Experimental Evaluation of Structural Hull Damping Coefficient of a Hydro-Elastic Segmented Submarine Model

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Abstract. This paper presents an experimental study of determining structural damping ratio of the hydro-elastic submarine model. The model test of a hydro-elastic submarine model was conducted at the Towing Tank facility of the Indonesian Hydrodynamic Laboratory (BTH-BPPT). To investigate the effect of the wet and dry damping ratio the vibration tests were performed in both air and water. The experiments in wet and dry condition were undertaken to investigate the influences of the damping ratio at zero forward speed. Vibration tests were also conducted in the Towing Tank at speed in calm water to explore the effect of the forward speed on both fixed and free pitch conditions on the damping ratio. From these experiments it has been found the damping ratios of wet condition are greater than the dry one. The damping ratios increased also as the submarine model speeds forward, whereas both the fixed and free pitch showed an increase in the damping ratio with increase forward speed.

1. Introduction
A submarine is not supposed to operate on the sea surface, however, when the submarine needs to rise to sea surface and sailing in it the static loads may change to dynamic one. The load can affect the response of the submarine structure when exposed to waves. The effects of ocean waves on the submarine hull can produce secondary loads or local loads. This phenomenon often occurs in high wave conditions, where at high waves it often makes the back of the submarine above the water and crashes back when hit by the next wave. This load can occur due to the submarine's structural response to waves when the submarine hits the waves. The effect of ocean waves on the hull of a submarine can produce a secondary load or local load called slamming. Slamming is the result of a transient response from the movement of the bow or stern of the ship when it is lifted or downed to the surface of the water. The effects of slamming loads on material fatigue damage. Slamming also affects electronic and mechanical equipment in submarine on operational conditions. These events also can cause high load and it is extremely dangerous.

This raises the need to have better understanding about dynamic structural loads on submarines, especially the whipping and slamming responses with concentrate to structural strength and fatigue life. One of the important aspects is to determine structural damping of the submarine hull in order to predict this structural response. Some references have been found to support the importance of the damping ratio to estimate the dynamic response of the ship. Ship Structure Committee, SSC - 359 (1991) made report that contains a research plan for ship vibration damping, including analytical calculations, model testing, and full-scale measurements. This study of damping ratio of hydro elastic
submarine model is inspired by work of the of Lafrov et al (2007) when they conducted a study of the effects of the criteria affect the whipping vibratory response of a high-speed catamaran to slamming were considered by using hydro-elastic model tests in calm water. Then Vorus (2010) presents simple continuous models representing idealization of real systems, are extremely valuable in understanding basic vibration concepts of the ship hull girder. Wibowo et al (2012) describe a method of determining the damping coefficient of damper made of cement in the vibration test laboratory. Tests were conducted by using an impact hammer and the plate response was measured by an accelerometer. Jialong Jiao et al (2015) had a study of hydro elastic response of an ultra large ship with comparative verification between experimental and numerical methods in order to estimate the wave loads response considering hull vibration and water impact. A segmented self-propulsion model ship with a steel backbone system is designed and tested in the tank test. The study of using a submarine hydro elastic model has also conducted by Wibowo et al (2016) to to predict slamming that occurs in the pressure hull through a combination of numerical analysis and hydro-elastic submarine model tests. In the same year Wibowo et al (2016) also carried out an experimental study for monitor the health of submarine hull structures using strain gage sensors and wireless communication technology.

This paper presents an experimental study of determining structural damping ratio of the hydro-elastic submarine model. The model test of a hydro-elastic submarine model was conducted at the Towing Tank facility of the Indonesian Hydrodynamic Laboratory (BTH-BPPT). To investigate the effect of the wet and dry damping ratio the vibration tests were performed in both air and water. The experiment in wet condition and dry condition were undertaken to investigate the influences of the damping ratio at zero forward speed. Vibration tests were also conducted in the Towing Tank at speed in calm water to explore the effect of the forward speed on both fixed and free pitch conditions on the damping ratio. The results of the hydro elastic segmented model may provide a base for further research in the future on the analysis of localized slamming and global wave loads.

2. Theoretical basis

Since SHIP STRUCTURE COMMITTEE [4] states that for a multi degree motion system the coefficient of damping appears to be almost constant over a wide range of frequencies. Then Lavrov et al [2] showed also that the damping ratio of the ship model flexural response remained relatively unaffected by the segmented ship model criteria variation. Hence, the damping ratio of the response can be determined in terms of the overall equivalent viscous damping ratio as written in Meriem [3]. To obtain the value of the damping ratio (ζ) it can only find through experiments. The experiment conducted by exciting the plate specimen with an impact load through an impact hammer. To understand the rationale of this experiment, this section will explain the theory of damped free vibration. In this system the damper causes the vibration of the system to decay when the energy of the system vibration is gradually dissipated by friction and other resistances. The motion gradually reduces or changes in frequency or intensity or cease and the system rests in its equilibrium position. Harmonic motion of light structures moves with almost the same frequency, but there are differences in amplitude and speed and energy over time. The time scale in which the amplitude decreases is related to the time constant. As the resistive force increases or the damping force increases, the amplitude travel time decreases. If the system gradually increased the amount of damping, the period and frequency begin to be affected, because damping opposes and hence slows the back and forth motion. If there is very large damping, the system does not even oscillate and it slowly moves toward equilibrium. The damper is usually displayed with a cylinder filled with a viscous fluid in which the liquid will move from one side of the cylinder to the other. Thus, the damper is called a viscous damper. The free body diagram for this spring – damper – mass system is shown in Figure 1.
The mathematical equation of the free body diagram above can be written as:

\[-kx - cx = mx\ddot{x}\]

or

\[m\ddot{x} + cx + kx = 0\]  \hspace{1cm} (1)

If \(\omega_n = \sqrt{\frac{k}{m}}\) is the circular natural frequency of the system and \(\zeta = \frac{c}{2m\omega_n}\) is the viscous damping factor or damping ratio. So the above equation can be written as:

\[\ddot{x} + 2\zeta\omega_n\dot{x} + \omega_n^2x = 0\]  \hspace{1cm} (2)

or

\[x = Ce^{-\zeta\omega nt} \sin(\omega_dt + \psi)\]  \hspace{1cm} (3)

where the damping circular frequency is \(\omega_d = \omega_n\sqrt{1 - \zeta^2}\) and the damping period is \(T_d = \frac{2\pi}{\omega_d}\). Constants \(C\) and \(\psi\) can be obtained by specifying \(x\) and \(\dot{x}\) at \(t = 0\) which is \(x_0\) and \(\dot{x}_0\). Seen in Figure 2 this equation describes the exponentially decaying harmonic function. This equation can be used to have the damping ratio of the system (\(\zeta < 1\)) by conducting the experiment. This is necessary because the damping ratio of a system is normally unknown. This experiment is usually carried out by applying an initial force to the system then the displacement \(x\) is plotted against the vibration time \(t\) so as to produce the graph shown in Figure 2.
if the logarithmic ratio is defined as:
\[
\delta = \ln \left( \frac{X_1}{X_2} \right) = \zeta \omega_n \tau_d = \zeta \omega_n \frac{2\pi}{\omega_n \sqrt{1-\zeta^2}}
\] (5)
so the damping ratio is:
\[
\zeta = \frac{\delta}{\sqrt{(2\pi)^2 + \delta^2}}
\] (6)

for very small damping ratios where \(X_1 \approx X_2\) and \(\delta < 1\) so that \(\zeta \approx \delta / 2\pi\). This means that if \(X_1\) and \(X_2\) have a value difference that is too small, the experimental results cannot be used. For this reason, the above analysis can be carried out by observing a wide range of periods.

3. Hydro elastic submarine model design
The hydro elastic segmented submarine model was developed as a geometrical similar of the 61.26 m submarines type U-209 for the main objective of measuring structural damping. The hydro-elastic model of the submarine based on the lines plan the submarine which is shown in Figure 3. To draw of this lines plan was very important and took quite a long time because there is no offset data available and it is only based on front projections of the hull from the internet. This line plan was then used as the basis for manufacturing the submarine model for hydrodynamic testing in the towing tank. The model scale of this submarine was selected at 1:30 with the consideration that this is the suitable size of the model that can be carried out on the towing tank.

Figure 3. Lines Plan of the Submarine Model

The submarine model was cut at two locations along hull to form three rigid segments. This construction system allows for the bending moments measurement at two points along hull girder. Each submarine model hull segment was constructed from fiberglass and the hull segments were bonded to an aluminum square solid section as a backbone beam. Each elastic segment connection will follow the elastic similarity rule as shown in Equation (7) and (8).

\[
(EI)_m = \frac{(EI)}{\lambda^5}
\] (7)

\[
\lambda = \frac{L_s}{L_m}
\] (8)
where \( E \) is the modulus of elasticity of the material and \( I \) is the moment of inertia of the cross-sectional area of the structure. \( \lambda \) is a geometric scale factor where \( L_s \) is the length of the ship and \( L_m \) is the length of the model. Then, \( (EI)_s \) is the flexural stiffness or bending strength of the submarine compression body structure which in this study was approximated by the U-209 submarine pressure hull structure which was in the form of a hollow cylinder with an outer diameter of 6200 mm and a thickness of 25 mm.

This hydro-elastic model scale 1: 30 with a steel bar backbone located in the keel which has flexural stiffness similar to the full-scale submarine (see Figure 4 and Table 1) from Wibowo H Nugroho et al [9].

![Figure 4. Submarine hydro-elastic model](image)

| Physical property                      | Units   |
|----------------------------------------|---------|
| EA/L (stiffness of backbone steel)     | 108.9 MN/m |
| LOA (length over all)                  | 2.040 m  |
| Breadth                                | 0.206 m  |
| Draught                                | 0.190 m  |
| Displacement                           | 55.45 kg |

### 4. Model experiments

The hydro elastic segmented submarine model tests were conducted at the towing tank facility of the Indonesian Hydrodynamic Laboratory (BTH, BPPT). Vibration experiments were carried out in both air and water to observe the dry and wet damping ratios. Strain sensor arrays were placed on top and sides of the steel backbone to detect changes in strain due to changes of impulse load from the hard rubber hammer. The position of the sensor is at a quarter of the length of the LOA calculated from the bow (ST 16), the middle of the ship (ST 10) and at a quarter of the LOA calculated from the stern of the ship (ST 04) where the largest normal stress is predicted. The strain gauge signals were measure at a sampling rate of 1000 Hz to have the response of the load. The data acquisition was done by a National Instruments (NI) Compact RIO running Labview FPGA and later sent via Ethernet to PC or laptop.

Wet mode and dry mode tests were carried out to obtain the attenuation of the submarine structure under conditions of zero forward speed. Vibration testing in dry mode is carried out by hanging the model in the air using a long soft elastic rope and applying an impulse load with a hard rubber mallet to a location selected from the position of the model's spine sensor. The same process is carried out for the wet mode vibration test with the model located on the surface of calm water.

Tests for vibration were also carried out on towing tank with carriage speeds of 1.03 m/s and 2.06 m/s. This test is to obtain the phenomenon of forward speed to hydro-elastic model during the test. A response of decay form oscillation at each of strain sensor location showed by the differential strain measurement.
Figure 5. Wet test of the submarine hydro-elastic submarine model

Figure 6. Hammer test in towing condition

Figure 7. Examples of measurement result of hammer test (filtered)
5. Results & discussions
This section discusses the analysis of the test results. The measurement results from the hammer test are shown in Figure 8 and 9.

Figure 8 presents the comparison of the structural damping ratio results between dry and wet test. The results of data processing for all impulse loads are in all three positions at the model ST 4 (0.25 L); ST 10 (0.5 L); ST 16 (0.75 L). These results indicate that there is a negligible variation in the value of the ratio for damping in the wet test and in the dry test at zero speed, namely in the range of 0.00547 – 0.00587 for dry conditions and 0.0174 – 0.0182 for wet conditions. The value of the dry condition seems close to the damping ratio of the backbone material which is Aluminum. The Aluminum (SSC 1991) has typical values of 0.005 – 0.007. However, the results clearly show a significant difference between the damping ratio in dry and wet condition. This may be due to the fluid interaction of the submarine model that increases the structural damping ratio.

The effect of forward speed on the damping ratio of the hydro-elastic segmented submarine model is carried out in the towing tank. Rigid hull segments are performed in fixed pitch and free pitch conditions which are used to determine the effect on the damping ratio. Figures 9 show the damping ratio as a function of increasing Froude number for test cases with fixed and free pitch condition. The results for the damping ratio are shown in Figure 9. This speed effect was also observed by Lavroff et al. [2]. The test results show clearly that the damping ratio increase when the forward speed is also increased. However, the highest damping ratio is produced in the pitch motion.

Figure 8. The comparison of the structural damping ratio results between dry and wet test

Figure 9. The structural damping ratio as a function of increasing Froude number
6. Conclusions
The results from this paper have found the conditions that affected the structural damping ratio. It has been found that the damping ratios of wet condition are greater than the dry one. The damping ratios also increased as the submarine model speeds forward. The test results show clearly that in both fixed and free pitch the damping ratio increase when the forward speed is also increased.

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