Effect of reducing the vibration of a frame-spacing arrangement

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Abstract. The methods of arranging frame spacing are explored for reducing the vibration of a cylindrical shell. According to the principle that an aperiodic structures will produce irregular reflections on structural waves, resulting in structural vibration attenuation, two methods of arranging frame spacing are adopted: periodic frame spacing and random frame spacing. Using the two methods, the frames of cylindrical shell models are arranged, and then, using the finite element method (FEM), the vibration response of the cylindrical shells is calculated and compared. It can be found that, for the finite-length cylindrical shells, the aperiodic arrangement of the frame spacing can decrease the vibration level of the cylindrical shells in some specific high-frequency band. For example, for the cylindrical shells with a diameter of 7.6 m and a length of 8.4 m, the frame-spacing arrangement is random, and the vibration level is reduced in about 350-600 Hz, with a maximum reduction of up to 9 dB.

Keywords: frame spacing, vibration, cylindrical shell, aperiodic arrangement, random arrangement.

1. Introduction

A submarine is mainly composed of a framed cylindrical shell, so the vibration characteristics of the framed cylindrical shell directly determine the vibration characteristics of the submarine. In addition, with the development of modern sonar systems and numerous advanced underwater detection technologies, the concealment of submarines is extremely important in modern warfare. Therefore, it is very important to study the characteristics of the vibration and radiated noise of a framed cylindrical shell, which has always been the focus of many scholars.

So far, in the study of the vibration characteristics of framed cylindrical shells, the arrangement of the frame spacing has usually been periodic. Wang et al. [1] analyzed the free flexural vibration of a cylindrical shell horizontally immersed in shallow water. In Refs. [2] and [3], the semi-analytical method was developed for vibro-acoustic analysis of submerged ring-stiffened cylindrical shells. Using analytical and experimental methods, Zhao et al. [4] analyzed the vibro-acoustic behavior of a semi-submerged finite cylindrical shell. In Refs. [5] and [6], boundary conditions were considered in the study of the vibro-acoustic behavior of cylindrical shells. For a periodic structure, such as coupled multi-span beams [7], a periodically ribbed plate [8-10], or a periodically framed cylindrical shell [11], the study of the vibration characteristics shows that the passband alternates with the stopband in the frequency domain. In the passband, the vibration of periodic structure can propagate freely, but in the stopband, the vibration of periodic structure exponentially decays away from the source.

At present, the study of aperiodic structures is mostly focused on one-dimensional structures or other simple structures, such as mono-coupled multi-span beams with large deterministic disorders [12], rectangular plates and membranes under fluid loading [13, 14], and disordered multi-span beams with damping [15-18]. Several important conclusions have been drawn, e.g., Anderson localization exists in the vibration characteristics of aperiodic structures [19-21]. Not many studies on the vibration characteristics of a cylindrical shell with frame aperiodicity exist [22-24], and those that do are mainly aimed at infinite-length cylindrical shells [25]. The results in [25] show that, except for the low-order vibration mode in the circumferential direction, the obvious phenomenon of Anderson localization exists and is increasingly obvious with increasing order of the circumferential-direction vibration mode. The main reason is that, with increasing order of the circumferential-direction vibration mode, the effect of the frame impedance increases, and the coupling action between the adjacent sub-structures is weakened by the influence of the frame impedance.

In practical engineering, a framed cylindrical shell is of finite length. In this paper, the control effect of the frame aperiodicity on the vibration of a cylindrical shell is investigated. The analysis method used in this paper is of practical significance for the study of the vibration mechanism of cylindrical shells, and the conclusions drawn herein can be used to guide the design of cylindrical shells.

2. Basic theory

2.1. Vibration localization of aperiodic structures

According to the vibration characteristics of simple structures, such as one-dimensional aperiodic mass-spring chain models, and one-dimensional aperiodic bending beams, the generation of vibration localization mainly meets the following two conditions [25]:

1) The parameter arrangement of the substructure is random.

2) The larger the number of the substructure, the quicker the amplitude of the structural wave decays with increasing propagating distance.

When the sum of frames is enough, and the frame spacing is random, then the vibration localization may be found for finite-length cylindrical shell.

The detailed positions of the frames along the shell are distributed randomly within a small region about the nominal periodic location; i.e., the probability density that the position of a particular frame near $x = nd$ has position x is:

$$P(x) = \begin{cases} 1/(2\Delta x), & |x - nd| < \Delta x, \\ 0, & \text{otherwise}, \end{cases} \tag{1}$$

where d is the mean spacing of the frames along the shell, n is the number of a particular frame, Δx is the maximum deviation of the position of a frame from its nominal periodic location. Δx reflects the extent to which the detailed positions of the frames along the shell are distributed randomly. For finite-length cylindrical shell, in order to obtain obvious vibration localization effect, the Δx should be increased as much as possible under the condition of guaranteeing the random distribution of frames spacing.

2.2. Calculation method of vibration localization

The mean square normal velocity of the surface is adopted to show the vibration characteristics of the framed cylindrical shell. For the arbitrary surface S, the normal velocity distribution of the shell is $v(x, y, z, t)$, and the mean square normal velocity $\langle \bar{v}^2 \rangle$ of the shell can be obtained:

$$\langle \bar{v}^2 \rangle = \frac{1}{TS_\partial} \int_S \int_t^{t+T} v^2(t) dt\, ds, \tag{2}$$

where t is time, T is integration time length, S_∂ is area of S.

At the typical frequency of the excited force, the vibration response of the cylindrical shell was calculated by FEM. Then, the mean square normal velocity was obtained by using the vibration response $v(x, y, z, t)$ as the input [26]:

$$\langle \bar{v}^2 \rangle = \frac{\frac{1}{2}\sum_{i=1}^{N} |v_i|^2 S_i}{\sum_{i=1}^{N} S_i}, \tag{3}$$

where N is the sum of the finite elements of the shell, $|v_i|$ is the amplitude of the normal velocity at the centroid of the i'th element, S_i the area of the i'th element. By taking the logarithm of $\langle \bar{v}^2 \rangle$, the $\langle \bar{v}^2 \rangle$ level $L_{\bar{v}^2}$ is obtained:

$$L_{\bar{v}^2} = 10\log_{10}\left(\frac{\langle \bar{v}^2 \rangle}{v_{ref}^2}\right), \tag{4}$$

where $v_{ref} = 5.0\times10^{-5}$ mm/s is the reference velocity.

By comparing the $\langle \bar{v}^2 \rangle$ of the cylindrical shells with frame periodicity and with frame aperiodicity, the reducing vibration effect of a frame-spacing arrangement is explored.

3. Comparison of vibration of framed cylindrical shell

3.1. Structural design of framed cylindrical shell

For the studied cylindrical shell with frame periodicity, the frame spacing was 0.6 m, the shell thickness 0.028 m, and the length 8.4 m, and the bulkhead on the two sides was kept invariant; the structural form is shown in Fig. 1.

On the base of the above cylindrical shell with frame periodicity, design the random arrangement of frame spacing. The probability density of the position of a particular frame x satisfies Eq. (1), in which $d = 0.6$ m, Δx equal to 3.5 % of the frame spacing of the cylindrical shell with frame periodicity, $\Delta x/d = 0.035$. There were 13 frames for each cylindrical shell; $n = 13$, that is to say, 14 frame spacings were generated by the random method, with each measuring 0.4–0.8 m in length. Fifty groups of frame spacings were generated; in other words, 50 framed cylindrical shell models were made, named models 1-50, as shown in Table A1 (in Appendix). Obviously, the arrangement of frame spacing is aperiodic, e.g. the sketches of the frame-spacing arrangement of model 1 is shown in Fig. 2.

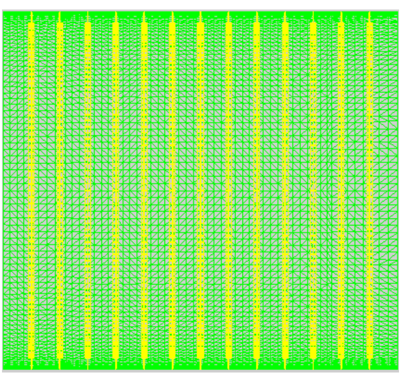


Fig. 1. Structure design of cylindrical shell with frame periodicity

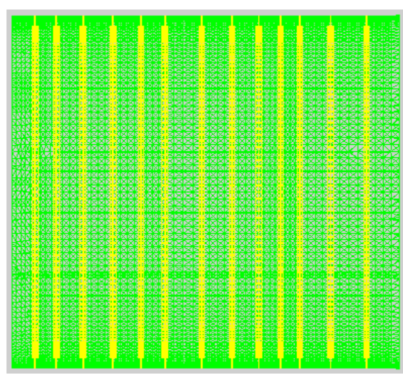
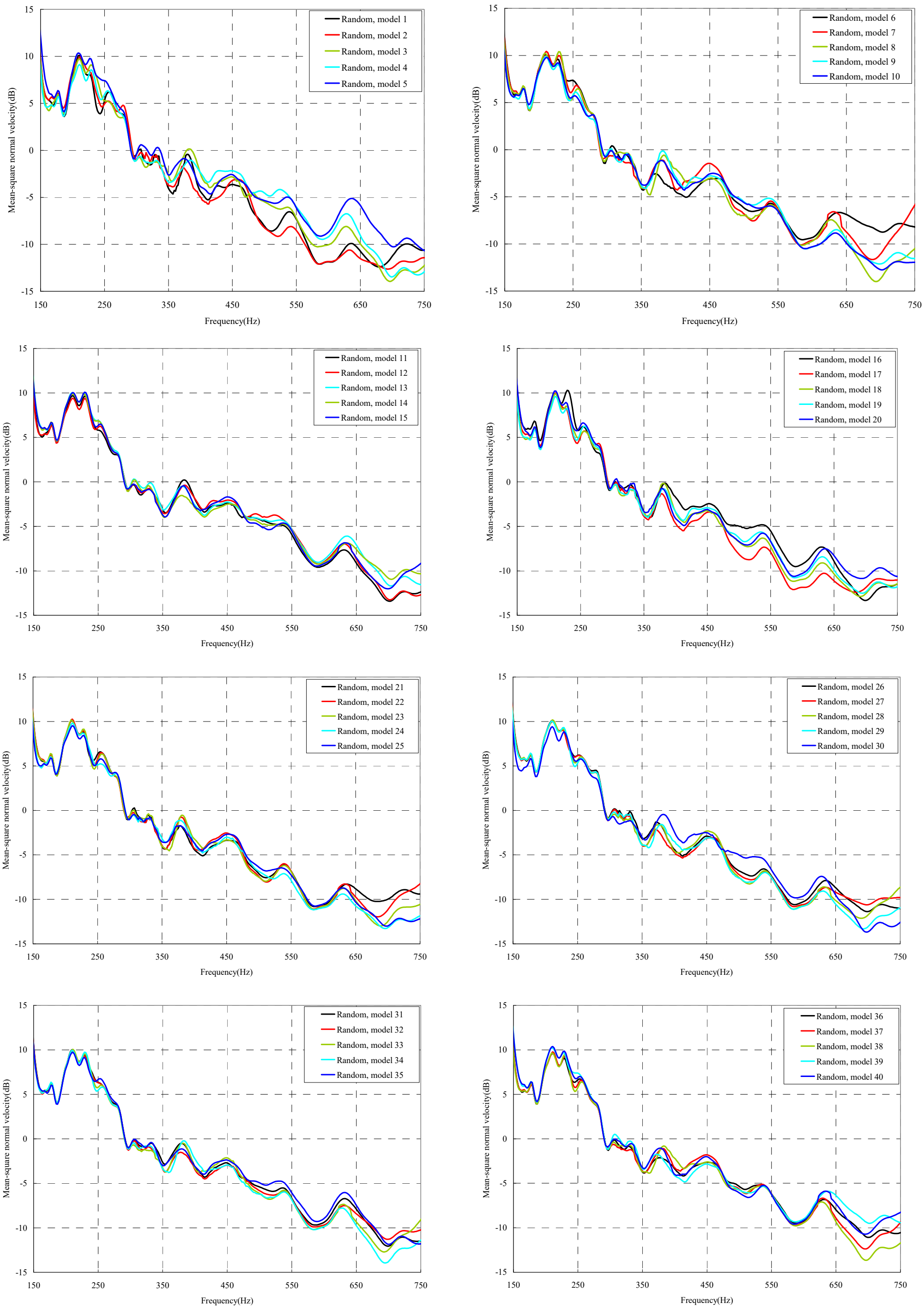


Fig. 2. Frame-spacing arrangement of model 1

3.2. Vibration response of framed cylindrical shell in air

The vibration responses of the cylindrical shell with frame periodicity and frame aperiodicity were separately calculated by FEM in air, and MSC.Patran was used to build the models of the framed cylindrical shell, therefore the accuracy of the calculation was guaranteed.

There are two calculation cases: one is the application of axial exciting force acting on the base of the floating raft, and the other is the application of vertical exciting force acting on the base of the floating raft. The frequency band of the exciting force is 20-1000 Hz, and the calculation result of the mean square normal velocity which was obtained by Eq. (3) is compared and analyzed.



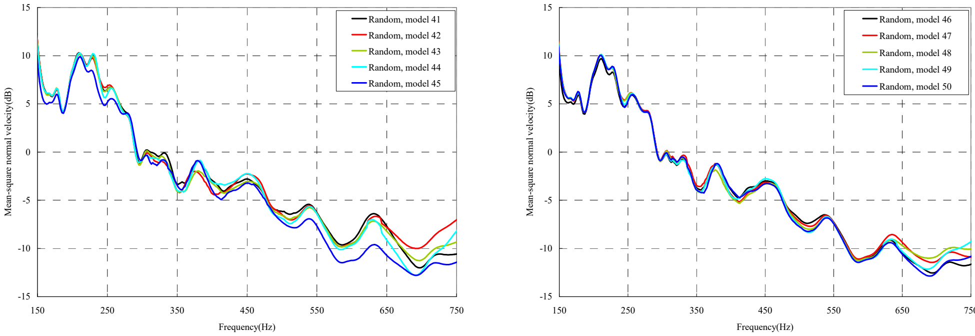
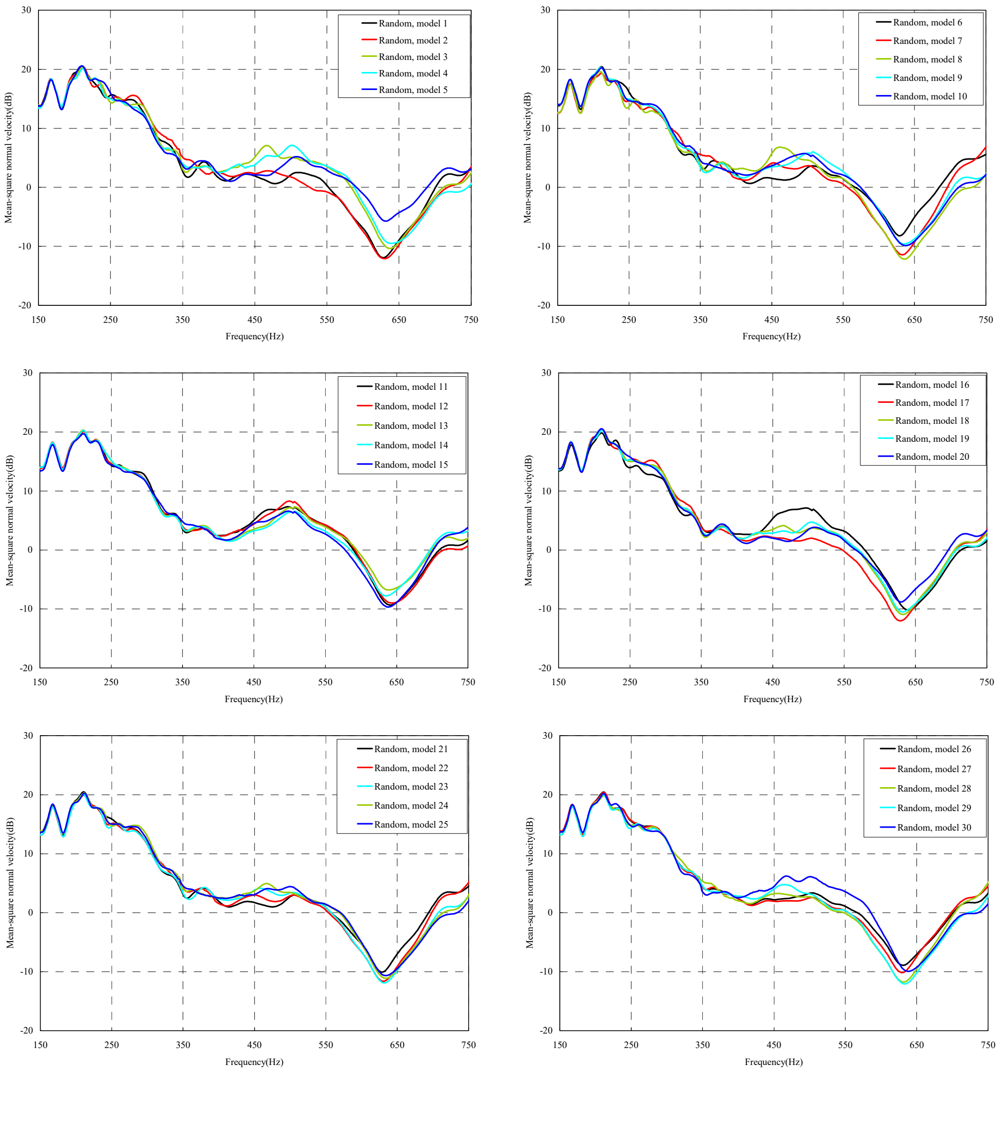


Fig. 3. Comparison among 50 models under axial loads



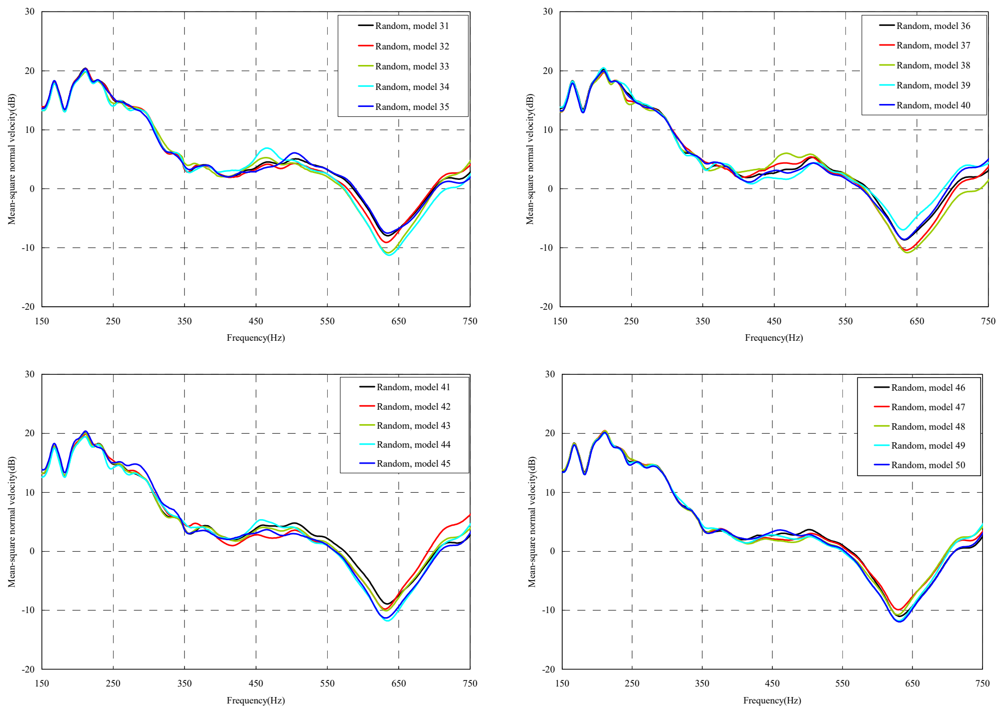
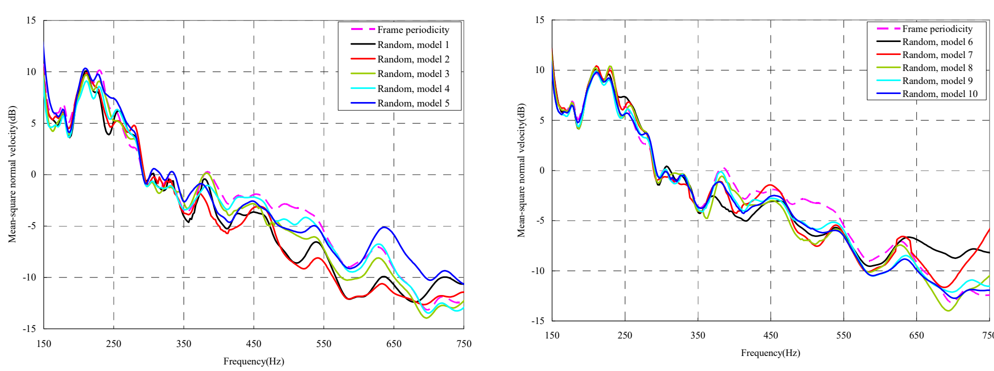


Fig. 4. Comparison among 50 models under vertical loads

The mean square normal velocity of the 50 cylindrical shells with random frame spacing arrangements introduced in Section 3.1 were compared and analyzed. It can be seen that before 700 Hz, the curves of vibration response cross-connect, and, on the whole, the difference among the results of the 50 cylindrical shells is small, except for some parts, which are shown in Figs. 3 and 4. Fig. 3 plots the vibration response of the case in which axial exciting force acting on the base of the floating raft is applied and Fig. 4 that of the case in which vertical exciting force acting on the base of the floating raft is applied.

The vibrations of the cylindrical shell with frame periodicity and that with a random frame-spacing arrangement were compared, as shown in Figs. 5 and 6. Fig. 5 plots the vibration response of the case in which axial exciting force acting on the base of the floating raft is applied and Fig. 6 that of the case in which vertical exciting force acting on the base of the floating raft is applied.



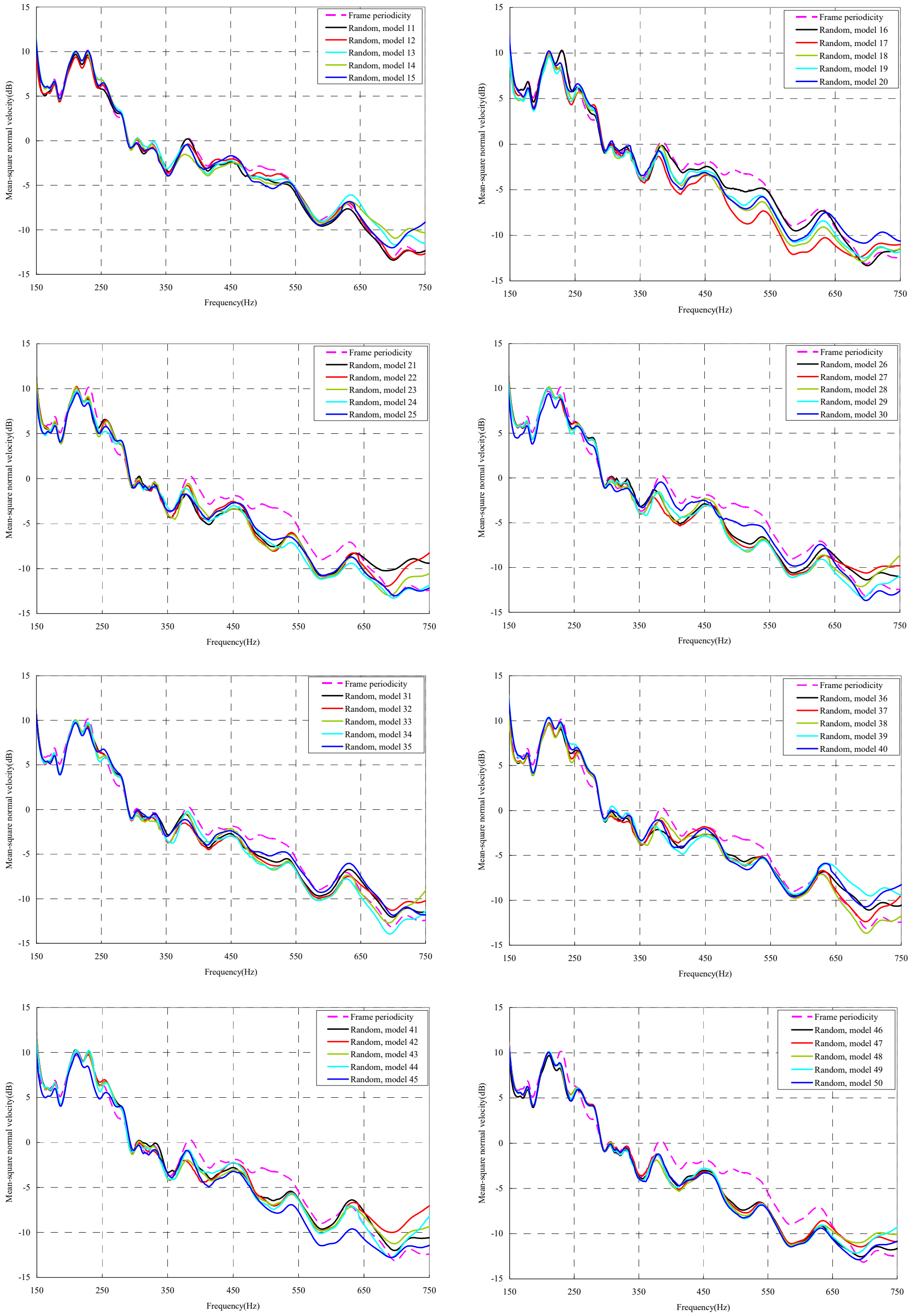


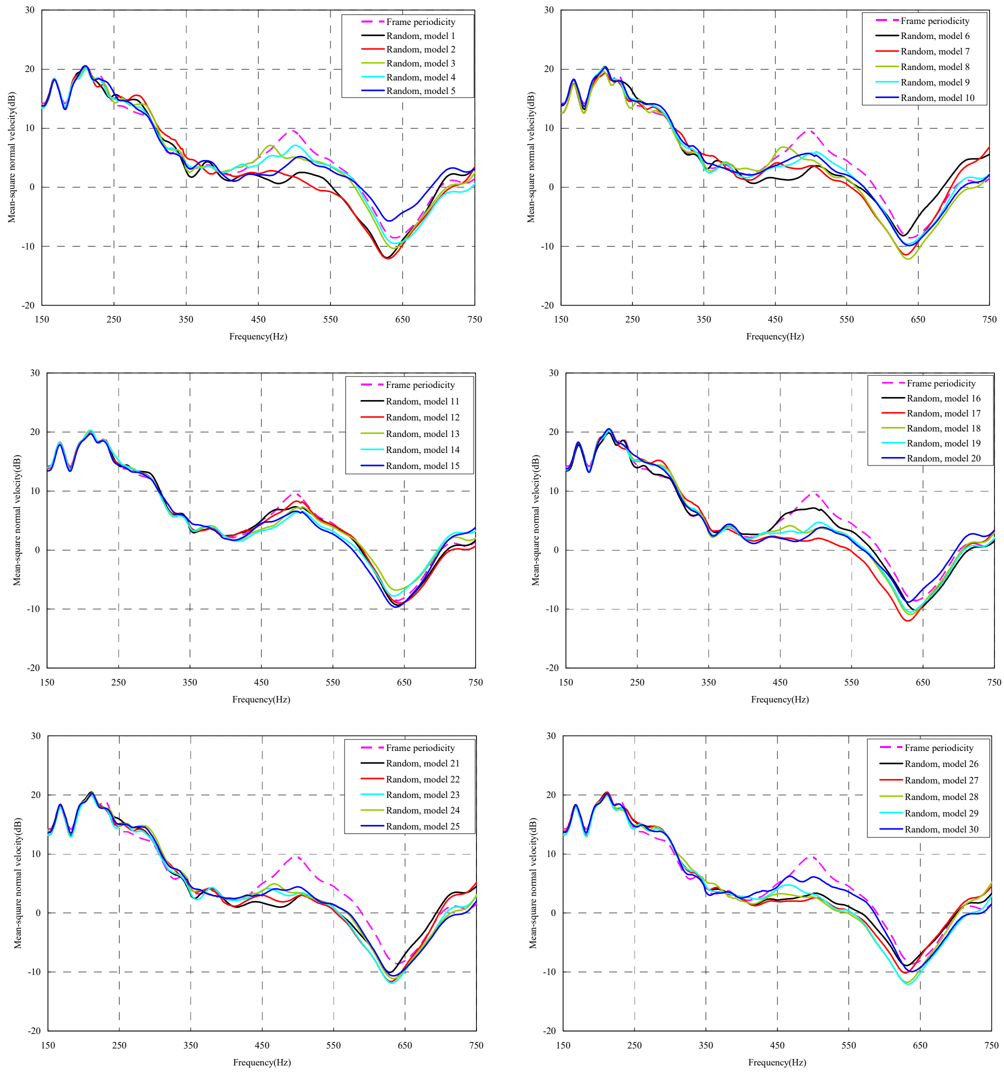
Fig. 5. Comparison between periodic and random arrangements under axial loads

From the results in Figs. 5 and 6, the following conclusions can be drawn:

1) For the finite-length cylindrical shell, when the frequency of the exciting force is less than

350 Hz, the difference in the vibration response between the cylindrical shell with frame periodicity and that with frame aperiodicity is not obvious. The reason is that in the low-frequency band the wavelength of the structural wave is relatively longer, so the method of arranging the frame spacing has little influence on the total vibration of the cylindrical shell. However, with increasing frequency, the wavelength of the structural wave becomes relatively shorter, and thus, the influence of the method of arranging the frame spacing on the total vibration of the cylindrical shell becomes increasingly more obvious. When the frequency of the exciting force is in the range of about 350-600 Hz, compared with the periodic arrangement of the frame spacing, the aperiodic arrangement of the frame spacing can decrease the vibration of the cylindrical shell with a maximum reduction of up to 9 dB.

2) For the finite-length cylindrical shell, the aperiodic arrangement of the frame spacing can decrease the vibration of cylindrical shell, but the effect exists only in some narrow frequency band, generally at medium-high frequency.



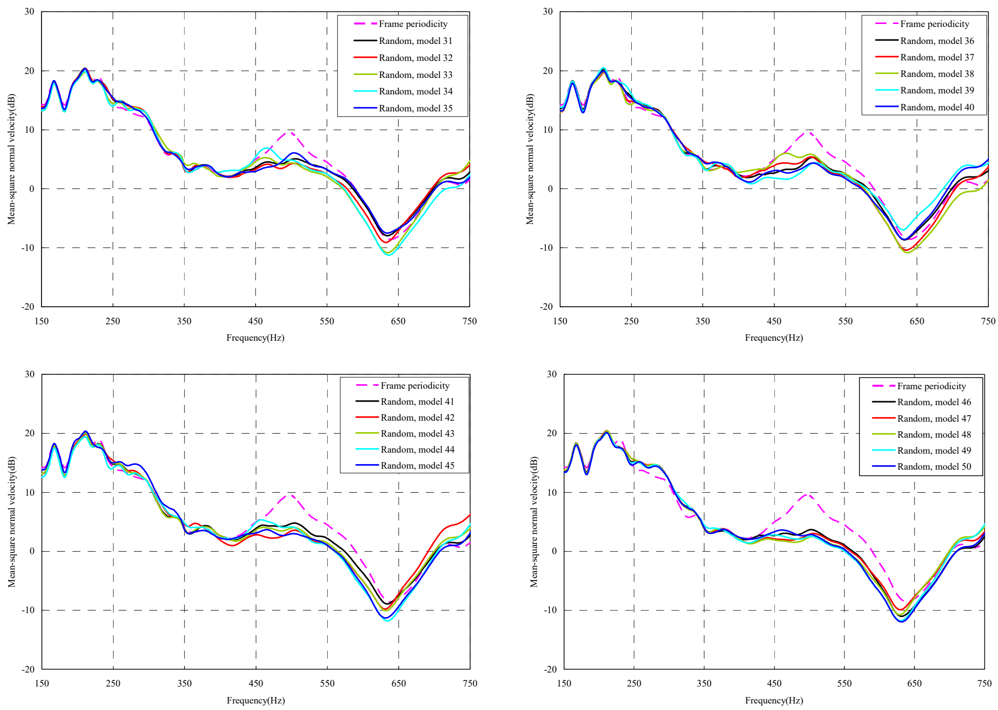
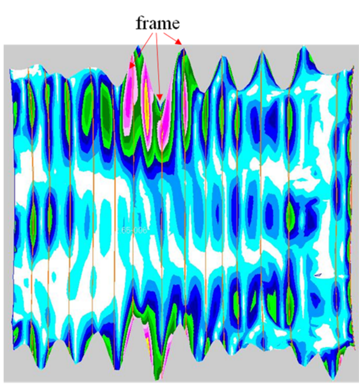
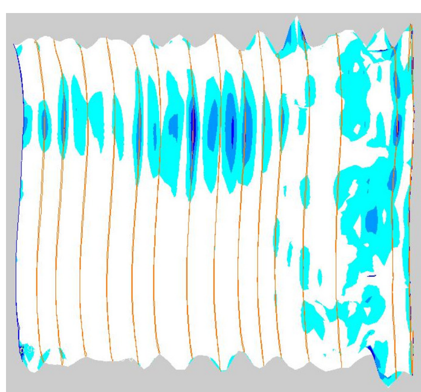


Fig. 6. Comparison between periodic and random arrangements under vertical loads

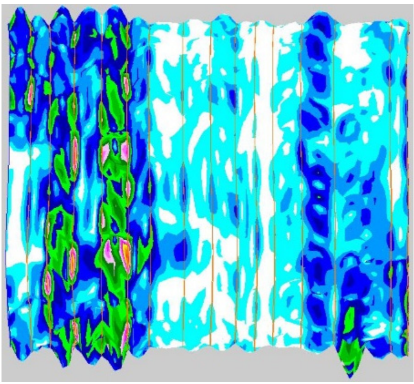
3) Changes in frame spacing mainly affect the propagation of the structural waves between adjacent frames. Taking model 1 as an example, the first 4000-order modes were obtained through numerical simulation, and it can be seen that when the natural frequency is less than 350 Hz, the structural wavelength on the shell plate exceeds the distance between adjacent frames, as shown in Fig. 7(a) and Fig. 7(b), so when the frequency is less than 350 Hz, the change in frame spacing has little effect on vibration. When the natural frequency is greater than 350 Hz, the vibration on the shell plate is dominated by the vibration between adjacent frames, and the structural wavelength is usually less than the distance between adjacent frames, as shown in Fig. 7(c)-7(f), so when the frequency is greater than 350 Hz, the frame-spacing changes affect the vibration, but when the frequency exceeds 600 Hz, the vibration on the shell plate is mainly local vibration, as shown in Fig. 7(g), the effect of frame-spacing changes on vibration is reduced.



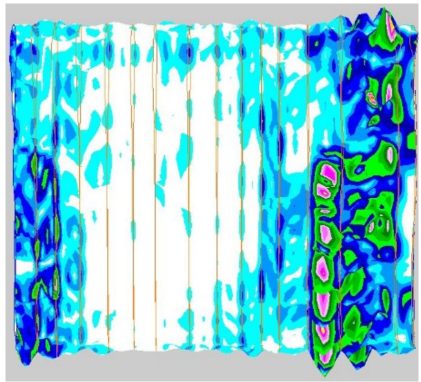
a) 250.59 Hz



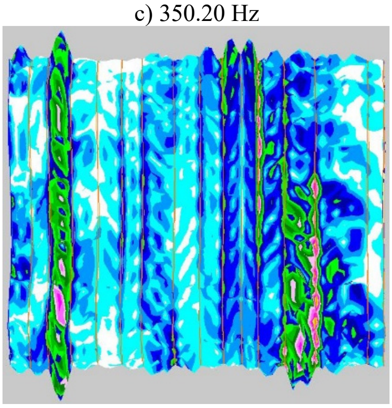
b) 312.95 Hz



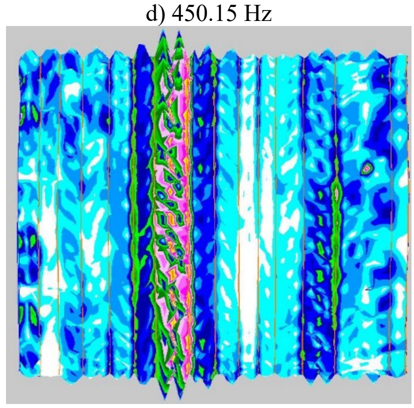
c) 350.20 Hz



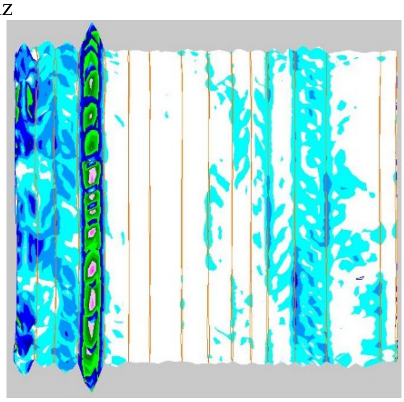
d) 450.15 Hz



e) 550.33 Hz



f) 590.61 Hz



g) 610.69 Hz

Fig. 7. Structural wavelength vs. frequency

4. Experimental verification of the effectiveness of the frame-spacing arrangement method

4.1. Experimental models

There were two experimental models, which were represented by Model s1 and Model s2. The structural scale of both models was the same, only the position of the frames was different. The models are made of steel material, and their main dimensions and material parameters are shown in Table 1.

Both models were arranged along the axial direction of the shell with 6 frames, of which Model s1 was equally spaced, the frame spacing was 700 mm. On the base of Model s1, design the random arrangement of frame spacing. The probability density of the position of a particular frame x satisfies Eq. (1), in which $d = 0.7$ m, Δx equal to 30 % of the frame spacing of the cylindrical shell with frame periodicity, $\Delta x/d = 0.3$. There were 6 frames for each cylindrical shell; $n = 6$, that is to say, 7 frame spacings were generated by the random method, with each measuring 0.5-0.9 m in length. Due to the axial length of the experimental models are shorter compared to the models in Section 3.1 of my paper, thirty groups of frame spacings were generated; in other words, 30 framed cylindrical shell models were made, named models 1-30, as shown in Table 2. Obviously, the arrangement of frame spacing is aperiodic. The vibration responses of the cylindrical shells with frame periodicity (Model s1) and frame aperiodicity (models 1-30) were separately calculated by FEM in air, and MSC.Patran was used to build the models of the framed cylindrical shell, therefore the accuracy of the calculation was guaranteed. The frequency band of the exciting force is 1-1000 Hz. On the base of the vibration responses, the mean square normal velocity was obtained by Eq. (3). Calculate the average area enclosed by the "mean square normal velocity" curve and the "frequency" axis for all models according to the following Eq. (5), and the result is expressed in $\bar{V}$. f_2 and f_1 in the Eq. (5) represent the upper and lower limits of the integration, respectively, where $f_1 = 1$ Hz, $f_2 = 1000$ Hz:

$$\bar{V} = \frac{1}{f_2 - f_1}\int_{f_1}^{f_2} \langle \bar{v}^2 \rangle df. \tag{5}$$

Table 1. Structure parameters and material

Structure parameters		Material	
Cylindrical shell length	4900 mm	Elastic modulus	2.05×10^5 N/mm^2
Cylindrical shell diameter	1500 mm	Density	7.80×10^{-6} kg/mm^3
Shell thickness of cylindrical shell	6 mm	Poisson ratio	0.3
Plate thickness of end cap	6 mm	Structural damping coefficient	0.06
Frame height	150 mm		
Frame thickness	6 mm		

Take the logarithm of $\bar{V}$ to get $L_{\bar{V}}$, and $L_{\bar{V}}$ represents the total vibration level of the model:

$$L_{\bar{V}} = 10\log_{10}\left(\frac{\bar{V}}{v_{ref}^2}\right). \tag{6}$$

The total vibration level of Model s1 is 10.90 dB, and the total vibration levels of the models in Table 2 are shown in Table 3. It can be seen that the vibration levels of all models in Table 3 are smaller than Model s1. In Table 3, the total vibration level of Model 23 is the smallest, therefore, based on the frame spacing of Model 23, the experimental model with the aperiodic frame-spacing was processed. For the convenience of processing, the frame spacing of Model 23 is fine-tuned, and the experimental model shown in in Fig. 8 is obtained, named Model s2.

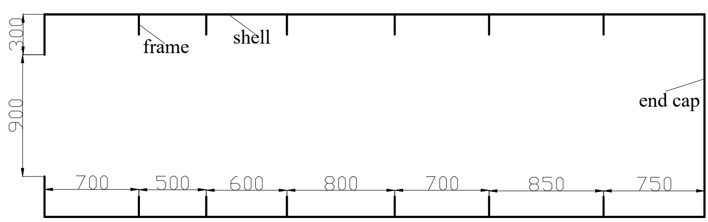


Fig. 8. Aperiodic frame spacing of Model s2

To facilitate the installation of the exciter, the cylindrical shell was hoisted horizontally (as shown in Fig. 9(a)). The exciter generated a sinusoidal excitation force F, which acted on the cylindrical shell through the excitation rod (as shown in Fig. 9(c)). To simulate the free boundary conditions, the cylindrical shell was suspended at both ends, and connected to the hoisting equipment with a soft rope (as shown in Fig. 9(b)).

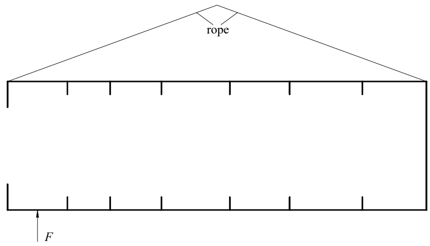
Table 2. Aperiodic frame spacing of all models for experiment

Model name	1st frame spacing (mm)	2nd frame spacing (mm)	3rd frame spacing (mm)	4th frame spacing (mm)	5th frame spacing (mm)	6th frame spacing (mm)	7th frame spacing (mm)
Model 1	700	880	580	630	860	540	710
Model 2	530	730	830	710	690	820	590
Model 3	690	710	740	510	680	790	780
Model 4	590	830	650	860	700	670	600
Model 5	630	800	640	570	750	660	850
Model 6	700	650	900	660	770	620	600
Model 7	620	850	820	510	540	720	840
Model 8	650	820	720	560	840	590	720
Model 9	800	550	810	710	640	630	760
Model 10	600	830	610	750	660	740	710
Model 11	750	640	600	800	630	870	610
Model 12	650	780	570	780	700	820	600
Model 13	620	840	570	600	540	860	870
Model 14	730	700	630	750	620	800	670
Model 15	600	760	570	830	640	780	720
Model 16	730	780	800	670	730	520	670
Model 17	660	540	790	620	750	750	790
Model 18	770	810	630	520	750	690	730
Model 19	740	830	780	700	650	480	720
Model 20	870	680	580	740	710	650	670
Model 21	570	830	820	630	690	750	610
Model 22	640	650	690	760	780	680	700
Model 23	710	490	620	790	690	850	750
Model 24	640	580	790	770	830	770	520
Model 25	640	720	830	690	700	620	700
Model 26	500	520	700	830	730	850	770
Model 27	530	570	730	800	850	670	750
Model 28	690	820	600	550	670	800	770
Model 29	730	530	690	610	680	830	830
Model 30	710	500	730	760	810	670	720

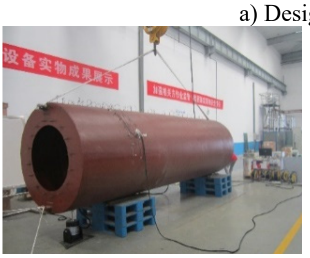
Table 3. Total vibration levels of models 1-30

Model name	Total vibration level (dB)	Model name	Total vibration level (dB)	Model name	Total vibration level (dB)
Model 1	2.75	Model 11	3.39	Model 21	3.70
Model 2	4.58	Model 12	5.32	Model 22	3.49
Model 3	4.25	Model 13	3.63	Model 23	2.01
Model 4	2.81	Model 14	5.07	Model 24	5.95
Model 5	2.78	Model 15	3.87	Model 25	4.70
Model 6	2.69	Model 16	3.35	Model 26	5.82
Model 7	5.41	Model 17	2.87	Model 27	5.52
Model 8	2.86	Model 18	5.91	Model 28	2.67
Model 9	3.00	Model 19	5.75	Model 29	3.38
Model 10	4.50	Model 20	3.52	Model 30	2.93

In this experiment, the acceleration sensor was used to measure the vibration response at each typical position of the structure, and the arrangement of the acceleration sensor mainly considered two factors: one was to ensure that one structure wave or half of the structure wave between the frames could be measured, and the other was that the sensor was densely arranged on the structure near the vibration source. The arrangement of the acceleration sensors was shown in Fig. 10.



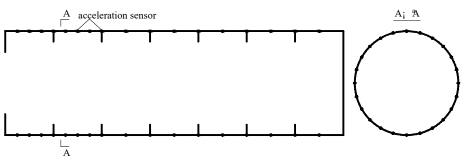
a) Design sketch of hoisting model



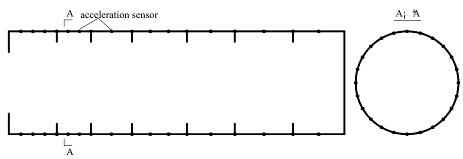
b) Experimental model

c) Installation of vibration exciter

Fig. 9. Hoisting of model



a) Sensor arrangement of Model s1

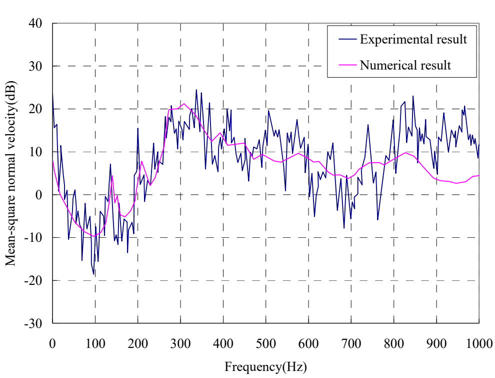


b) Sensor arrangement of Model s2

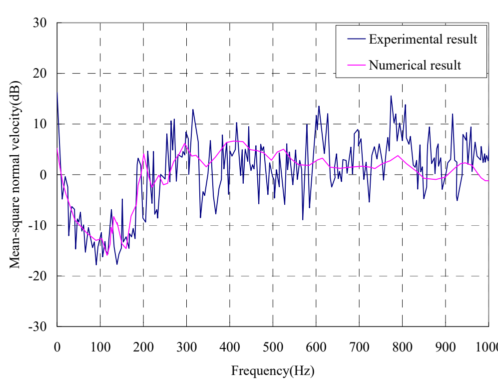
Fig. 10. Sensor arrangement of models

4.2. Experimental validation

The vibration response of Model s1 and Model s2 was obtained experimentally, and the mean square normal velocity of the cylindrical shell was calculated by Eq. (3), which was compared with the result by the numerical method, as shown in Fig. 11. It can be seen that the experimental and the numerical results were in good line.



a) Result of Model s1



b) Result of Model s2

Fig. 11. Comparison between experimental results and numerical results of mean square velocity

Comparing the experimental results of Model s1 and Model s2, as shown in Fig. 12, it can be seen that in some frequency band (e.g., 220-440 Hz, 460-600 Hz), the aperiodic frame-spacing arrangement can reduce the vibration of the cylindrical shell.

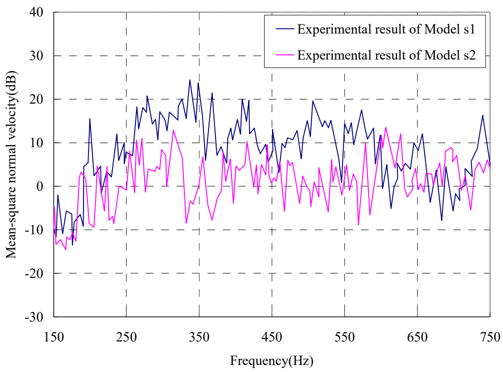


Fig. 12. Comparison between experimental results of two models

5. Conclusions

The vibration characteristics of a cylindrical shell with frame aperiodicity were studied in this paper, and the theory of Anderson localization was extended to the frame acoustic design of the cylindrical shell. The frame spacing was designed by the random method. The cylindrical shell with frame aperiodicity was then constructed. The vibration response of the framed cylindrical shell was calculated and compared, and the following conclusions can be drawn:

1) The influence of the form of the frame arrangement on the vibration of the cylindrical shell is shown mainly in the medium-high frequency band, in which the structural wavelength is smaller than the frame spacing. The influence of the form of the frame arrangement on the vibration of the cylindrical shell is not obvious in the low-frequency band.

2) In some frequency bands, the aperiodic arrangement of the frame spacing can decrease the vibration of the cylindrical shell.

3) The method proposed, and the conclusions drawn in this paper can be taken as the basis of further studies of similar problems.

Acknowledgements

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Appendix

Table A1. Frame spacing of all models

Model No.	1st frame spacing (mm)	2nd frame spacing (mm)	3rd frame spacing (mm)	4th frame spacing (mm)	5th frame spacing (mm)	6th frame spacing (mm)	7th frame spacing (mm)	8th frame spacing (mm)	9th frame spacing (mm)	10th frame spacing (mm)	11th frame spacing (mm)	12th frame spacing (mm)	13th frame spacing (mm)	14th frame spacing (mm)
1	509	459	577	650	605	519	796	655	579	466	418	667	777	723
2	476	456	593	578	684	737	695	720	679	445	413	723	667	534
3	797	732	442	640	546	559	591	473	684	448	792	713	439	544
4	698	610	643	570	613	564	717	425	532	601	507	421	788	711
5	608	755	689	415	577	748	505	789	489	466	465	624	734	536
6	436	731	614	612	403	709	773	665	677	520	407	518	733	602
7	486	709	457	761	428	439	600	585	779	427	687	507	754	781
8	550	709	431	792	752	563	685	516	679	636	572	431	652	432
9	756	487	611	736	484	770	693	509	590	438	632	565	444	685
10	658	536	722	569	656	568	544	768	445	572	549	456	565	792
11	789	582	536	481	599	788	454	645	434	611	449	693	739	600
12	520	629	652	550	773	522	534	705	644	700	531	683	545	412
13	770	665	579	729	453	441	596	433	572	660	631	507	664	700
14	618	519	584	722	531	694	755	666	471	666	589	400	499	686
15	704	507	499	443	642	450	677	522	563	663	490	738	795	707
16	473	701	789	676	567	458	414	438	622	701	591	743	521	706
17	779	717	657	441	604	458	640	483	697	559	500	554	521	790
18	737	687	694	501	456	600	450	765	563	629	563	757	540	458
19	447	465	646	597	571	453	709	619	778	663	739	551	758	404
20	461	656	725	663	488	450	593	691	628	505	661	522	632	725
21	612	775	618	525	541	452	418	722	496	663	405	747	723	703
22	794	572	613	532	781	702	550	429	485	618	666	640	499	519
23	789	617	574	613	787	516	493	711	540	494	652	504	452	658
24	450	683	496	571	519	787	506	592	522	609	563	605	738	759
25	579	463	599	733	700	449	544	710	789	481	608	529	655	561
26	516	462	495	455	621	723	711	714	797	677	605	417	763	444
27	481	508	574	421	576	558	638	689	672	710	731	671	593	578
28	439	476	602	725	509	787	494	798	401	593	748	647	556	625
29	445	754	635	622	643	624	439	620	534	488	706	655	492	743
30	797	473	496	621	462	694	462	639	733	546	441	598	793	645
31	738	469	451	575	597	467	477	532	675	766	667	753	726	507
32	633	661	477	548	706	664	733	611	473	595	541	478	585	695
33	413	676	641	569	513	580	727	591	587	634	755	689	427	598
34	618	718	595	559	717	766	437	518	628	486	574	648	498	638
35	485	587	702	446	762	721	625	784	473	572	499	453	770	521
36	521	518	736	506	593	513	667	610	715	714	651	508	433	715
37	473	440	567	711	524	426	558	633	667	484	785	735	622	775
38	602	430	497	663	588	441	411	603	697	747	613	700	715	693
39	556	500	742	557	611	554	496	731	758	446	576	715	544	614
40	494	469	653	785	757	425	562	550	794	580	532	405	617	777
41	772	528	431	658	667	570	451	674	511	688	626	587	787	450
42	555	460	625	700	637	461	670	678	559	677	537	684	680	477
43	522	765	765	537	673	707	654	508	610	487	707	489	539	437
44	496	792	572	641	767	496	596	780	428	637	466	576	597	556
45	701	445	586	775	464	735	474	622	531	590	699	653	506	619
46	605	621	757	666	465	645	544	557	402	759	562	605	720	492

47	615	593	667	762	470	437	598	787	540	432	663	652	570	614
48	561	432	520	675	788	744	661	717	554	740	444	691	418	455
49	505	670	772	532	554	459	675	608	430	542	738	726	767	422
50	650	734	459	449	652	603	767	604	423	519	763	539	623	615

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