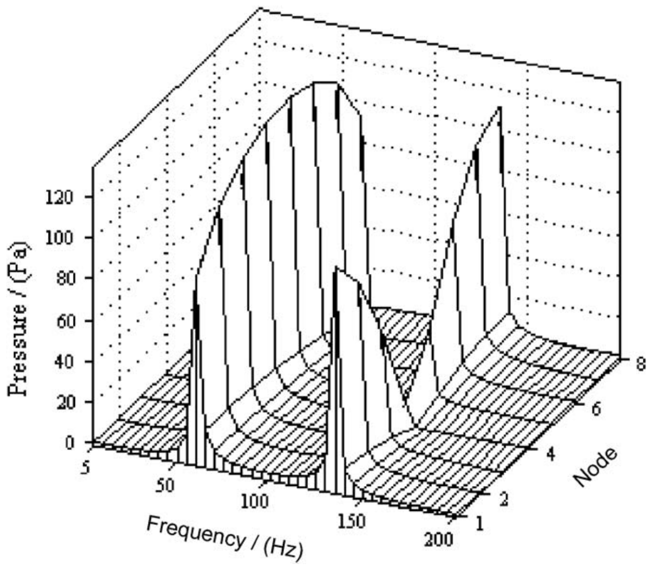

Fig. 11. The optimized acoustic pressure by minimizing acoustic pressure when having a added weight of 20 kg.

Table 2
The optimized design variable values when having a added weight of 20 kg.

Design variable	1	2	3	4	
Variable value (mm)	5.0	5.0	5.0	5.0	
Design variable	5	6	7	8	9
Variable value (mm)	5.0	4.7	4.7	4.7	7.0
Design variable	10	11	12	13	
Variable value (mm)	7.0	7.0	7.0	7.0	

equation Eq. (15) is actually formulated as follows

$$\begin{cases} p_{k,j}(\mathbf{X}, \omega_j) \le 40\,\text{Pa}, & j = 1, 2, \ldots, 40 \\ & k = 1, 2, \ldots, 8 \\ & \omega_j = 5 * j \\ 0.003\,\text{m} < x_i < 0.007\,\text{m}, & i = 1, 2, \ldots, 13 \end{cases} \tag{18}$$

Eq. (18) indicates that the node number of the acoustic pressure restrictions is 8, the stimulating frequency number is 40, they are 5, 10, 15,…,195, and 200 Hz, the restrictions of acoustic pressures are 40 Pa, the design variable number is 13, the upper bounds and the lower bounds of design variables is 0.003 and 0.007 m, respectively.

The optimized design variable values are listed in Table 3, it can be found that the thickness of the lower circle plane is about all decreased except the design variable 2, the thickness of the upper circle plane is about all increased except the design variable 10 and the thickness of the cylinder surface is gradually increased from lower end to upper end. The optimized structure weight is 240.47 kg, it is that a reduction of 1.45 kg is achieved compared with the original structure weight.

The optimized acoustic pressures are shown in Fig. 12, it can be found that the all peaks of acoustic pressures are small than 40 Pa, which is the restriction value of acoustic pressure. Fig. 12 is compared with Fig. 3, it can be found that the original peaks of acoustic pressures at 175 Hz have been decreased to about 10 Pa from original 251 Pa, a 30 dB reduction of the pressure level is achieved for optimum designs, the optimized design has gotten a remarkable effect on reducing noise, and the optimized structure weight has also gotten a reduction of 1.45 kg than the original structure weight, but optimized acoustic field has been more complexity, the 4 pressure peaks appear in the Fig. 12 compared with the Fig. 11 that have 2 pressure peaks. Fig. 12 is compared with the Fig. 10, we can get the conclusion as follows: the weight optimization with acoustic pressure restriction has a better effect on reducing noise than the acoustic pressure optimization with weight restriction. Whether the weight optimization or the acoustic pressure optimization, the acoustic pressure reductions are more great within higher frequency range.

The value of acoustic pressure used as restriction should be chosen in reason basing on both the response acoustic pressure values and the demand of noise reduction, the optimization design may not be gotten if the restriction acoustic

Table 3
The optimized design variable values by minimizing structure weight.

Design variable	1	2	3	4	
Variable value (mm)	3.3	6.0	3.6	3.0	
Design variable	5	6	7	8	9
Variable value (mm)	4.0	5.2	5.2	5.2	5.8
Design variable	10	11	12	13	
Variable value (mm)	3.4	6.9	6.9	7.0	

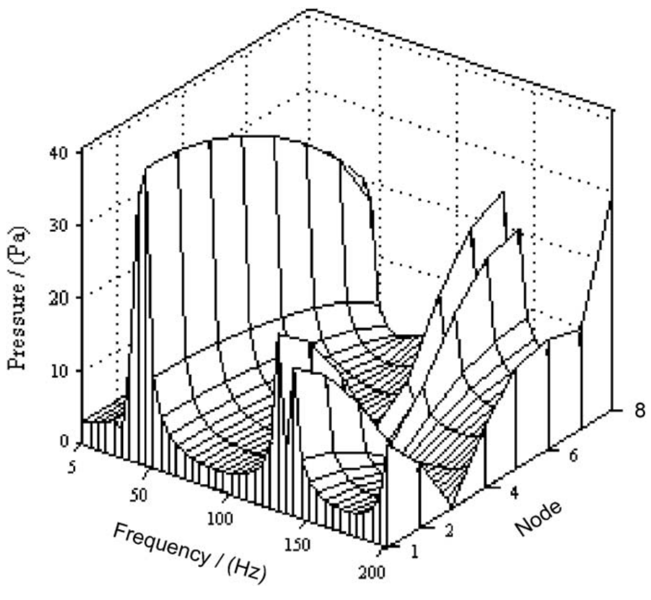

Fig. 12. The optimized acoustic pressure by minimizing structure weight.

pressure is too lower. The more higher the restriction acoustic pressure is, the more smaller the optimized structure weight is; if the restriction acoustic pressure is 160 Pa, then the optimized structure weight is decreased to 234.8 kg while the maximal value of the acoustic pressure of optimization design is within 160 Pa.

5. Conclusion

The optimization designs of cylinder container structure for reducing interior noise are achieved basing on the finite element model of coupled acoustic-structure system in this paper. The two optimization models are set up. One is that the sum of the square of the response acoustic pressures at the design nodes is taken as the objective function with the restriction of the structure weight, this optimization can decrease the acoustic pressure by redistributing of shell thickness while the structure weight is almost unchanged; the other one is that the weight of cylinder container structure is taken as the objective function with the restriction of the acoustic pressure at the design nodes, this optimization can simultaneously decrease the response acoustic pressures and the structure weight if the restriction acoustic pressure is chosen in reason. The weight distribution of optimized cylinder container shell is not uniform, generally, the shell thickness is gradually become to large from down end to upper end. The optimization of minimizing weight with the restriction of the acoustic pressure has been achieved a better effect on the reducing noise than the optimization of minimizing acoustic pressure, and it can simultaneously reduce the structure weight. Whether or not, the acoustic pressure reductions are more great within higher frequency. The optimization model of weight minimum with the restriction of acoustic pressure has achieved a more better effect than the other. As a result of optimization, the reason of noise reduction is that the coupled modal frequency is changed, the coupled modal frequency is shifted to apart from the stimulating frequency, because of the acoustic field shape is not changed, so we also think that the structure modal frequency is shifted to apart from the stimulating frequency and results the noise reduction.

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