On synchrophasing control of vibration for a floating raft vibration isolation system

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ABSTRACT

A large-scale floating raft vibration isolation system installed on a flexible hull-like structure is described in this paper. Four vibration exciters, in which two counter rotating shafts with the same balanced masses are driven by a phase asynchronous motor, are used to simulate rotating machines on the raft. The phase of the electrical supply to the motors is adjusted by synchrophasing control scheme to reduce the vibration transmitted to the host structure to which the machinery raft is attached. This kind of synchrophasing control for the floating raft vibration isolation system is investigated theoretically and experimentally. At last some results are presented and discussed.

Keywords: Vibration isolation, Floating raft, Hull-like structure, Synchrophasing control I-INCE Classification of Subjects Number(s): 51.4

1. INTRODUCTION

Vibration isolation using passive isolators is widely used in marine applications through different configurations, such as single-stage, double-stage and floating raft isolation system. Active vibration isolation is available(1-9) for purchasing better isolation performance especially in low frequency range since good low frequency vibration isolation often cause weight cost or stability problem by using passive isolation. Active control is often achievable by introducing some kinds of secondary vibration sources, for example, hydraulic type, electromagnetic type or piezoelectric type actuators in series or parallel with passive resilient mounts to generate forced vibration responses aiming to cancel the primary vibration transmission. So active vibration isolation system will cost more, requires power supply and digital controllers, and still needs a back-up system of conventional passive isolators.

Synchrophasing control is another alternative strategy suitable for more machines on a common raft, which idea can tracing back a long time ago to control vibration by adjusting speed and phase angles of two engines in a steam ship(10).Simply in speaking, it is only need to make the two engines run at the same speed, but in anti-phase. Apart from two patents (11,12), this control scheme seems to have received very little attention in the literature for controlling the vibration of raft mounted machinery. For controlling sound in aircraft cabins(13-15) and duct(16,17), synchrophasing is a well-established technique, and recent research on synchrophasing in aircraft has concentrated on active synchrophasing, using microphones and accelerometers positioned throughout the aircraft, together with adaptive optimisation techniques to minimise the cabin noise and vibration over a wider range of flight conditions (18,19). The Propeller Signature Theory (PST) (20) is employed to find out the optimum solution in all the possible synchrophase angle combinations.

In recent research by Brennan (21) the principle of synchrophasing is described and its application to multiple machines on a raft in marine application is discussed. An experimental study together with

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some simulations on a one-dimensional structure is used as the machinery raft are also presented to demonstrate the efficacy of synchrophasing in the control of vibrations.

This paper is an extension of the work in reference (21). The application of synchrophasing control to multiple machines on a large scale floating raft system is investigated theoretically and experimentally. Four vibration exciters, in which two counter rotating shafts with the same balanced masses are driven by a phase asynchronous motor, are used to simulate rotating machines on the raft. The phase of the electrical supply to the motors is adjusted by synchrophasing control scheme to reduce the vibration transmitted to the host structure on which the machinery raft is attached through some passive isolators.

Following this introduction, the principle of synchrophasing control is described in section 2. In section 3, along with details of the test rig and synchrophasing control system, a description of the synchrophasing experiment is given, and some conclusions are drawn in section 4.

2. PROBLEM FORMULATION

2.1 The Cost Function

A schematic presentation of a typical machinery installation is shown in Figure 1, in which a number of synchronous vibrating sources (assumed to be rigid masses) attached to a machinery raft. The raft is then isolated from a flexible supporting structure by a number of discrete isolators. A suitable cost function, which captures the vibration to be controlled from a machinery raft, is the sum of the squares of the modulus of the accelerations at the isolator positions on the flexible hull-like structure. It is given by

$$J = \mathbf{w}_m^{\mathrm{H}} \mathbf{w}_m \tag{1}$$

in which, $\mathbf{w}_m$ is the m length vector of accelerations on the hull-like structure at the m isolator positions and the superscript H denotes the Hermitian transpose. This vector of accelerations, can be expressed as

$$\mathbf{w}_m = \mathbf{Y}_{mn} \mathbf{f}_n \tag{2}$$

where $\mathbf{f}_n$ is the n-length vector of forces generated by the machines on the raft and $\mathbf{Y}_{mn}$ is the acceleration mobility matrix between n machines to m isolators positions on the hull-like structure.

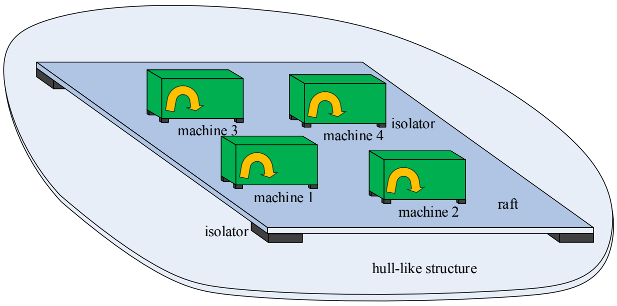

Figure 1 – A schematic presentation of a typical floating raft vibration isolation system

The acceleration mobility matrix $\mathbf{Y}_{mn}$ can be written in terms of m-length vectors, so that

$$\mathbf{Y}_{mn} = \begin{bmatrix} \mathbf{y}_1 & \mathbf{y}_2 & \cdots & \mathbf{y}_n \end{bmatrix} \tag{3}$$

in which, $\mathbf{y}_i$ is the transfer function vector from the i th machine to m isolators positions on the hull-like structure.

Moreover the force vector can be written in terms of the magnitudes of the forces generated by the machines, and the phase φ_i of those machines

$$\mathbf{f}_n = \begin{bmatrix} A_1 e^{j\varphi_1} & A_2 e^{j\varphi_2} & \cdots & A_n e^{j\varphi_n} \end{bmatrix}^{\mathrm{T}} \tag{4}$$

Combining Eqs. (1), (3) and (4) gives

$$J=\begin{bmatrix} A_1e^{-j\varphi_1} \\ A_2e^{-j\varphi_2} \\ \vdots \\ A_ne^{-j\varphi_n} \end{bmatrix}^{\mathrm{T}} \begin{bmatrix} \|\mathbf{y}_1\|^2 & \mathbf{y}_1^*\mathbf{y}_2 & \vdots & \mathbf{y}_1^*\mathbf{y}_n \\ \mathbf{y}_2^*\mathbf{y}_1 & \|\mathbf{y}_2\|^2 & \vdots & \mathbf{y}_2^*\mathbf{y}_n \\ \vdots & \vdots & \ddots & \vdots \\ \mathbf{y}_n^*\mathbf{y}_1 & \mathbf{y}_n^*\mathbf{y}_1 & \vdots & \|\mathbf{y}_n\|^2 \end{bmatrix} \begin{bmatrix} A_1e^{j\varphi_1} \\ A_2e^{j\varphi_2} \\ \vdots \\ A_ne^{j\varphi_n} \end{bmatrix} \tag{5}$$

Machine 1 is taken as the reference machine and to adjust the relative phases of other machines with respect to the reference one, Eq.(5) can be written as the sum of three parts, so that

$$J=J_A+J_B+J_C \tag{6}$$

in which

$$J_A=\sum_{i=1}^{n}\|A_i\mathbf{y}_i\|^2 \tag{7a}$$

$$\begin{aligned} J_B &= 2\sum_{i=2}^{n}A_iA_1\,\mathrm{Re}\left\{\mathbf{y}_1^*\mathbf{y}_ie^{j(\varphi_i-\varphi_1)}\right\} \\ &= 2\sum_{i=2}^{n}A_iA_1R_{i1}\cos(\theta_{i1}+(\varphi_i-\varphi_1)) \end{aligned} \tag{7b}$$

$$\begin{aligned} J_C &= 2\sum_{i=2}^{n}\sum_{j=3}^{n}A_iA_j\,\mathrm{Re}\left\{\mathbf{y}_i^*\mathbf{y}_je^{j(\varphi_j-\varphi_i)}\right\}\Big|_{i\neq j} \\ &= 2\sum_{i=2}^{n}\sum_{j=3}^{n}A_iA_jR_{ij}\cos(\theta_{ij}+(\varphi_j-\varphi_i))\Big|_{i\neq j} \end{aligned} \tag{7c}$$

where

$$R_{i1}=\sqrt{\mathrm{Re}^2\{\mathbf{y}_1^*\mathbf{y}_i\}+\mathrm{Im}^2\{\mathbf{y}_1^*\mathbf{y}_i\}} \tag{7d}$$

$$R_{ij}=\sqrt{\mathrm{Re}^2\{\mathbf{y}_i^*\mathbf{y}_j\}+\mathrm{Im}^2\{\mathbf{y}_i^*\mathbf{y}_j\}} \tag{7e}$$

$$\theta_{i1}=\arccos\left\{\frac{\mathrm{Re}\{\mathbf{y}_1^*\mathbf{y}_i\}}{\sqrt{\mathrm{Re}^2\{\mathbf{y}_1^*\mathbf{y}_i\}+\mathrm{Im}^2\{\mathbf{y}_1^*\mathbf{y}_i\}}}\right\} \tag{7f}$$

$$\theta_{ij}=\arccos\left\{\frac{\mathrm{Re}\{\mathbf{y}_i^*\mathbf{y}_j\}}{\sqrt{\mathrm{Re}^2\{\mathbf{y}_i^*\mathbf{y}_j\}+\mathrm{Im}^2\{\mathbf{y}_i^*\mathbf{y}_j\}}}\right\} \tag{7g}$$

Equation (7a) contains no phase information and thus cannot be changed by adjusting the phase of any of the machines. The part of the cost function in Eq. (7b) is related directly to the relative phases between the control machines and that of the reference machine. Equation (7c) is only included in the cost function when there are three machines or more (two or more control machines), as this term is due to the interaction between the control machines.

The phase of the reference machine is set to be zero, $\varphi_1=0$, the phase of the adjusting machine or control machine is shown as Figure 2.

When there are only two machines, there is only one control machine and the cost function is given by

$$J=J_A+J_B=A_1^2\|\mathbf{y}_1\|^2+A_2^2\|\mathbf{y}_2\|^2+2A_1A_2R_{21}\cos(\theta_{21}+\varphi_2) \tag{8}$$

Hence, to minimize J requires that $\cos(\theta_{21}-\varphi_2)=-1$, implying that the phase angle which minimizes the cost function is given by

$$\varphi_{2,\min} = (2k+1)\pi - \theta_{21}, \quad k = 0,1,2\cdots \tag{9}$$

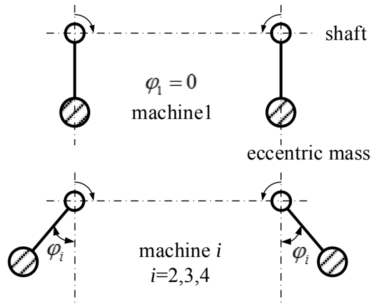

Figure 2 - The phase angles of the control machine and reference machine

So, if θ_{21} is known, $\varphi_{2,\min}$ can be calculated. It can also be seen that the phase angle to maximize the response is given by

$$\varphi_{2,\max} = 2k\pi - \theta_{21}, \quad k = 0,1,2\cdots \tag{10}$$

Equation (7f) shows that θ_{21} only depends on the transfer functions from the two machines to the isolator positions on the flexible hull-like structure. So the phase angles to minimize and maximize the responses have nothing to do with the magnitudes of the forces but depend on the locations and frequency (rotating speed) of exciting forces. But the equation below is obviously

$$\varphi_{2,\min} = \varphi_{2,\max} + \pi \tag{11}$$

When there are three or more than three machines, the cost function contains three parts, J_A, J_B and J_C. The phase angles to minimize and maximize the responses depend on the force locations, frequency and force amplitude simultaneously.

2.2 Determination of the Optimum Phase Angles

The simplest search strategy is that of stepping through the phase of each control source from 0 to 360° in suitable steps for each operating frequency of interest. The number of measurements required at each frequency is $(360/p)^{n-1}$ for a p° step size and n machines. This type of searching can be very time consuming when the machine number increases. In this paper, on the basis of the alternative PST, which has been adopted in aircraft propeller noise and vibration studies, a genetic algorithm is adopted to search the optimum phase angles. It is based upon measuring the frequency response function between each machine on the machinery raft and each control sensor, which is located at the isolator position on the hull-like structure and then using this data to determine the effect of changing the phase angles on the value of the cost function.

Force vector $\mathbf{f}_n$ in Eq.(4) can be expressed in matrix term

$$\mathbf{f}_n = \mathbf{A}\boldsymbol{\phi} \tag{12}$$

in which, $\mathbf{A}$ is a $n \times n$ diagonal matrix whose elements are amplitudes of the forces generated by the n machines, $\boldsymbol{\phi}$ is a n-length vector of phases of the n machines on the raft

$$\mathbf{A} = \begin{bmatrix} A_1 & 0 & \cdots & 0 \\ 0 & A_2 & \cdots & 0 \\ \vdots & \vdots & \ddots & \vdots \\ 0 & 0 & \cdots & A_m \end{bmatrix}, \quad \boldsymbol{\phi} = \begin{bmatrix} e^{j\varphi_1} \\ e^{j\varphi_2} \\ \vdots \\ e^{j\varphi_n} \end{bmatrix}$$

Substituting them into the cost function gives

$$J = \boldsymbol{\phi}^{\mathrm{H}}\boldsymbol{\Psi}^{\mathrm{H}}\boldsymbol{\Psi}\boldsymbol{\phi} \tag{13}$$

where $\mathbf{\Psi} = \mathbf{Y}_{mn}\mathbf{A}$. For a specific case, $\mathbf{\Psi}$ is determined and $\boldsymbol{\phi}$ will determine the amount of J . The simplest way of determining the coefficients in the frequency response function matrix $\mathbf{\Psi}$ at the operating frequency is to switch on all the machines at a certain phase combination, and then measure output of each control sensor. After n measurements, the columns of $\mathbf{W}_{mn}$ can be determined. This is given by

$$\mathbf{W}_{mn} = \mathbf{\Psi\Phi} \tag{14}$$

in which

$$\mathbf{\Phi} = \begin{bmatrix} 1 & 1 & \cdots & 1 \\ e^{j\varphi_{21}} & e^{j\varphi_{22}} & \cdots & e^{j\varphi_{2n}} \\ \vdots & \vdots & \ddots & \vdots \\ e^{j\varphi_{n1}} & e^{j\varphi_{n2}} & \cdots & e^{j\varphi_{nn}} \end{bmatrix}$$

where φ_{ij} is the phase angle appointed at the jth measurement for ith machine. The phase of the reference machine is assumed to be 0, $\varphi_{1j} = 0$, implying that $e^{j\varphi_{1j}} = 1$, so

$$\mathbf{\Psi} = \mathbf{W}_{mn}\mathbf{\Phi}^{\mathrm{T}}\left[\mathbf{\Phi\Phi}^{\mathrm{T}}\right]^{-1}$$

Then the optimum phase angle vector $\left[0, \varphi_2 \cdots \varphi_n\right]_{\min}$ for J can be determined by genetic algorithm through Eq.(13).

3. EXPERIMENT WORK

3.1 Test Rig

A photograph of the experimental test rig is shown in Figure 3. A 6.2m×5.2m ×0.4m floating raft is constructed by 16 I-shaped steel beams, which are bolted together. It is supported on a flexible hull-like structure by 18 BE-400 type resilient isolators mounted on 18 steel pillars. The 7m×6m×0.7m flexible hull-like structure is made from steel plates which are welded together and supported by 26 pneumatic isolators. Four vibration exciters, in which two counter rotating shafts with the same balanced masses are driven by a phase asynchronous motor, simulate ship machinery to excite the raft. One of these kind of machines can be seen in the lower right corner of Figure 3.

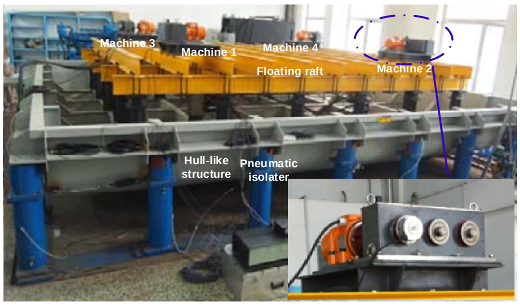

Figure 3 - The floating raft vibration isolation system on a hull structure

Modal analysis and test were carried out to determine the modal characteristics of the system (22). The first four modes were found to be rigid-body modes with natural frequencies of 1.7 Hz - heave, 2.1 Hz - pitch, 3.4 Hz - roll and 5.1 Hz - torsion. The 5th mode at 6.0 Hz is a warping mode. The 6th mode at 20.3 Hz is a beam-like bending mode of the floating raft, and the 7th mode at 22.1 Hz is a bending mode similar to that of the first mode of a simply-supported plate.

A block diagram of the experimental set-up is shown in Figure 4. One of the machines (M1) was appointed to be the reference machine and the other three machines (M2, M3 and M4) were control machines, for which the phase of the supply voltage could be adjusted. 22 acceleration sensors were located at 18 pillars and 4 additional positions on the hull-like structure, which outputs were used to calculate the cost function.

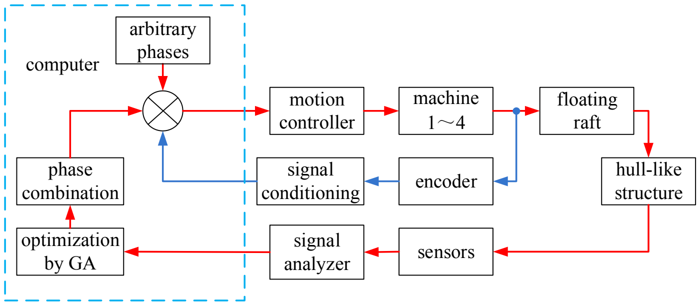

Figure 4 - Block diagram of the synchrophasing control system

The synchrophasing control procedure can be described as follows:

a) *n* arbitrary phase combinations are given firstly to the motion controller which drive the motors of the machines and cause vibration of the floating system;

b) The vibrations on the hull-like structure are measured and the acceleration measurement matrix $\mathbf{W}_{mn}$ comes into being, then $\mathbf{\Psi}$ can be determined;

c) Then the optimum phase combination is found by genetic algorithm through Eq.(13) and fed back to the motion controller.

3.2 Results and Discussion

The experimental cases of 2, 3 and 4 machines working together were conducted. *n*+1 data samples were measured to search the optimum phase angle combination, where *n* is the machine number. The results are shown in Figures 5-8.

Figure 5 shows that machine 1 and machine 4 run at 1800r/min and 2400r/min, the exciting force ratios, which can be achieved by adjusting the rotation radius of the eccentric masses, are 1:1, 1:0.35 and 1:0.42 respectively. From three measured data samples, the minimum and maximum phase combinations are found out by genetic algorithm. The cost function amounts of each case including the minimum and maximum ones, the three data samples are plotted as a bar graph.

It is shown from Figure 5-8 that there was a reduction of about 2.5dB-13.2dB in cost functions with minimum and maximum phase angles.

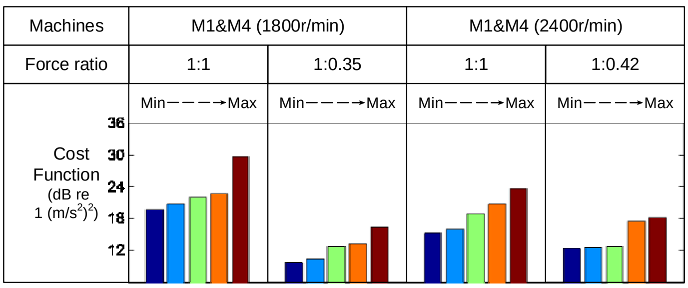

Figure 5 - Experimental results of synchrophasing with two machines of M1& M4

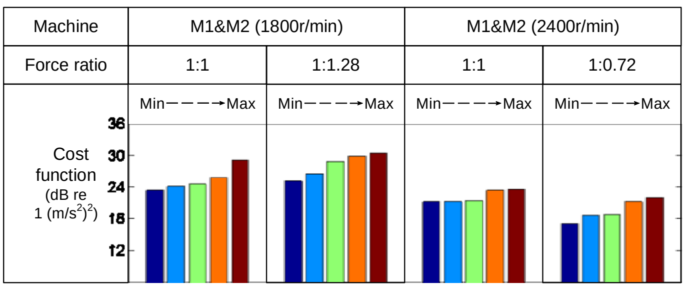

Figure 6 - Experimental results of synchrophasing with two machines of M1& M2

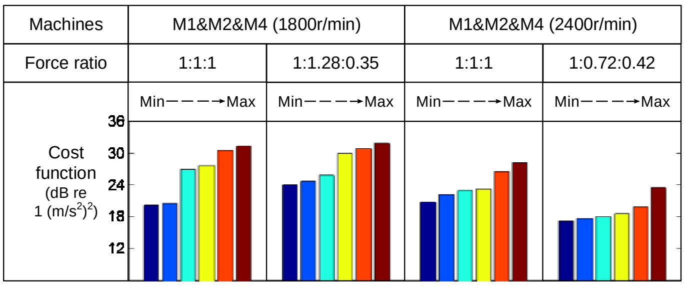

Figure 7 - Experimental results of synchrophasing with three machines of M1& M2 & M4

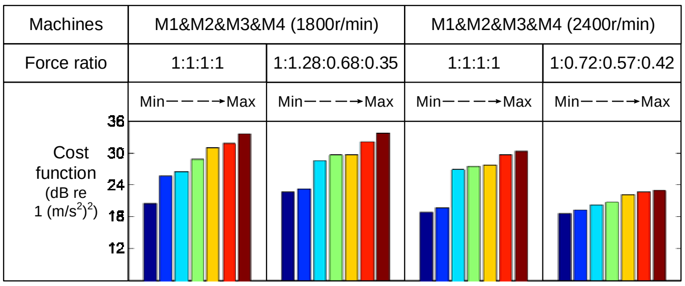

Figure 8 - Experimental results of synchrophasing with four machines of M1& M2 & M3 & M4

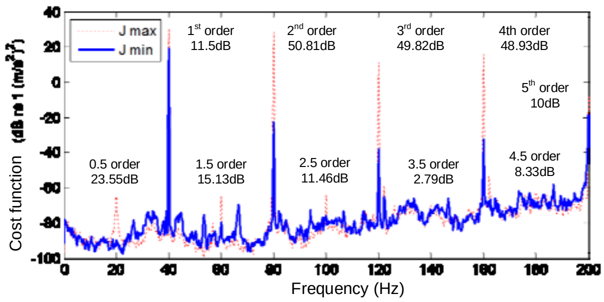

Figure 9 - The sum of the squared acceleration responses measured by 22 error sensors when maximize and minimize the cost function

Figure 9 shows the cost function in the range of 0-200Hz when four machines running at 2400 r/min and the forces ratio is 1:1:1:1. It can be seen that when the machines works, the forced vibration responses include not only the fundamental frequency 40Hz but also the harmonics component such as 80Hz, 120Hz and 160Hz. The cost functions at different frequencies can be decreased up to 51dB.

When there are two machines, there is only one control machine and the cost function is given by J_A and J_B. The phase angles to minimize and maximize the cost function are shown in Table 1. It is can be seen that the phase difference between these two cases are 180 degrees.

Table 1 - Phases which minimize and maximize the cost function with two machines

Case	Machine	Speed (r/min)	Force ratio	J_{min} φ_2	J_{max} φ_2
1	M1&M4	1800	1:1	333.9°	153.9°
	M1&M4	1800	1:0.35	340°	160°
2	M1&M4	2400	1:1	4.6°	184.6°
	M1&M4	2400	1:0.42	8.5°	188.5°
3	M1&M2	1800	1:1	160.1°	340.1°
	M1&M2	1800	1:1.28	165.8°	345.8°
4	M1&M2	2400	1:1	37.8°	217.8°
	M1&M2	2400	1:0.72	46.2°	226.2°

When there are three machines, the phase angles to minimize and maximize the cost function are shown in Table 2. There are no obviously characteristics between the minimum and maximum phases, since they depend on locations of machines, frequency and force ratios simultaneously.

Table 2 - Phases which minimize and maximize the cost function with three machines

Speed (r/min)	Force ratio	J_{min} φ_2	J_{min} φ_3	J_{max} φ_2	J_{max} φ_3
1800	1:1:1	178.7°	329.9°	339.9°	291.7°
1800	1:1.28:0.35	174.4°	297.7°	343.9°	239.9°
2400	1:1:1	128°	339°	258°	191°
2400	1:0.72:0.42	62°	68°	274.3°	204.7°

4. CONCLUSIONS

The phases of multiple rotating or reciprocating machines installed on a floating raft are adjusted by synchrophasing control to minimize vibration transmission from the floating raft to a hull-like structure. Relative phases between each machines are optimized in order to minimize the sum of squares of acceleration responses of the hull-like structure by using genetic algorithm. The results show that synchrophasing control can reduce the vibration transmission to the hull structure. Reductions of the sum of squares of acceleration responses measured by 22 error sensors on the hull-like structure are up to 13.2 dB totally and 51 dB at some harmonic frequency component by synchrophasing control.

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