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Technical note

Vibration damping of naval ships based on ship shock trials

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ABSTRACT

Presently, the vibration damping mechanism of naval craft subjected to underwater shock is not well understood, and ship shock trials remain the most effective means of researching ship vibration damping problems. A damping modeling strategy for naval ship systems is presented for ship shock transient time-domain analysis. The total vibration damping ratios of a naval vessel under different shock conditions are calculated using the logarithmic decay rate based on measured shock trial data. The least-squares complex exponential method is used for the extraction of modal parameters in the time domain. The obtained modal parameters were employed to construct synthesized time histories, which compared favorably to the original time histories, verifying the proposed method. The structural damping is quantified and evaluated based on the extracted modal frequency and modal damping ratio, and Rayleigh damping coefficients and are experimentally obtained. The values of and presented in this work are respectively less and greater than those proposed for the U.S. Navy ship DDG-53.

1. Introduction

When subjected to underwater explosions, the total structural vibration of the hull of a naval craft appears as damped free vibrations. Damping is one of the basic parameters employed to describe structural response clearly. Numerous studies have been conducted regarding the total vibration damping of ships [4–7]. The total vibration of a submerged light craft subjected to underwater explosion was analyzed theoretically by Stettler [4], and a numerical solution was also acquired. The calculated value was compared with that obtained from whipping experiments employing the U.S. Navy Red Snapper experimental platform. The numerical method was applied to calculate the total vibration damping of a 7400 ton nuclear attack submarine. The U.S. Navy also conducted a series of shock trails for DDG-53 and DDG-81 [5] from 1994 to 2001. The coefficient of the total damped vibration of DDG-53 for an underwater explosion was 4%. Shin et al. [6] analyzed experimental data associated with structural damping in the response of hull structures subjected to underwater explosion, and it was concluded that Rayleigh damping parameters are recommended for ship shock response prediction.

The present study evaluates the coefficient of the total damped vibration of a naval craft based on ship shock trails owing to the lack of research regarding the coefficients of total damped ship vibration. The least-squares complex exponential (LSCE) method [2] is used for the extraction of the modal parameters in the time domain. The obtained modal parameters were employed to construct synthesized time histories, which compared favorably to the original time histories, verifying that the time-domain extraction of modal parameters is feasible and valid. The structural damping of a ship is then quantified and evaluated based on the modal parameters. In addition, the Rayleigh damping coefficients are calculated based on ship shock trials, which are then compared with those obtained from ship shock trials conducted for the U.S. Navy ship DDG-53.

2. Calculation of the coefficient of total damped ship vibration based on the logarithmic decay rate

For calculation of the logarithmic decay rate, the peak strain values are obtained from the measurement data, and the decay rate is determined from the decay curves acquired by fitting to the peak strain values. The damping ratio (also known as the relative damping coefficient) is then calculated as follows [1]:

$$\xi = \frac{n}{\sqrt{(2\pi f)^2 + n^2}}, \tag{2-1}$$

where f is the frequency of the damped free vibration (Hz), which is the first eigenfrequency of hull vibration, and n is the decay rate obtained from fitting to the decay curves.

In an underwater explosion, the hull response exhibits three stages: the shock wave stage, bubble pulse stage, and damped free vibration stage. As shown Fig. 2.1, point A represents the time of the first bubble pulse, and point B represents the time of the second bubble pulse. The

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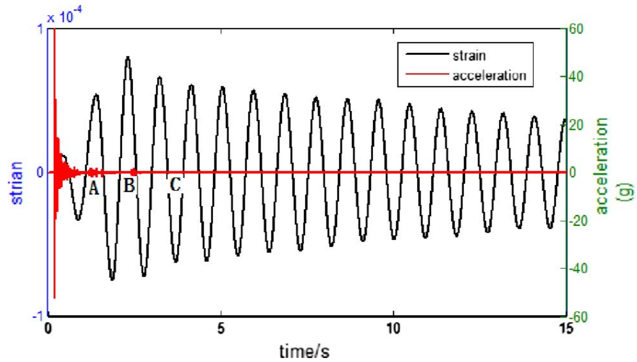

Fig. 2.1. Measured data (beginning at 0 s).

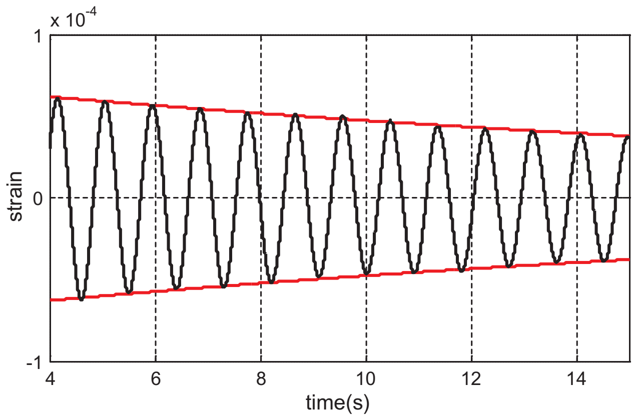

Fig. 2.2. Measured data (beginning at 4 s).

shock and bubble pulse stages occur prior to point C. The damped free vibration stage follows point C owing to the dissipation of energy, and it is therefore reasonable to calculate the value of ξ for a hull girder based on data obtained after point C (beginning at about 4 s).

Band pass filtering techniques were employed to acquire the decay curve of the damped free vibration, as shown Fig. 2.2, from which n and f are obtained. The value of ξ is then calculated from Eq. (2-1). All strains are calculated for each test signal. Then, in addition to ξ, the keel shock factor (KSF) [1] was calculated as follows:

$$KSF = \frac{\sqrt{W}}{R}\left(\frac{1+\cos\theta}{2}\right), \tag{2-2}$$

where W is the magnitude of the explosion in terms of the equivalent weight of TNT, R is the distance between the explosion source (TNT) and the keel, and θ is the angle between the explosion source (TNT) and the bottom surface of the hull. These parameters are illustrated in Fig. 2.3. The values of ξ and f for various KSF are listed in Table 2.1.

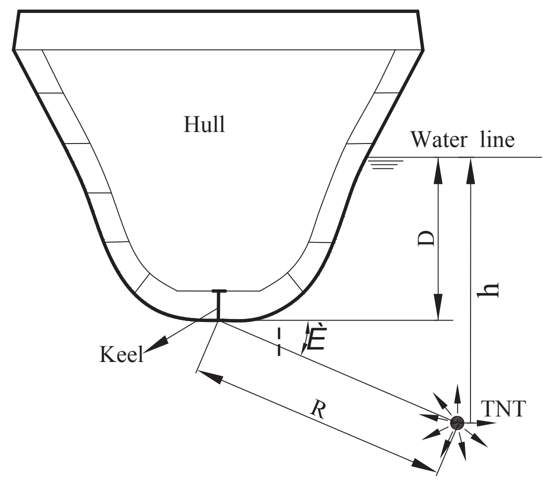

Fig. 2.3. Schematic diagram of explosion conditions.

Table 2.1
Damping ratios ξ and vibration frequencies f for some KSF values.

KSF	0.2	0.5	0.6
f (Hz)	1.05	1.1	1.15
ξ	0.006	0.0069	0.0072

3. Least-squares complex exponential (LSCE) method

The LSCE method [2] is one of the effective modal extraction methods employed in the time domain. However, the LSCE method cannot eliminate noise modes. Therefore, in the solution process [3], empirical mode decomposition (EMD) is firstly employed to decompose measured data X(t) as a function of time t into intrinsic mode functions (IMFs), and high-frequency IMFs representative of noise modes are eliminated to acquire data $\tilde{y}(t)$ without noise modes. The conversion factor Z what contains modal parameters to be identified is represented here as an impulse response. The Prony polynomial is established, and then the values of Z be equal to the zero point of the polynomial. Thus, the solution of Z is converted to a solution of polynomial coefficients. To solve the coefficients of the Prony polynomial, an autoregressive (AR) model whose coefficients are equal to the Prony coefficients of the polynomial model is created. Taking samples at different starting points allows us to obtain linear equations of the AR coefficients. Thus, the polynomial roots can be obtained using linear least-squares fitting to determine the AR coefficients. Finally, the modal parameters are extracted depending on the relationship between Z and parameters.

3.1. Elimination of noise modes in measured data

After the extrema of $X(t)$ are determined, the upper envelope curve E1 and the lower envelope curve E2 of the damped free vibration can be compared. Thus, the mean function m is given as follows:

$$m = (E1 + E2)/2, \tag{3-1}$$

and we can define the function

$$h_1 = X(t) - m. \tag{3-2}$$

The criteria of h_1 as an IMF are given as follows. (1) Either the number of extreme points and the number of zero points for h_1 are equivalent, or their D-value is 1. (2) The mean values of the upper and lower envelopes of h_1 are always zero. If h_1 fails to meet these criteria, then $X(t)$ is replaced with h_1 in Eq. (3-2), and the steps are repeated with the new h_1. When h_1 meets these criteria, it will be the first IMF of $X(t)$, such that

$$C_1 = h_1. \tag{3-3}$$

Separating C_1 from h_1 yields the following residual function:

$$r_1 = X(t) - C_1. \tag{3-4}$$

Then, $X(t)$ is replaced with r_1 in Eq. (3-2), and the above steps are repeated. After n iterations when r_n is monotonic, we obtain the following:

$$r_n = r_{n-1} - C_n. \tag{3-5}$$

So, $X(t)$ is given as

$$X(t) = \sum_{i=1}^{n} C_i(t) + r_n. \tag{3-6}$$

Once the critical order is determined, the high-frequency IMFs indicative of noise modes can be eliminated, yielding