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Multidisciplinary Design Optimization of Sound Radiation from Underwater Double Cylindrical Shell Structure

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1. Abstract
Underwater acoustic radiation analysis for the shell structure mainly yields to the coupling of structural vibration analysis and acoustic analysis. For example, a weak coupling relay analysis is usually performed by combining an ANSYS software-based structural vibration analysis and a SYSNOISE software-based acoustic analysis, and a strong coupling analysis for vibration and acoustic is normally based on VAONE software, which is an integrated analysis tool. However, it is much easier to use ABAQUS software for the integrated analysis of structural vibration and acoustic. On the other hand, constructing surrogate models for the acoustic analysis is the best way to simplify the acoustic analysis procedures and formula accuracy and efficient structural acoustic design optimization model, where an explicit analytical formulation of the acoustic radiation characteristics regards to the structural sizes is provided. Therefore, this paper utilizes the Latin Hypercube Sampling method to select sampling points, and considers three types of surrogate models, polynomial response surface approximation, Kriging, and radial basis neural network, to approximate the acoustic radiation of a double cylindrical shell structure. Through comparison of the approximation accuracy of three types of surrogate models, the appropriate surrogate models are chosen to construct the optimization model, and the optimization model is solved by using Matlab optimization toolbox. This research provides references for predicting the acoustic radiation of underwater structures and performing acoustic design.
2. Keywords: underwater cylindrical shell, acoustic radiation, acoustic design, surrogate model, structural optimization

3. Introduction
The probing for underwater structures is mostly about the water sound. How to reduce the sound radiation of underwater structures becomes the key point of improving the invisibility of underwater objectives. It is also significant for the acoustic optimization design of the naval structures. Reducing the underwater sound radiation of naval structures cannot only improve the invisibility, but also increase the reaction distance of their sound navigation and raging (SONAR) system. The sound-solid interaction characteristic of underwater structures is currently the main consideration regarding to the coupling of structural vibration and sound radiation. Because of the complexity of underwater structures, analysing approaches for structural vibration and sound radiation include analytical method and numerical method. The numerical method can be used to solve sound radiation problems of relative complex structures and structures with complex boundary conditions. However, it is impossible to obtain the characteristics of structural vibration and sound radiation for complex structures by using the analytical method. Transfer matrix method (TMM), finite element method (FEM), boundary element method (BEM), FEM combined with BEM (FEM/BEM), and FEM combined with infinite element method et al. [1] are numerical methods often used. With the developments of computing techniques and large calculation softwares, FEM/BEM is the most popular method. For example, a weak coupling relay analysis is usually performed by combining an ANSYS software-based structural vibration analysis and a SYSNOISE or Virtual. Lab software-based acoustic analysis, and a strong coupling analysis for vibration and acoustic is normally based on VAONE software, which is an integrated analysis based on FEM/BEM.

The sound radiation characteristic of an underwater double cylindrical shell structure is studied in this paper. Although the double cylindrical shell structure is used widely in naval engineering, such as submarine, oil platform, the sound radiation caused by the internal vibration is hardly predicted precisely, and normally underestimated in the process of noise control. In order to simplify the computing procedure and improving calculating efficiency, this paper utilizes a large commercial software ABAQUS for the coupling analysis of structural vibration and acoustic. The acoustic medium is adopted to describe the fluid. The boundary impedance technology is used to simulate sound wave spreading in the infinite water and to provide underwater sound radiation results of the double cylindrical shell structure.

The factors that infect the structural sound radiation of double cylindrical shell structure are complicated, such as structural shaping, inner and outer shell thicknesses, and number of ribs et al. [2]. Calculation for the relationship between structural design parameters and sound radiation is complicated and time-consuming. If this numerical calculation is performed directly in optimization design, it is impossible to be carried out smoothly due to numerous calculations. An effective solution for this issue is to use surrogate models [3]. The thicknesses of inner and outer shells of the double cylindrical shell structure significantly effect on structural vibration models and underwater radiation noise. In this paper, the effects of the shell thicknesses on the structural sound radiation are studied by utilizing surrogate modeling method. Three surrogate models, Polynomial Response Surface approximation (PRS), Kriging, and Radial Basis Neural Network (RBNN), are employed to approximate the function expression of the sound pressure level in near field regarding to the inner and outer shell thicknesses. The optimization model of minimizing structural mass with the near field sound pressure level constraint is constructed at the end. The "fmincon" function in MATLAB is utilized to solve the optimization model, and the optimal shell thicknesses are obtained.

4. ABAQUS-based integrated analysis for acoustic-solid interaction of double cylindrical shell structure

4.1. Finite element model of double cylindrical shell structure

The double cylindrical shell structure is 10m long along the axial direction. It has ribs between the inner and outer shells, and the distance between two ribs is 1m. The ribs are round with constant sections, height of 0.06m, and thickness of 0.01m. The diameters of the inner and outer shells are, respectively, 1.0m and 1.12m. The excitation force is 50N and applied at the centre point of the outer shell along the circumferential direction. As shown in Figure 1, both the inner and outer shells have round ribs, and both ends have seal plates.

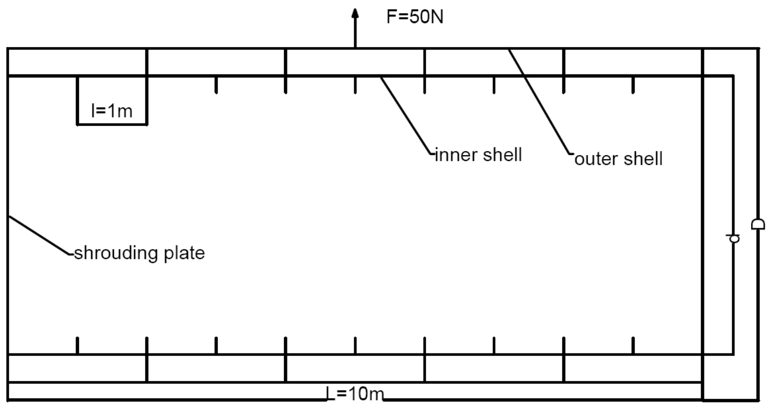

Figure 1: Intersection of the double cylindrical shell structure

For the material properties of the cylindrical shell structure, Young's modulus is 2.05×10^{11}Pa, the Poisson's ratio is 0.3, and the density is 7800kg/m^3. The initial design of the cylindrical shell structure is that, the thickness of inner shell is 0.015m, and the thickness of outer shell is 0.01m. For the material properties of the fluid, the speed of the sound is 1460m/s, the density of water is 1000kg/m^3, and the volume modulus is 2.1204×10^9N/m^2.

According to paper [4], when the space distance for mesh, Δx, is $\Delta x / \lambda < 1/6$, where λ is the wave length, the discrete meshes would satisfy the requirement of accuracy. It means that, in the fluid, there are at least 6 nodes in a wave length. The S4R elements are used for meshing the structure, which is shown in Figure 2.

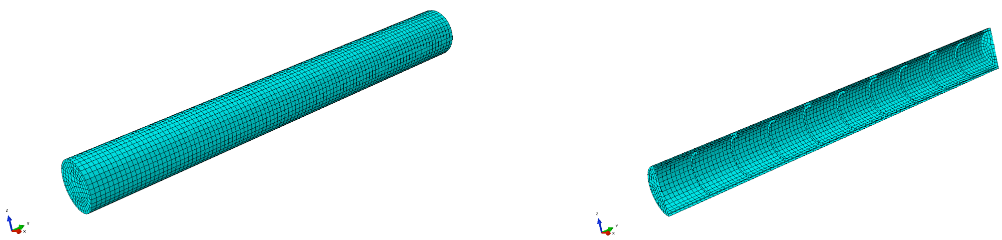

Figure 2: Finite element model of the double cylindrical shell structure

ABAQUS software has special acoustic elements for the acoustic-solid interaction model. In order to assure the computing accuracy, the second order tetrahedron acoustic element, AC3D10, is used for meshing fluid. Zero impedance is set along the boundaries of the outside fluid to achieve the no-reflecting boundary condition. Therefore, the infinite fluid field can be simulated, and the sound waves can be absorbed by the fluid boundaries in distance field. In order to get a better result, the outside boundaries should be far from the structure as much as possible. Generally, it requires a distance that is larger than 1/3 of a sound wave length. A fluid area that is 10 times of the structural overall dimensions is built for the fluid field, which is shown in Figure 3.

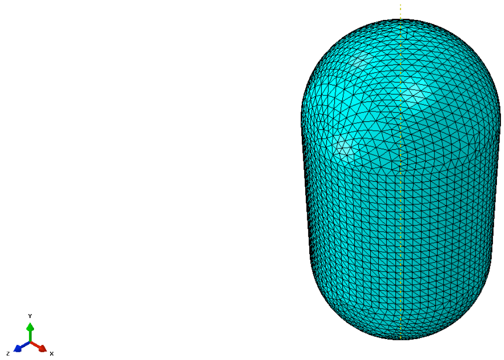

Figure 3: Finite element model of the fluid field

4.2. Average Sound Pressure Level

The acoustic module in ABAQUS is utilized for structural sound radiation analysis. The node that is 1m far from the outer shell under the initial design, Node 7078, is selected as the designated point, and the sound pressure frequency response curve at this node from 0 to 350Hz is obtained and shown in Figure 4.

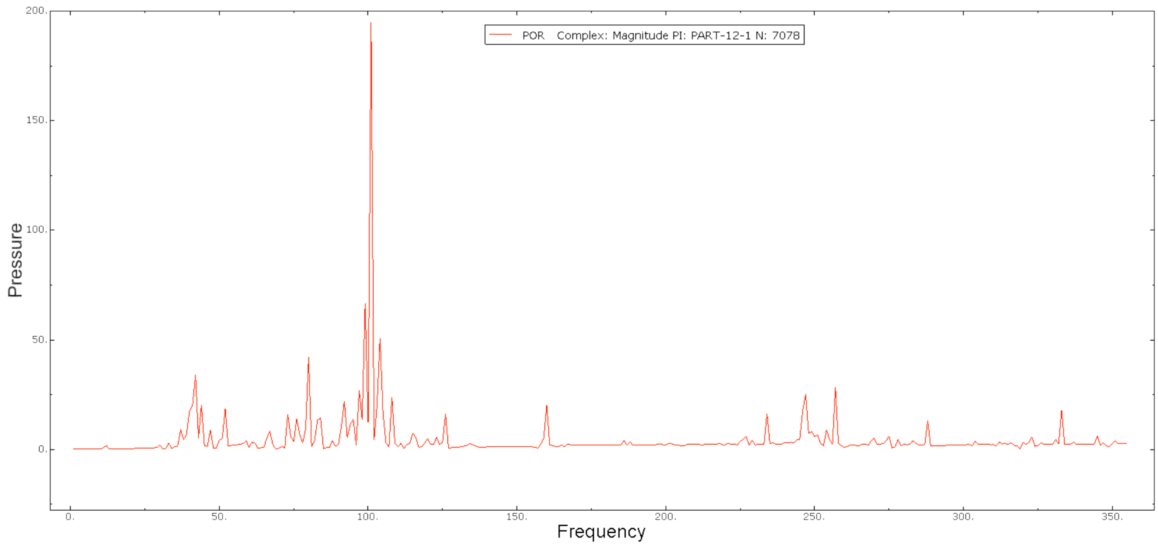

Figure 4: Sound pressure frequency response curve at Node 7078

The sound pressures (SP) of the designated point at the center frequencies of 1/3 oct frequency band within 300Hz are observed in this paper. The sound pressure level (SPL) can be obtained according to the following formula

$$L_p = 20\lg\frac{p}{p_0} \tag{1}$$

where L_p is the sound pressure level (SPL) (unit: dB), p is the sound pressure, and p_0 is the standard sound pressure, which equals to 10^{-6}Pa in the water. Table 1 shows SP and SPL values at Node 7078.

Table 1: Sound pressure and sound pressure level values at middle frequencies of 1/3 octave

Frequency/Hz	SP/Pa	SPL/dB

16	0.295	89.406
20	0.166	84.417
25	0.486	93.738
31.5	0.836	98.446
40	17.659	124.939
50	4.201	112.467
63	0.434	92.747
80	41.944	132.453
100	12.341	121.827
125	3.135	109.925
160	20.180	126.098
200	2.125	106.547
250	5.815	115.291
315	2.311	107.275

Then, the average sound pressure level (APL) at Node 7078 is calculated based on the flowing formula

$$L_{AP} = 10\lg \sum_{k=1}^{N} 10^{0.1L_{p_k}} \tag{2}$$

where L_{AP} is the APL value, which is also called the structural synthetic sound pressure level. "k" is the number of the sound pressure, and N is the total number of sound pressures that are observed, which equals to 14 according to Table 1. L_{p_k} is the k-th sound pressure. Therefore, the APL value for the structure under the initial design is 134.32bB.

Since different thicknesses of inner and outer shells will yield different underwater sound radiation characteristics, the APL value at the designated point is considered as a designated indicator, and it should be no more than 125dB. Based on this constraint, optimization design for the cylindrical shell structure will be performed in the following sections to seek the lightest structure and a lower sound radiation pressure at the same time.

5. Optimization model

The thicknesses of the inner and outer shells are considered as the design variables, and t_1 and t_2 are for the thicknesses of the inner and outer shells separately. They range from 0.01m to 0.05m. The mass of the shell structure is the objective, and the APL value at the designated point is the constraint. Therefore, the optimization model for sound radiation of the cylindrical shell structure is formulated as below

$$\begin{aligned} &\min Mass \\ &\text{s.t.}\quad L_{AP} \le 125 \\ &\qquad 0.01 \le t_1 \le 0.05 \\ &\qquad 0.01 \le t_2 \le 0.05 \end{aligned} \tag{3}$$

where *Mass* is the structural mass, and it is expressed as below

$$Mass = 98195.76\pi t_1 + 78000\pi t_2 \tag{4}$$

Because the expression of L_{AP} regarding to the design variables is unknown, a well-established method, the surrogate modeling method is employed to construct an explicit expression of L_{AP}. Then, the optimization model in Eq. (3) can be solve smoothly based on the surrogate model of L_{AP}.

6. Surrogate modeling for APL

6.1. Design of experiment

The Latin Hypercube Sampling (LHS) method is an effective way to select sampling points in the process of design of experiment (DOE). Compared to the random sampling method, LHS can make sure the range of each design variable is completely covered, and sampling points are distributed at each level as much as possible. So the LHS method is used here to select sampling and test points.

In order to decrease the approximation error, 4 corner points in the design area are firstly selected as sampling points. Then, another 16 points are selected from the design area by using the LHS method. A total of 20 sampling points are selected for approximating surrogate model and their distribution in the design area is shown in Figure 5.

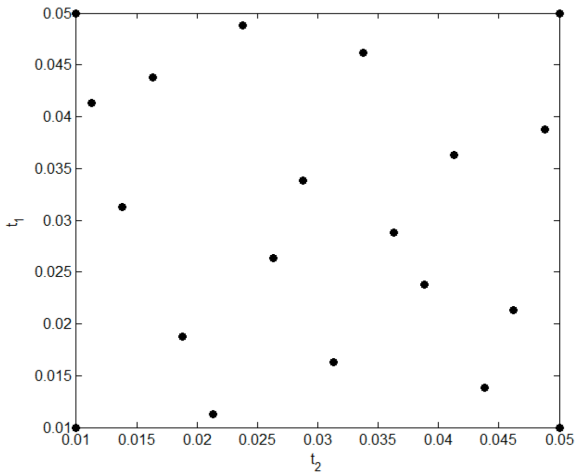

Figure 5: Distribution of 20 sampling points

Based on the finite element analysis in Section 3, the responses at each sampling point are obtained from ABAQUS, where the script language Python is used for pre- and post-process of the FE model. Programming is coded to change the thicknesses of inner and outer shells, submit FEA job for calculation, and read results from result files. Therefore, 20 APL values of sampling points are obtained automatically and quickly.

Because of the large differences between the values of design variables and APL, it is necessary to normalize the data. All variables are normalized such that '0' corresponds to the minimum value and '1' corresponds to the maximum value of the variable. The normalization is carried out with the following formulations

$$x_i = \frac{t_i - t_{i\min}}{t_{i\max} - t_{i\min}} \quad (i=1, 2) \tag{5}$$

$$y = \frac{L_{AP} - L_{AP\min}}{L_{AP\max} - L_{AP\min}} \tag{6}$$

where "i" is the number of design variable. t_i and x_i are, respectively, the design variables before and after normalization. $t_{i\min}$ and $t_{i\max}$ denote, respectively, the minimum and maximum values of design variables among 20 sampling points. L_{AP} and y are, respectively, the APL values before and after normalization. $L_{AP\min}$ and $L_{AP\max}$ denote, respectively, the minimum and maximum values among 20 APL values.

In order to test the accuracy of surrogate models, 9 more points are selected as testing points by using the LHS method, and they are used to calculate the error of the approximation. The Root Mean Square Error (RMSE) is used to test the approximate accuracy, and it is expressed by the following formula

$$\text{RMSE} = \sqrt{\frac{1}{n}\sum_{j=1}^{n}(y_j - \hat{y}_j)^2} \tag{7}$$

where "j" is the number of test points. n is the total number of test points and $n = 9$. y_j and $\hat{y}_j$ represent, respectively, the approximated and true values of normalized APL. A lower RMSE value indicates a surrogate model with a higher accuracy.

6.2. Surrogate modeling

Three types of surrogate models, polynomial response surface (PRS), Kriging, and radial basis neural network (RBNN), are utilized in this paper to approximate the APL function. The best surrogate models from each type will be used as the constraint function in Eq. (3).

All of three types of surrogate models are obtained by using the "SurrogateToolbox" integrated within MATLAB, which was developed by Haftka and Viana. The tool box includes several types of surrogate models, and it can be used for regression analysis of surrogate models, multiple errors calculation, and figure plotting [5-6]. It is easily to be used and conducted.

In this paper, 2nd order, 3rd order and 4th order polynomials (represented by order2, order3, and order4 separately) are considered for PRS approximation; zero order, 1st order, and 2nd order polynomial regression functions (represented by regpoly0, regpoly1, and regpoly2 separately) with Gaussian correlation function are considered for Kriging approximation; three values of the spread constant, 0.2, 0.5, and 1.0 for the radial basis function (represented by spread_0.2, spread_0.5, and spread_1.0 separately) are considered for RBNN approximation. RMSE values for each observed surrogate models are shown in Table 2.

Table 2: Comparison of surrogate models in RMSE

PRS	RMSE	KRG	RMSE	RBNN	RMSE
order2	0.309	regpoly0	0.342	spread_0.2	0.386
order3	0.307	regpoly1	0.347	spread_0.5	0.427
order4	0.530	regpoly2	0.343	spread_1.0	0.524

The RMSE values in Table 2 indicate that the 3rd order PRS, "regpoly0" Kriging, and "spread_0.2" RBNN are three nominations of the APL function. Therefore, three optimization models based on three approximations of APL functions are constructed, and the solutions are carried out in MATLAB with "fmincon" function.

6.3. Optimization results

Based on the 3rd order PRS, the optimal point is $[t_1^*, t_2^*] = [0.01, 0.01]$ (/m), and structural total mass is *Mass*=5.535×10^3kg. At the optimal point, the APL value from surrogate model is L_{AP}=124.83dB, and the true value of APL is L_{AP}^*=129.80dB. The relative error between them is 3.83%.

Based on the "regpoly0" Kriging, the final results are $[t_1^*, t_2^*] = [0.0348, 0.0240]$ (/m), *Mass*=1.661×10^4kg, and L_{AP}=125.00dB. The true value of APL is L_{AP}^*=122.43dB. The relative error between them is 2.10%.

Based on the "spread_0.2" RBNN, the final results are $[t_1^*, t_2^*] = [0.0349, 0.0231]$ (/m), *Mass*=1.661×10^4kg, and L_{AP}=125.00dB. The true value of APL is L_{AP}^*=122.31dB. The relative error between them is 2.20%.

The results indicate that, the relative errors between the approximated and surrogated APL at the optimal points are less than 5%, and the "regpoly0" Kriging and "spread_0.2" RBNN yield better results than the 3rd order PRS. Therefore, both Kriging and RBNN can approximate the APL function very well. Their results demonstrate the validity and effectiveness of the employed methods in the optimization design for structural sound radiation.

7. Conclusions

This paper uses the ABAQUS-based integrated analysis method for acoustic-sold interaction problems to compute sound radiation of the underwater double cylindrical shell structure. The surrogate modeling method is employed successfully to obtain the approximation of average sound pressure level at the designated point. Fine results are achieved in the multidisciplinary design optimization of structural sound radiation. This research provides references for predicting the acoustic radiation of underwater structures and performing acoustic design.

8. Acknowledgements

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