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SIMULATION ABOUT THE VIBRATION CHARACTERISTIC OF A SMALL AUTONOMOUS UNDERWATER VEHICLE BASED ON TRANSFER FUNCTION METHOD

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Abstract

The use of an autonomous underwater vehicle (AUV) has been increasing in scientific, military, and commercial areas due to its autonomous operation without human control for a hazardous mission. Since it is impossible to use the devices with electromagnetic waves for gathering information in an underwater environment, only sonar systems that use sound waves can be used in underwater environments. Since the thruster system is widely used as the propulsion system of an AUV and is mounted on the outside of an AUV’s stern, the thruster system can generate vibration and that vibration can be transferred throughout the shell of the AUV from the stern of AUV to its bow. The transferred vibration can affect the performance of various sonar systems that equip the AUV such as side-scan sonar or forward-looking sonar. Therefore, it is necessary to predict the effect of the transferred vibration of the AUV on the sonar systems. Even if various numerical methods were used to analyze the vibration problem of an AUV, it would be hard to predict the vibration phenomena of an AUV at the initial design stage. In this work, a simulation study was carried out to analyze a vibration feature of a small real AUV in the air. The transfer functions of vibration acceleration based on the simulation results are presented.

Key words: Autonomous underwater vehicle, Frequency response function, Numerical simulation, Transfer function, Underwater radiated noise

Introduction

Autonomous underwater vehicles (AUVs) are increasingly being used in scientific, military, and commercial applications because they can be utilized instead of humans in hazardous circumstances such as mine countermeasure missions (MCM). AUVs require a high level of autonomy in order to gather information about the environment without human intervention. As it is impossible to employ electromagnetic waves for gathering information underwater, only sonar systems, which utilize sound waves, can be used. Self-radiated noise can affect the performance of sonar systems that equip an AUV. Therefore, it is necessary to reduce this noise for the high-level performance of AUVs. In general, a significant amount of self-radiated noise is generated by the propulsion system, because excitation of the hull, which is transmitted from the propulsion system through the shaft and water, is a significant source of radiated noise. It is essential to identify the mechanism of a self-radiated noise phenomenon, including its source and propagating path, in order to reduce it. As it is difficult to reduce the self-radiated noise of an AUV after its construction, it is important to estimate and consider this noise during the design stage. For this reason, the estimation method is the main research topic for reduction of self-radiated noise in AUVs. Some studies estimate the self-radiated noise from a fixed vibration source using numerical tools such as the finite element method (FEM) and boundary element method (BEM) for a surface ship. Askari and Daneshmand (2009) proposed a finite element method using the Galerkin method to analyze the coupled vibration of a partially-fluid-filled cylindrical container with a cylindrical internal body. Sigrist and Garreau (2007) used the finite element method to carry out coupled fluid-structure dynamic analysis with pressure-based formulation, using the modal and spectral method. Ugurlu and Ergin (2008) investigated the effects of different end conditions on the response behavior of thin circular-cylindrical-shell structures fully in contact with flowing fluid, using the finite-element and boundary element methods. However, these numerical methods require long calculation times to analyze new cases when the conditions for self-radiated noise, such as vibration frequency or length of the structure, change. Given that an AUV is usually slender with a circular cross section and submerged in water, its self-radiated noise phenomena will differ from that of a surface ship. Thus, it is quite difficult to apply the numerical model for a surface ship to an underwater vehicle. To decrease the computational burden of the numerical model, partial validation must be performed through vibration experiments. One alternative to the numerical method for estimating self-radiated noise is the transfer function method. In this method, the transfer function is defined as the relationship between a single source or propagating path and the derived self-radiated noise level. It is thus possible to simplify the generating mechanism of self-radiated noise by separating the vibration source and propagating paths. The total self-radiated noise of the structure is then calculated by the summation of the transfer function over vibration sources and propagating paths. If there is little difference between two structures, their transfer functions should be similar. The transfer function method could then be effective for estimating the self-radiated noise of an underwater vehicle at the design stage. In this study, several simulations were performed to identify the vibration transfer function propagation for small AUV with a unit force, and several experiments were performed with an impact hammer. The transfer function of an underwater vehicle is defined based on the simulation and experiment results.

Theoretical Background

The transfer function about vibration is usually defined as a mathematical representation of the relationship between the source of vibration and output of a system. The transfer function is a function that represents the characteristic of the system as a ratio of the system response to the system input and also a function that displays the response when a unit impulse is applied to the system in the frequency domain. The frequency response function is the one of transfer function and describe the relationship between the source of vibration and output of a system in frequency domain. For experiment with impact hammer which attach an accelerometer at a particular point and excite the structure at another point with a force gauge instrumented hammer, the frequency response function can be described as a function of frequency the relationship between those two points on the structure Then by measuring the excitation force and the response acceleration.

The resulting equilibrium of such system for experiment with impact hammer can be represented by the following differential equation:

$$M\ddot{x}(t)+C\dot{x}(t)+Kx(t)=y(t) \tag{1}$$

where $x(t)$ is the output of system and $y(t)$ is the excitation force in time domain.

Through Laplace transformation, the second- order differential equation given in Eq. (1), can be represented by the following algebraic equation:

$$\{Ms^2+Cs+K\}X(s)=Y(s)\,. \tag{2}$$

The transfer function of the system can be denoted by

$$H(s)=\frac{Y(s)}{X(s)}\,. \tag{3}$$

In Eq. (3), $X(s)$ is the output of system and $Y(s)$ is the excitation force in frequency domain. Frequency response functions are most commonly used for single input and single output analysis, normally for the calculation of the H1(s) or H2(s) frequency response functions. The H1(s) frequency response function is used in situations where the output to the system is expected to be noisy when compared to the input and H2(s) frequency response function is used in situations where the input to the system is expected to be noisy when compared to the output. In this paper, H1 frequency response function will be used as the transfer function in frequency domain because the experiment with impact can be considered as the noisy output system. H1 frequency response function is described as follows.

$$H1(s)=\frac{S_{xy}(s)}{S_{xx}(s)} \tag{4}$$

where $S_{xy}(s)$ is the cross spectral density in the frequency domain of $x(t)$ and $y(t)$ and $S_{xx}(s)$ is the auto spectral density in the frequency domain of $x(t)$.

Although the frequency response function can provide the relationship between the excitation force and the vibration acceleration and is widely used to analyze the vibration feature of structure, it needs to analysis the transfer function of acceleration because the acoustic wave can be described as acceleration in mathematical. Thus, transfer function about vibration and acceleration can be efficient to effect of the vibration on the sonar system of AUV. In this paper, the transfer function which is described (5) was used to verify the effect of vibration on the sonar system.

$$TF=20\log\frac{a_f}{a_i}\,. \tag{5}$$

In Eq. (5) a_f is a measured acceleration for target point and a_i is a measured acceleration for vibration source.

Simulation for transfer function of AUV

Simulations were performed to obtain the vibration transfer function of AUV for a developed AUV. The AUV used for simulation was an OKPO 300; Fig. 1 shows the overview of an OKPO 300 which was developed and manufactured by Daewoo Shipbuilding and Marine Engineering (DSME) Co. Ltd. The detailed specifications of OKPO 300 are listed in Table 1. The simulation on the vibration analysis of an AUV was performed using PATRAN, which is the commercial numerical computing program for vibration analysis. Fig. 2 shows the numerical model about OKPO 300 for simulation. The assumption for simulation was that the OKPO would be located in aerial condition and that there were no devices in the OKPO 300. Since the thruster system combined with motor and propeller in a single structure is widely used as the propulsion system of the AUV, the excitation point was assumed at the stern of the AUV, which would be connected to the thruster system. The measuring point was assumed at the head of the AUV, where the forward-looking sonar or range sonar would be usually mounted. The detailed conditions of the simulation are summarized in Table 2.

Figure 1. Overview of OKPO 300

Figure 2. The simulation model of OKPO 300 for PATRAN

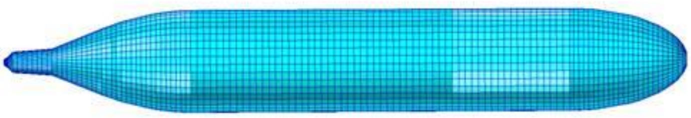


Table 1. Specifications of OKPO 300.

Items	Value
Dimesion (LxWxH)	1.80 m x 0.26 m x 0.26 m
Hull Material	Acetal
Weight	55.00 kg
Maximum Depth	300.00 m

Table 2. Simulation conditions.

Conditions	
Excitation points	Thruster
Input force	1 N
Acceleration measuring point	Center of the head nose
Environment	Air

Initially, we performed the mode analysis under free vibration condition to verify the vibration characteristic of the OKPO 300. The simulation results obtained under forced vibration conditions are depicted in Fig. 3 and Fig. 4. Fig. 3 describes the analysis results about the head nose position of OKPO 300 where the forward-looking sonar was installed and Fig. 4 illustrates the simulated acceleration results about the excitation point. From Figs. 3 and 4, it can be deduced that the simulated acceleration about the OKPO 300's head nose has a maximum value of approximately 200 Hz while the simulated acceleration about the excitation point has a maximum value of 800 Hz. The fact that there is a difference between the peak value of acceleration about the head nose and about the excitation point shows that the vibration mode of OKPO 300 could influence the vibration transmission strongly. Furthermore, since the low frequency of a passive sonar system for an OKPO 300 is usually between 100 Hz and 1000 Hz, the vibration from the stern of the AUV can affect the performance of the passive sonar system. Hence, it is necessary to analyze the effects of transferred vibration acceleration.

Figure 3. Simulation result of acceleration about the OKPO 300 head nose.

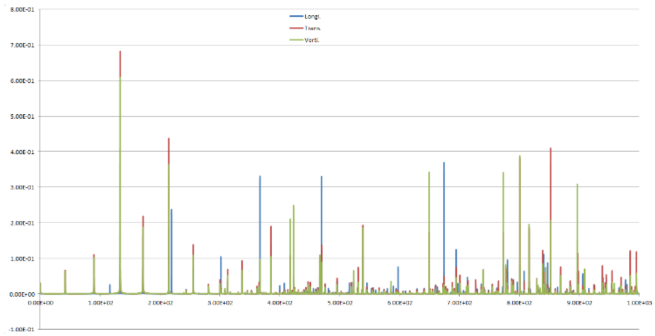
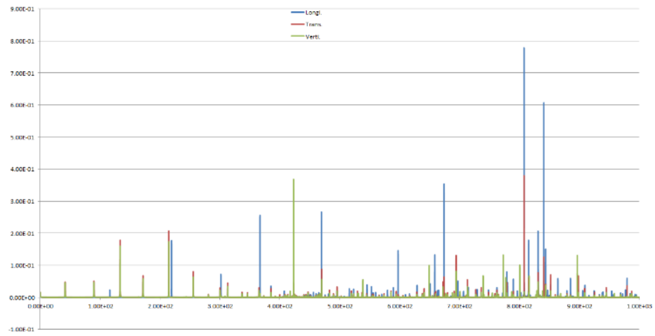


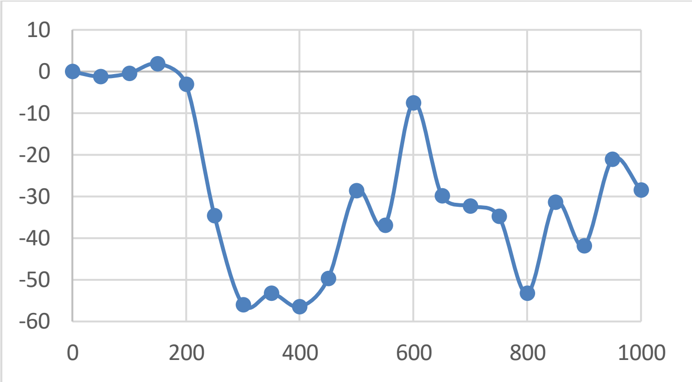
Figure 4. Simulation result of acceleration about the excitation point.



We calculated the transfer function in Eq. (5) to verify the effects of the vibration from the excitation point to the head nose of OKPO 300, which is equipped with the forward-looking sonar. The transfer function based on Eq. (5) is depicted in Fig. 5. From Fig. 5, we can observe that the vibrations with a frequency near 200 Hz transferred with the lowest reduction- approximately 0 dB and that the vibrations with a frequency near 800 Hz had the maximum reduction while transfer- approximately 60 dB.

These observations - the maximum value of transfer function near 200 Hz and the minimum value of transfer function at 800 Hz—helped determine the fact that even though acceleration at excitation point has a maximum value of almost 800 Hz, the vibration characteristic of OKPO 300, like vibration mode and natural frequency, has the most significant effect on the actual transmission of vibration.

Figure 5. The transfer function between the head nose and the excitation point as obtained from the simulation.



Conclusions

In this study, we performed several simulations to examine the vibration transfer function, which can describe the feature of vibration of AUV and analyzed the simulation results under unit force condition in view of the vibration acceleration transfer function. The results of the simulations showed that the first natural frequency of OKPO 300 was close to 100 Hz, the other mode frequencies were between 100 Hz and 400 Hz, which is usually a mid-frequency band for a passive sonar system of AUV, and the vibrations with a frequency close to 200 Hz transferred well. For future work, we need to predict the vibration acceleration transfer function under the submerged condition. The other future work will involve the experiment with real thruster as the vibration source for the calculation of real vibration transfer function of AUV.

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