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To cite this article: Mohammad Basuki Rahmat *et al* 2020 *IOP Conf. Ser.: Earth Environ. Sci.* **519** 012045

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Vibration Analysis of Ship-RUV Structure in Operational Conditions

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Abstract. Ship-RUV (Remotely Under Vehicle) structure design must be focused on the principles of engineering, where the structure must be strong and resilient with regard to the SF (Safety factor) in operations. One of the excessive vibration conditions in the Ship-RUV structure will hinder the process of image capture and poor motion optimization. So the analysis of Ship-RUV design in operational conditions needs to be considered the existence of the frequency of excitation that occurs with the natural frequency of the Ship-RUV structure. At first, the technology developed in Ship-RUV with design and analysis using finite element. Finite element to solve the problem, it subdivides a large system into smaller, simpler parts with a numerical approach. Focus to strength a Ship-RUV Structure with analysis vibration effect load using finite element method. Actually, structural Ship-RUV is acceptable when actual excitation is 104.7 Hz. Frequency natural are 92.98 Hz and 122.8 Hz. Judgment to Structure Ship-RUV is acceptable of vibration effect.

1. Introduction

In operational conditions, there are some damages to the hull, including deformation, cracking, fatigue, fouling, and corrosion. To prevent further damage, it is necessary to investigate the hull process earlier. The investigation process in identifying the leaked hull must be carried out quickly and accurately before the ship sinks. Therefore very special equipment is needed to deal with leaks. Vessel leaks can be detected early if the dive is carried out to the side of the damaged hull. The conventional method that is often done is to send divers. These activities require great diving experts, who are able to fight ocean currents and bad weather and are also reliable in dealing with bad weather.

This can be overcome by using Ship-RUV. In this paper, the initial design of Ship-RUV is explained as shown in **Fig 1**. The initial design of this Ship RUV, some parts in the form of material that has a standard marine used. some components of the Ship-RUV compilation system, among others, propeller, Main Board Chip, Camera, Infra-Red and others are designed with good material



components and can be easily assembled. The design was made so easy to maneuver so that good data information was obtained in the Ship-RUV operation.

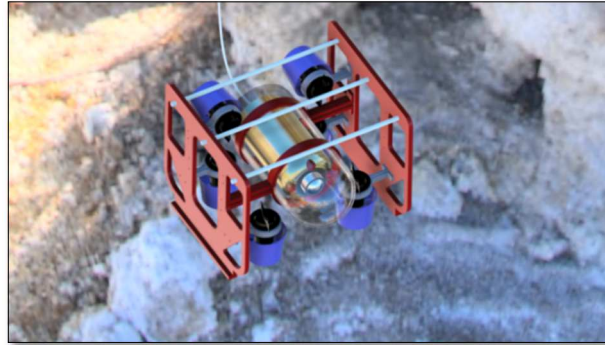


Fig 1 Ship-RUV Design

Every year the condition of the ship's hull always changes. Among them is the bio-fouling condition in the hull, which is a condition where the thickness of the hull plate is reduced due to the presence of marine biota attached to the hull, the condition can be in the form of deposit formation, encrustation, curding, deposition, scaling, scale formation, slagging, and mud formation. Therefore, every year the Indonesian Classification Bureau always checks the thickness of the hull plate. This is done to decide whether to give the feasibility of the ship or not give the feasibility of the ship in operation at sea.

The process of checking the thickness of the hull of the ship is done through a docking process. This process requires expensive costs and a long time. Therefore, Ship-RUV technology is needed that can directly investigation in the condition of the hull dipped in water, both due to fouling and even the deformation or crack conditions in the hull. With the presence of this Ship RUV the investigation process becomes low cost. Therefore the design of the Ship-RUV must be well designed to be reliable in operations, so as to provide accurate benefits. Ship-RUV structure design must be focused on the principles of engineering, where the structure must be strong and resilient with regard to the SF (Safety factor) in operations. One of the excessive vibration conditions in the Ship-RUV structure will hinder the process of image capture and poor motion optimization. So the analysis of Ship-RUV design in operational conditions needs to be considered the existence of the frequency of excitation that occurs with the natural frequency of the Ship-RUV structure. Where the structure can potentially have excessive vibration if the natural frequency value is equal to the excitation frequency that occurs [1]-[8].

2. Methodology

2.1 Modal Frequency

Determine the modal frequencies of the model. Structural Loads and Boundary Conditions can be included. The Results include Vibration Mode shapes, corresponding Frequencies and their mass participation factors. A new approach to free vibration analysis using boundary elements. A boundary element method for the analysis of free vibrations in solid mechanics is developed using a non-standard boundary integral approach. It is shown that, utilizing the statically fundamental solution and employing a special class of coordinate functions. [9].

The following are three special events that need to be considered in relation to the magnitude of the X vibration amplitude and phase angle when assuming that attenuation can be ignored. Preface and an example of static condensation for pinned joints is added. Orthogonally of Eigen vectors and the modal matrix and its orthonormal form enable concise presentation of basic equations for the diagonal eigenvalue matrix that forms the basis for the computation [10],

1. If the excitation frequency (ω) is much smaller than the natural frequency of vibration (ω_n), i.e. if $\omega / \omega_n \rightarrow 0$, then $X \rightarrow Y$ and $\alpha = 0$. In other words, at low frequencies, the response will move together with the excitation .
2. If the opposite occurs, ie if $\omega \gg \omega_n$ then $X / Y \rightarrow 0$ and $\alpha = 180^\circ$. In other words, if the excitation frequency (ω) is much greater than the natural frequency of vibration (ω_n), the vibrations that occur are close to zero and in the opposite direction to the excitation. This can be described as follows:

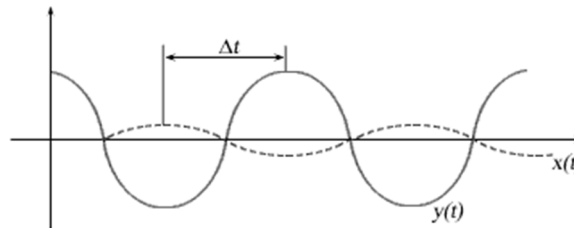


Fig 2 Frequency Natural and excitation

3. If $\omega = \omega_n$ (while $\zeta = 0$) then $X / Y \rightarrow \infty$ and α cannot be determined. In other words, if the excitation frequency (ω) is the same as the natural frequency of vibration (ω_n), then the vibration will be very large. This condition is called resonance, a condition that is always sought to be avoided by designers. Sketch the relationship between X / Y and α vs. ω / ω_n can be seen in the graph below:

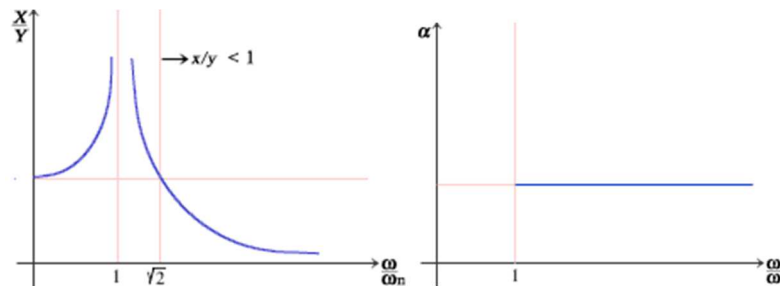


Fig 3 Resonance

2.2 External forcing function

A method to predict the effectiveness of application of damped dynamic vibration absorbers to suppress stationary random vibration of rectangular simply supported plates is given. Numerical examples of two different spatial distributions of the random-in-time forcing function are explored and optimal absorber parameters are presented [11]

External forcing function $F(t)$ varies with time and is externally applied to the mass M .

We will assume,

$$F(t) = F(m) * \sin(\omega) \dots\dots\dots (1)$$

Where :

F_m is the maximum applied force.

M is the mass of the vibration object that is equal to W/g .

ω is the angular frequency as defined below.

g the gravitational constant, $9,81 \text{ m/s}^2$.

X is the displacement from the equilibrium position.

C is the damping constant force per second velocity. and is proportional to velocity.

K is the spring stiffness force per inch.

Frequency of vibration f is cycles per second or Hz.

In the **Fig** bellow it is showed by component vibration effect. Excitation Frequency of object can be calculated by formulae empiric as show pattern above. It can vibration if value both as frequency occur.

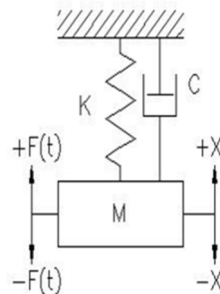


Fig 4 Suppression of vibration

2.3 Forced Un-damped Vibrations

Nonlinear coupled longitudinal and bending equations of motion are derived in non-dimensional form using the Hamilton principle. The first-order analytical solution of the equations of motion is obtained using the Galerkin technique combined with the multiple scales method (MSM) [12]. Numerical simulations are then performed for various increasing rates of the internal tensile force and performance of the vibration suppression strategy is studied. A very close agreement between the simulation results obtained by the numerical integration and the first-order analytical solution is achieved. Forced vibrations of the system for input excitations of either a sinusoidal or a random function with white noise time history are considered. The simulation results and dynamic performance of the suppressed system for an externally excited rotating beam show an interesting phenomenon of the form of remarkable effectiveness of the proposed vibration reduction strategy.

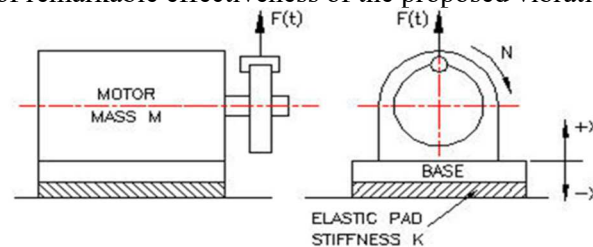


Fig 5 Stiffness

2.4 Finite Element Method

A boundary element method for the analysis of free vibrations in solid mechanics is developed using a non-standard boundary integral approach. It is shown that, utilizing the statically fundamental solution and employing a special class of coordinate functions, the algebraic eigenvalue problem results. The method has been realized numerically for the two-dimensional electrostatics, and a number of examples demonstrating its accuracy have been included [13]. In designing the ship's structure shall refer to the rules in accordance with applicable classification standards[14].

3. Discussion And Result

3.1 Un-damped Vibrations

If the mass m shown above is displaced through distance x and released it will vibrate freely. Un-damped vibrations are called free vibrations. Both x and g are measured in inch units.

$$\begin{aligned} \text{Weight, } W &= 0.25\text{Kg} \\ \text{Spring stiffness, } k &= 2.50\text{kg/mm} \\ \text{Gravitational Content } g &= 9.81 \text{ m/sec}^2 \\ \pi &= 3.142 \\ \text{Static Deflection, } x &= W / k = 0.10 \text{ mm} \end{aligned}$$

$$\begin{aligned}
 \text{Mass, } M &= W / g = 0.025 \text{ kg-sec}^2/\text{m} \\
 \text{Natural Frequency, } f_n &= 1/(2*\pi)*((k/12)*M)^{.5} \quad \text{Hz} \dots\dots\dots (2) \\
 &= 0.46 \text{ Hz} \\
 \text{Angular frequency, } \omega &= 2*\pi*f_n \text{ radn/sec} \quad \dots\dots\dots (3) \\
 &= 2.9 \text{ radn/sec}
 \end{aligned}$$

In the calculation above shows the smallest natural frequency value in the structure in an empirical formula that can be used as an initial comparison in determining the initial design.

3.2 Excitation Source Component

3.2.1 Main Engine

DC Motor as the driving force of ship-RUV can produce thrust power which has been calculated beforehand. So that Ship-RUV can operate properly and in accordance with its function. Excitation The shape of the DC motor that has been designed can be shown in the picture below as follows:



Fig 6 DC Motor

Specification of DC motor is applied in Ship-RUV design.

Motor diameter : 25 mm
 Motor length : 60 mm
 Supply voltage : 12 V
 Rpm : 1000 rpm

3.2.2 Propeller

Propeller is a component of the propulsion system in ship-RUV where it is planned that there are six propellers placed on the sides and bottom of the ship-RUV, so it is expected to have the function of driving the RUV in accordance with the planned design. The propeller ship-RUV can be shown as follows:

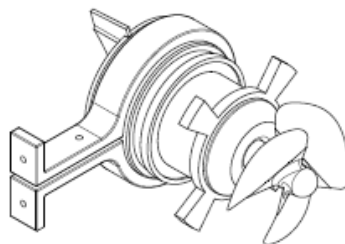


Fig 7 Propeller Ship-RUV

Specification of propeller is applied in Ship-RUV design.

The diameter of the propeller : 7 cm
 Number of blades : 3 pieces
 The diameter of the propeller housing : 10 cm
 Housing height : 2.5 cm

3.2.3 Frequency Excitation

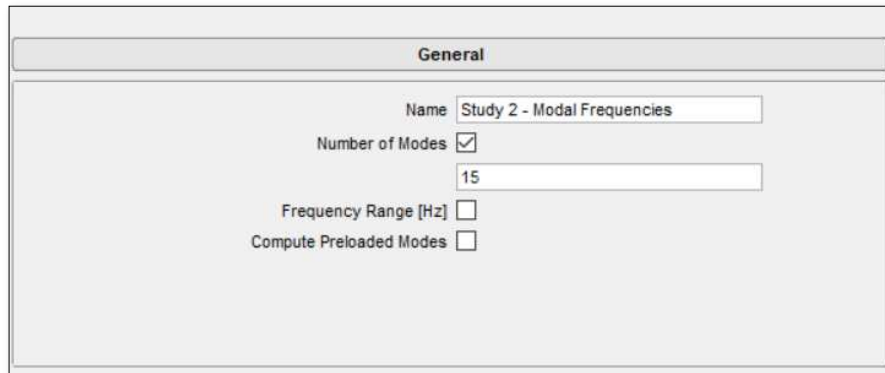
Frequency excitation is the frequency produced from the vibrating source. Where the vibrating source will be at high risk if the value is equal to the natural frequency they have. Therefore the easiest and cheapest way is to strengthen the structure by providing reinforcement. The excitation frequency calculation can be shown in the calculation below:

$$\begin{aligned} \omega &= 2\pi N/60 \dots\dots\dots (4) \\ &= 104.70 \text{ Hz} \end{aligned}$$

3.3 Setting Modal Analysis

Setting Shape Mode

Arrangements in conducting the analysis process using the finite element method can be done in the settings below. This aims to get the total modal frequency



Setting Meshing

The meshing arrangement can be used in determining the meshing size with the results of the fine mesh which was carried out with a series of previous convergence tests.

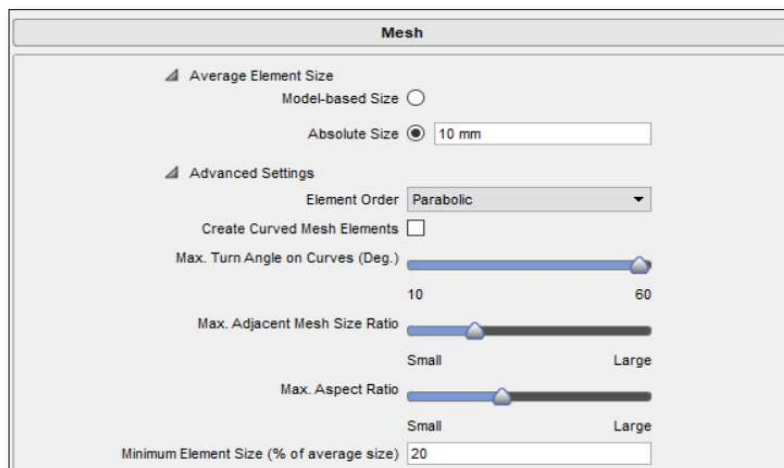


Fig 8 Setting S/W structural analysis

Modal Frequency Result

The calculation results of the analysis in the vibration analysis can be shown in the following Fig, which shows the 15 modal analyses which is the eigenvalue (natural frequency) of the structure that can be displayed in the calculation below:

Modal Analysis 1 of 15 (shown modal frequency 14)

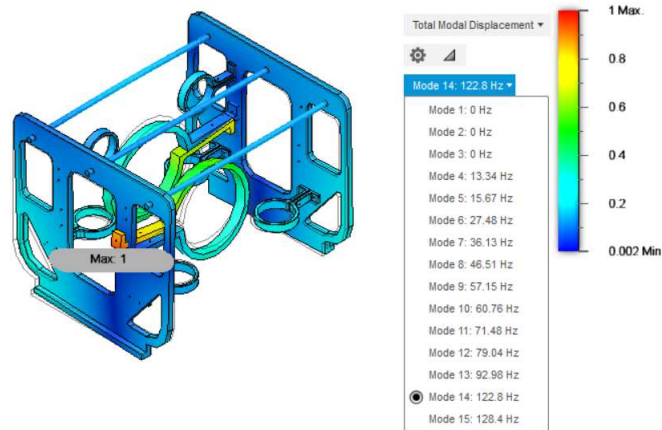


Fig 9 Result FE

Table 1 Table Summary Modal Frequency

Frequency	Participation X	Participation Y	Participation Z
Mode 1: 0 Hz	1.95029993	89.6628976	8.38679969
Mode 2: 0 Hz	97.7221012	1.35540003	0.92249997
Mode 3: 0 Hz	0.327600003	8.98180008	90.6907022
Mode 4: 13.34 Hz	0	0	0
Mode 5: 15.67 Hz	0	0	0
Mode 6: 27.48 Hz	0	0	0
Mode 7: 36.13 Hz	0	0	0
Mode 8: 46.51 Hz	0	0	0
Mode 9: 57.15 Hz	0	0	0
Mode 10: 60.76 Hz	0	0	0
Mode 11: 71.48 Hz	0	0	0
Mode 12: 79.04 Hz	0	0	0
Mode 13: 92.98 Hz	0	0	0
Mode 14: 122.8 Hz	0	0	0
Mode 15: 128.4 Hz	0	0	0

4. Conclusions

Frequency excitation is applied to strength a Ship-RUV Structure by Finite Element Analysis. It is a solution specific with Ship-RUV technology was developed by prioritizing simpler technology of size and strength of function capacity but using reliable structure technology and materials. At first, the technology developed in Ship-RUV with design and analysis using finite element. Finite element to solve the problem, it subdivides a large system into smaller, simpler parts with a numerical approach. Focus to strength a Ship-RUV Structure with analysis vibration effect load using finite element method. Actually, structural Ship-RUV is acceptable when actual excitation is 104.7 Hz. Frequency natural are 92.98 Hz and 122.8 Hz. Judgment to Structure Ship-RUV is acceptable of vibration effect

Acknowledgments

Thanks to Ministry of Research, Technology and Higher Education (Kemenristekdikti) on the grantees of research schemes applied by number contract: 1616.11/PL19/LT/2019 and several lectures in Politeknik Perkapalan Negeri Surabaya (PPNS) with sharing knowledge until finish this research.

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