Optimization Of Excitation Source And Dva Mass From The Weight Point In The 2-Dof Main Systems In Reducing Translation And Rotation Vibration

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Abstract. Excessive natural frequency (\( n \)) in a system is very undesirable because it will damage the system. This study used Single Dynamic Vibration Absorber (SDVA) to reduce translational and rotational vibrations to changes in the position of excitation sources. Prototype was built with a 2-DOF translation-rotation vibration system and with the addition of SDVA to 3 DOF. The prototype was modeled mathematically and simulated to determine the vibration response to changes in vibration characteristics that occurred. Simulation was done by varying the change in SDVA excitation distance to the main system mass center, and change in moment arm distance. The result of the translation and rotation direction attenuation was able to reduce vibration. The main system was not coupled if dva is in the center of the mass so it does not affect rotational motion. The largest percentage reduction occurs in the natural frequency area of the system 12.9 Hz on DVA the ratio of negative moment arm \( r_1 = -0.5 \) for the translation direction of 94.81%, and the rotation direction of 79.48% with an excitation distance of 0.148 m.

Keywords: Excitation, SDVA, DOF, Translation and Rotation Vibration.

1. INTRODUCTION

Vibration is a back and forth motion through equilibrium point in a certain time interval. Vibration is related to the oscillation of an object and the force associated with that motion. All objects that have mass and elasticity are able to vibrate, so that generally machines and engineering structures experience vibrations to a certain degree of freedom. The emergence of a vibration in the system can be viewed as one side of the benefits but also very detrimental. Vibration reduction has been developed previously from several studies therefore the concept was created to reduce the vibration. The Dynamic Vibration Absorber (DVA) is one concept that is used to reduce translational vibration response. DVA is an additional mass system imposed on a vibrating system supported by dampers. The DVA concept is also used to reduce translation and rotation vibrations simultaneously.

Previous research was performed by modifying DVA using Dual DVA-Independent on the main system. Changes in the arm moment have no effect on the vibration of the translation direction. The DVA concept by modifying the Dual Dynamic Vibration Absorber/DDVA arranged in series on the main system mass absorber 1 in reducing vibration of the main mass was reduced due to the mass absorber 2 which reduced vibration capability.

The DVA concept can be applied to multi-storey buildings to vibrate earthquakes and vibrations also occur on beams, bridges and overpasses. With many of these concepts, this research was conducted to understand the vibration characteristics to design a concept of vibration absorber dynamically. DVA also has another term Tuned Mass Damper (TMD) which is widely used Passive vibration control. In this study, it will be focused on the study of the position of the excitation force distance on the beam, and the change in the mass ratio of the absorber by changing the damping distance to the translational and rotational vibration response in the main Multi-Degree Of Freedom (M-DOF) system.

2. REVIEW OF RELATED LITERATURE

a. Dynamic Vibration Absorber(DVA)
Experiencing vibration problems can cause losses if the engine structure design is not designed by the control system. A system can experience excessive vibration if the force acting approaches the natural frequency of the system. In some vibration cases, a system can be reduced by adding DVA which consists of mass by connecting springs. DVA is designed to keep the system's natural frequency away from the frequency of the force applied to the main system. The vibration amplitude on the system is reduced by the smaller time function as shown in Figure 1. Dynamic vibration absorber (DVA) is a method that is widely used and is quite capable of reducing vibrations that occur in an engineering structure. Dynamic vibration dampers found by Hermann Frahm (US Patent # 989958, 1911) and these dampers are widely used to overcome vibration problems in various types of mechanical systems that experience external force or interference.

![Figure 1 The vibration amplitude of the system is attenuated by the time function](#)

b. Dynamic Vibration Absorber (DVA) Without Damper

![Figure 2 DVA Without Damper and Response Effect](#)

In figure 2. The transmissibility graph value changes from engine vibration amplitude \( X_1 \) because of the change in engine rotating speed \( \Omega \). Two peaks of the main system amplitude correspond to the two natural frequencies of the composite system.

c. Dynamic Vibration Absorber (DVA) with Damper

The original system only had one resonant peak, and then had two resonance peaks with added DVA. Thus the engine must pass peak resonance quickly when the engine is turned on or off to avoid very large amplitude. The amplitude of this machine can be reduced by providing a vibration damper as shown in Figure 2. Giving DVA without dampers results in changes in the value and number of peak resonances of the engine.

3. Research Method
3.1 DVA System Dynamic Modelling

In this study, it can determine the system dynamic modeling with the equation of the system motion. Using the Matlab Program can determine the Simulink block to obtain a vibration response. There is a main system in the form of a prototype in the following picture.

Figure 3 Research Flow Chart

Figure 4. Prototype DVA Machine (Dynamic Vibration Absorber)
Equations of system motion by decreasing mathematical models that is in accordance with a simplified free body diagram (FBD) system. The FBD used the Newtonian principle 2 for translation direction. Displacement of the beam can be derived from the equation of the motion system to analyze the direction of translation and rotation. This system affected the number of DOF (Degree of Freedom) in placing the position of the excitation force and the mass of the absorber.

**Figure 5** FBD for y direction for translation motion

- The state variable equation of y direction displacement
  
  \[
  \ddot{y}_a = \frac{1}{M_a} \left[ -(c_1 b + c_2 + c_{xy}) \dot{y}_b - (c_1 l_1 - c_2 l_2 + c_{xy} b) \theta - (c_1 + c_2 + k_{xy}) y_b \\
  - (k_1 l_1 - k_2 l_2 + k_{xy} b) \theta c_{xy} y_a + k_{xy} y_a + ma^2 R \sin \theta \right]
  \]

**Figure 6** FBD for direction for rotational motion

- The state variable equation of Angular Displacement
  
  \[
  \ddot{\theta} = \frac{1}{J} \left[ -(c_1 b + c_2 + c_{xy} b^2) \ddot{\theta} - (k_1 l_1 - k_2 l_2 + k_{xy} b) \theta + (-c_1 l_1 + c_2 l_2 + c_{xy} b) \ddot{y}_b \\
  - (k_1 l_1 - k_2 l_2 + k_{xy} b) \dot{y}_b - c_{xy} b \dot{y}_a + k_{xy} b \theta + ma^2 R \sin \theta \right]
  \]

- The state variable equation of Mass Absorber (M_a)Displacement
  
  \[
  \ddot{y}_a = \frac{1}{M_a} [ -c_{xy} \ddot{y}_a - k_{xy} y_a + k_{xy} y_a - k_{xy} b \theta + c_{xy} \ddot{y}_b - c_{xy} b \theta ]
  \]

3.2 Block Diagram Simulation
The state variables equation from dynamic modeling is the basis for building block diagram simulation in the form of Simulink. This simulation was performed to get a vibration response from the main system with DVA.

![Block Diagram Simulation (Simulink)](image)

3.3 System Parameter

Parameter values that represent a part of the DVA prototype measured by testing with formula calculations.

| Parameter Sistem | Simbol | Nilai | Satuan |
|------------------|--------|-------|--------|
| Ballast mass     | $m_k$  | 5     | kg     |
| Mass motor       | $m_m$  | 5     | kg     |
| Mass $\mu$       | $m_\mu$| 0,14  | kg     |
| Motor support plate and ballast weighting | $m_p$ | 3,09 | kg |
| Mass disk plate  | $m_d$  | 0,75  | kg     |
| Mass total sistem| $m$    | 13,88 | kg     |
| Motorcycle distance from CG | $a$ | 0,145 | m |
| Cantilever distance 1 from CG | $\ell_1$ | 0,23 | m |
| Cantilever distance 2 from CG | $\ell_2$ | 0,23 | m |
| Diameter disk plate | $r$ | 0,06 | m |
| Frequensi of motor work | $f$ | 0 sd 30 | Hz |
| Constanta Stiffness 1 and 2 | $k_1, k_2$ | 44802,7 | N/m |
| System Attenuation Coefficient | $c_s$ | 50,7 | N/m² |
| Absorber Damping coefficient | $c_a$ | 1,63 | N/m² |
| Moment of Inersia | $I$ | 0,2997 | kg.m² |

To obtain stiffness, the mechanics formula was obtained by giving a load to the cantilever tip of the system so that cantilever was deflected or deflated. Addition of load was carried out repeatedly with different load masses to obtain significant stiffness results.
For the damping constant value, the measurement required special test equipment, namely oscilloscope, accelerometer and power supply. In the attenuation phase measurement with the installation of an accelerometer on cantilever, where the excitation force or initial deviation is $x_0$, and allowed to vibrate until it does not vibrate again. Vibration that arises with the provision of deviation is recorded on the oscilloscope display. Vibration data stored in the oscilloscope is still in the form of voltage data so that converting is needed.

The cantilever attenuation measurement uses the logarithmic decrement method with the following equation:

$$c = 2 \cdot m \cdot \sqrt{k/m} \cdot \sqrt{\left[1/(2 \cdot \pi \ln(a_n/a_{n+1}))\right]^2 + 1}.$$ 

In which $a_n$ is the magnitude of the $u_n$ vibration amplitude, $a_{n+1}$ the magnitude of the vibration amplitude -(n + 1). The attenuation value of cantilever absorber can be obtained using the logarithmic decrement equation with the same step the difference in the accelerometer placement on the cantilever absorber.

Analysis of fundamental calculations for natural frequency values with DVA. The natural frequency value for $m_a = \frac{1}{10} m_0$ whereas the moment absorber arm value is $b = 0.13$ and $b = 0.26$

| m   | Natural Frequency(1) Hertz | Natural Frequency(2) Hertz | Natural Frequency(3) Hertz |
|-----|----------------------------|----------------------------|----------------------------|
| 0   | 10.9                       | 14.9                       | 28.4                       |
| 0.13 | 10.8                       | 14.8                       | 28.7                       |
| 0.26 | 10.7                       | 14.6                       | 29.5                       |

4. RESULTS AND DISCUSSIONS

Vibration response generated in the simulation by compiling the Simulink block which has known parameters of the prototype constituent. The simulation was designed to determine Root Mean Square (RMS) value. By knowing this, the vibrations obtained can be measured with both the Non-DVA and DVA systems.

Changes in vibration frequency in the main system will certainly give a change in the change in angle ( ) in the main system. In the picture above, a graph of the given frequency is shown, so the maximum amplitude and angular displacement values will decrease. At the frequency of 5.79 Hertz, the displacement response is 0.006399 rad. While the frequency of 12.972 Hertz produces a 0.021 rad displacement response.
The graph of the translational motion response system has a rms displacement resonance peak of 0.003237 m which occurs at the $r_f$ frequency ratio = 1. In the rotational motion response the system has a rms resonance peak displacement of 0.04585 rad which occurs at the $r_f$ frequency ratio = 1.

4.1 Placement of the moment arm ratio ($r_l$) at an excitation distance of 145 mm to the center of mass

The results of this study can be seen the characteristics of the vibration response rms. In figure 10 is a combination consisting of several graphs of analysis of the translation direction vibration response to the ratio of moment arm DVA to the position of the excitation source distance of 145 mm. Black graphics are rms displacement charts for systems without DVA. For graphs in red, green and blue, the graph of the negative momentary DVA arm ratio assumes that it is close to the excitation source and the magenta and cyan colors, which means that the plus position indicates that the positive DVA moment arm ratio away from vibration excitation.

With the addition of DVA mass to the main system, there is certainly an increase in the number of degrees of freedom and damped natural frequencies from the main system. When the system is given the same frequency as $\omega$, the system will show a maximum vibration response (resonance). As in the graph of the simulation results, the mass absorption of absorber at the center of the main system ($r_l = 0$) results in increasing the number of resonances from the displacement to 2 pieces, which occurs at $r_f = 0.8524$ and $r_f = 1.168$. Where for resonance without DVA there is still one resonance that is shown in a black gradation line.
The RMS graph of angular displacement at the first, second and third resonances became higher when with greater $r_f$. Things that were slightly different occurred at the second natural frequency, which at this frequency RMS angular displacement became large when the absorber mass was placed at the end of the system that approaches the excitation source.

From the simulation results on the rotational motion graph that giving DVA center of mass in the system is able to cause ant resonance at a certain frequency ($r_f = 1$) where the resonance peak value is influenced by the magnitude of the translation motion of $1.08 \times 10^{-2}$.

Rotational motion graph that giving DVA the center of mass in the system can cause ant resonance at certain frequencies ($r_f = 1$) where the resonance peak value is influenced by the magnitude of the ant resonance of translational motion by $0.0108$ rad. Compared to systems with DVA that have decreased resonance with amplitude. Decreasing the resonance value in rotational motion is affected by the giving of the ratio of the greater moment arm to the lower source of excitation and also applies to stay away from the source of excitation. In this description the translational vibration reduction affects the vibration response at a certain ratio where there is a reduction in the direction of rotation.

### 4.2 Reduction Optimization of Translation and Rotational Vibration

Simulation results with a percentage graph method of reducing rms displacement and angular displacement with the effect of the excitation source distance in the vibration reduction frequency range. The frequency of excitation given was in the range of 11.15 Hz - 15.15 Hz. The percentage reduction area was obtained from the intersection of lines on the system without dva with the graph moment arm ratio dva ($r_f$).
Figure 12 (a) Graph of Decreased Translation displacement and (b) Angular displacement to the effect of the source of excitation

Based on the graph where some DVA in the translation direction has the same damping which is only influenced by the distance of the excitation source and the moment arm ratio $(r_l)$. The percentage of reduction that occurs at a frequency of 12.8 Hz was 94.35% at the moment arm ratio $(r_l = -0.5)$. That reduction of vibration occurs in dva which is close to the source of excitation. While for the rotation direction with the excitation distance has a large reduction of 94.81% on the excitation frequency of 12.9 Hz at the negative moment arm ratio $(r_l = -0.5)$.

With the great errors percentage that occur due to the displacement or vibrational conductivity of most of the system to the body base to the table as the main holder. This happens with the addition of the given frequency so that the added excitation force on the vibrations caused is very high. Thus the vibration will affect the vibration at the experiment table. With the difference in the average overall percentage of great errors, it can be seen from the graph trend line above, it can be interpreted that there are similarities in the results of the research in simulation and experimentation.

5. CONCLUSIONS

Based on the results of the study, the attenuation of translational and rotational directions is able to reduce vibration. The main system is not coupled if dva is in the center of the mass so it does not affect rotational motion. The largest percentage reduction occurs in the natural frequency area of the system 12.9 Hz on DVA ratio of negative moment arm $r_l = -0.5$ for translation direction of 94.81% and rotation direction of 79.48% with excitation distance of 148 mm.

The construction design and spring parameter values determine the vibration response fluctuation. As a result, the DVA system reduces vibrations more in the direction of translation and rotation, the shape of the spring as a support system is replaced by a spring that has flexibility in the direction of the rotation force. However, the stiffness value needs to be considered in order to reduce vibration according to the desired design.

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