Capabilities for Monitoring and Controlling Dynamic Parameters of Extended Structures Based on Distributed System of Integrating Sensors

S. Vassilyev¹, A. Galyaev², E. Yakushenko³, V. Zaletin⁴

¹Academician of Russian Academy of Sciences, Chief Researcher, Institute of Control Sciences of RAS, Moscow, Russia
²Corresponding member of Russian Academy of Sciences, Chief Researcher, Institute of Control Sciences of RAS, Moscow
³Academician of Russian Academy of Sciences, JSC "NPO Special Materials", Saint Petersburg
⁴Senior Researcher, RPC "Vacuum developments", Saint Petersburg

vassilyev_sn@mail.ru

Abstract. A range of issues and tasks related to the construction of a measurement system based on integrating fiber-optic sensors for monitoring and controlling low-frequency acoustic vibrations of ship hull structures is considered.

1. Introduction

It is impossible to control the dynamic parameters of extended structures without knowing the characteristics of their spatial and temporal distribution. At the same time, monitoring the dynamic parameters of extended structures, such as aircraft hulls, ships, bridges, etc., with an a priori unknown amplitude-phase distribution of the levels of normal surface displacement, has always been a difficult task. Solving the problem of restoring a two-dimensional vibration field requires installing a sensor system in the grid nodes, the step of which does not exceed the smallest spatial scale of the field. As a result, in order to control the entire surface area of such a structure, it is necessary to use thousands (tens of thousands) of accelerometers, pre-amplifiers, and other equipment, which makes it almost impossible to solve this problem. In this regard, the use of local sensors for these purposes seems impractical.

In addition to high requirements for metrological characteristics, sensors must have high reliability, durability, stability, small dimensions, weight and power consumption, compatibility with electronic information processing devices, and have a low cost.

The integrating fiber-optic sensors (IFOS) meet these requirements to the maximum degree [1-3]. For example, to control the dynamic parameters of the hull of a TI Europe type supertanker 380 meters long and 68 meters wide, a system consisting of only 250 IFOS is needed to divide the surface of the ship's hull into elementary sections with an area of 1 square meter (Fig. 1).

Vibration of the extended structure body occurs due to the impact of the energy of vibro-active equipment on it. In this case, acoustic energy from the source of excitation (for example, a diesel generator) is distributed through the ship's structures, which in this case are transmission elements, and affects the ship's hull, causing it to vibrate (Fig. 2).
Figure 1. Tanker model with a hosted IFOS system.

Figure 2. Enlarged scheme of acoustic energy transmission.

Moreover, knowing the vibration levels on the legs of the mechanism, or the level of air noise in the compartment, or the vibration level of the pipeline, etc., it is almost impossible to restore the vibration field of the body surface (vibrating element) due to complex processes of acoustic energy propagation through the body structures (transmission elements).

Analysis of the scheme of exchange of vibrational energy between transmission and vibrating elements shows that all elements are connected to each other. The coupling coefficients between elements at different points have different frequency characteristics for each type of elastic wave and for different transfer elements. It is important that it is not possible to specify in advance which path of vibrational energy transfer is the most important, and which paths of energy transfer can be considered secondary and can be ignored in the first approximation.

The solution of the problem of elastic wave propagation over hull structures has long attracted the attention of researchers [4-6]. Analysis of the results of exact solutions applied to real hull structures shows that they can only be used to describe the propagation of elastic waves in a qualitative way. It is almost impossible to take into account all the structural inhomogeneities of buildings of extended structures, and the results of accurate solutions are extremely sensitive to the degree to which these inhomogeneities are taken into account.

2. Problem of controlling and monitoring of dynamic parameters of extended structures

In addition, bending waves excited in the area of the source location, when passing through the inhomogeneities of the hull structures, are converted into longitudinal and shear waves, which pass almost freely through typical structural inhomogeneities and, weakly damping, propagate to the sections of the hull structures remote from the vibration source. In these sections, the longitudinal and shear waves are again transformed into bending waves and form an induced field of bending waves, which may exceed the value of the levels of bending waves that reached the same sections without
transformation. At the same time, there is a noticeable decrease in the rate of attenuation of elastic wave energy as they propagate along the hull structures.

When a bending or longitudinal wave falls on the stiffener, part of the energy will be reflected in the form of a bending (traveling and inhomogeneous) and a longitudinal wave, and part will pass through the edge in the form of the same waves. Standing bending and longitudinal waves are excited in the stiffening edge (Fig. 3).

Fig. 3. Scheme of consideration of the problem of the passage of bending waves through the stiffener.

The propagation of elastic waves through a connected two-layer structure supported by stiffeners is characterized by the following possible channels for the propagation of elastic wave energy:

- bending waves in the inner hull;
- longitudinal waves in the inner hull;
- shear waves in the inner hull;
- bending waves in the outer hull;
- longitudinal waves in the outer hull;
- shear waves in the external enclosure;
- acoustic waves in the inter-hull space;
- acoustic waves outside the external enclosure;
- bending waves in the connections between hulls;
- longitudinal waves in the connections between hulls;
- shear waves in the connections between hulls.
As can be seen from the above list, the energy of elastic waves can be transmitted through 11 channels. All channels are interconnected and the propagation of elastic waves depends on the propagation conditions in each channel and the coupling coefficients between them.

The results of experimental studies performed under field conditions show [7] that the correspondence between the calculation and the experiment is observed only for sections of structures located in close proximity to the vibration source. At the same time, there is a significant difference between the calculation results and the experimental data for the sections of the hull that are remote from the vibration source. This difference reaches 10 dB or more (Fig. 4).

![Graph showing vibration levels across the hull](image)

Fig. 4. Distribution of vibration levels across the hull.

The reason for the discrepancy between the experiment and the calculated vibration levels is the lack of consideration for the transformation of wave types on inhomogeneities of hull structures. When several mechanisms are working simultaneously and taking into account the natural vibrations of the shells, the difference between the actual vibration levels and the calculated ones is even more increased. Thus, it is almost impossible to build a vibroacoustic portrait of the hull with sufficient reliability based on the results of measurements of vibration levels on the legs of mechanisms, pipeline reference points, air noise in compartments, etc. Practical measurements of the dynamic parameters of the external housing surface are required.

The use of a system of integrating fiber-optic sensors provides a simple and effective solution to this problem. Moreover, the monitoring of the entire surface of the body by IFOS system makes it possible to control its vibration parameters due to realization of specified effects on the transmission elements on the distribution paths of acoustic energy.

At the same time, in the enlarged scheme of acoustic energy transmission through the hull structures, we get feedback (Fig. 5), which provides the necessary resource for controlling the vibration field of the external body of an extended structure.

![Feedback diagram](image)

Fig. 5. Enlarged feedback diagram.
Moreover, the control of acoustic energy propagating through ship structures is possible in various ways: by changing the stiffness of pneumatic shock absorbers, using vibro-active devices, phase shift in "controlled sound bridges", etc.

To reduce the current level of acoustic noise, passive and active vibration damping systems are used. It is known that at low frequencies the main energy is carried by a plane wave, while at high frequencies the wave front has a more complex form. Therefore, to reduce the level of the acoustic signal at high frequencies, passive means of protection are usually used, while at low frequencies the most effective are active means of damping, which use additional sources of vibrational energy to generate compensating radiation.

Active noise reduction is a method of reducing primary noise by actively suppressing it with secondary drives. These methods are of interest for reducing low-frequency noise where passive methods are not applicable or do not work effectively. One important potential application of active noise reduction is to control acoustic radiation in confined spaces, such as cavities located in the ship's hull. The predominant component of internal noise is caused by disturbances acting on elastic structures surrounding the cavity. The noise generated depends on the characteristics of the disturbances acting on it, as well as the characteristics of the structural and acoustic system of the resonator itself. In particular, low-frequency noise has a negative impact on the physical condition of the ship's crew. Therefore, ideally, the goal of acoustic radiation control is to reduce the noise level not at a single frequency, but in the entire frequency range. However, when solving this problem, it is necessary to take into account that signals at different frequencies during propagation in the water environment fade out differently. Therefore, the solution of the problem of acoustic noise suppression and compensation can be directly related to the detection criteria.

The problem of active regulation of internal noise in structural-acoustic cavities can be solved by direct suppression of primary noise using acoustic actuators [8-10], for example, such as a phase shifter for active noise cancellation systems. The solution of the active quenching problem is based on the Huygens principle.

Feedback control is one of the strategies for control active noise reduction. This strategy can be useful when the noise is broadband and there is no suitable reference signal. Direct output feedback is probably the simplest feedback control strategy, but more complex control methods based on optimal control using a linear quadratic regulator (LQR) or linear quadratic Gaussian (LQG) control strategy can also be applied. The purpose of the control is to eliminate not only unwanted noise, but also to prevent the appearance of an audio signal from the active noise reduction system. One of the key problems in the practical implementation of active noise reduction is the use of LQG controllers, for which accurate state observers must be developed. State observers based on an analytical or numerical model may be inaccurate due to the uncertainty of the models associated with them, but to solve the identification problem, the frequency range and noise in the experimental measurement mode must be limited.

3. Conclusion
Putting together all of the above, the development and creation of a low-frequency measurement system for a ship based on the IFOS system entails a comprehensive solution of wave dynamics problems for determining noise sources, identification problems for creating a control system for an active noise reduction system, as well as joint consideration and solution of problems for controlling the ship's modes according to the criteria for its detection at these frequencies.

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