Response of Vibration Reduction with Additional of Dual Dynamic Vibration Absorber to The Main System

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Article Information

Manuscript Received 2023-03-29
Manuscript Revised 2023-04-11
Manuscript Accepted 2023-04-19
Manuscript Online 2023-06-30

ABSTRACT

Excessive vibration in a system occurs when the force acting on a system is close to the natural frequency of the system. This vibration can be reduced by adding a dynamic vibration absorber (DVA) to the system. Several researchers have done a lot of research related to DVA and dual DVA (DDVA) placement in the main system. In this research, we conducted a study of the effect of DDVA-dependent addition on translational and rotational vibration responses in the 2-dof main system. This research was started by building a prototype of a 2 DOF vibration system without DDVA and with DDVA. From the prototype, equations of motion and block simulations were then made to determine changes in vibration characteristics that occur in the main system. From the research results it was found that the placement of the DDVA-dependent cantilever absorber at the end of the system with rI 1/10 was able to reduce the vibration of the main system with a percentage reduction at a frequency of 12.78 Hz of 94.1681% for the translation direction and 15.3878% for the rotation direction. Changes in the distance ratio and the inertia ratio of the absorber mass do not affect the DDVA-dependent ability to reduce vibrations in the translation direction.

Keywords: *DDVA, vibration, rotational vibration, translational vibration*

1. INTRODUCTION

Natural frequency is the frequency when an object vibrates naturally without any outside interference or influence [1]. If an operating system experiences excessive vibration, it can cause damage to the components of the system and can even interfere with overall system performance [2]. Therefore, it is important to pay attention to factors such as natural frequency when designing and installing operating systems to ensure that they work optimally and safely. An operating system can experience excessive vibration if the force acting on it approaches its natural frequency [3]. This vibration can be reduced by adding a DVA to the main system [4][5][6]. DVA is designed to distance the system's natural frequency from its excitation frequency [3]. Initially, DVA was used to reduce vibrations in multi-story buildings due to earthquakes [7][8][9][10][11]. Even so, it is possible that DVA can be applied to many cases of vibration. Dynamic vibration absorbers are often used in engineering applications where vibration can cause damage, such as in buildings [12][13][14], bridges [15][16][17][18], wind turbines [19][20][21] and aerospace structures[22]. They are also used in machines [23], such as propulsion engines, turbines, and pumps, to reduce wear and tear caused by vibration.

Basically, DVA is an additional mass attached to the main system that experiences vibration [24][25][26]. Dynamic Vibration Absorber (DVA) is a mechanical device used to reduce vibrations in structures or machines by diverting vibration energy to a secondary system [27]. This tool is also known as a tuned mass damper or harmonic absorber [6]. With the additional mass, the number of degrees of freedom from the main system increased [5]. With the increasing number of degrees of freedom in the system, some of the vibrational energy present in the system will be channeled into additional masses[28][29]. The additional mass will vibrate against the direction of the system mass vibration; this aims to reduce the vibration of the main system [24][26].

In some cases, DVA is applied to reduce vibrations only in the translation direction [30][31]. Several studies, for example, have applied DVA to reduce translational and rotational direction vibrations in the main system in the form of a beam [5][32][33]. However, in this study [5] vibration reduction was only carried out in one direction of translational motion. The translational motions of DVA 1 and other DVAs are not related to each other. In this study, the main system vibration reduction process was carried out by adding DDVA-dependent The DVA used has two movements, namely translational and rotational, which are interrelated to reduce vibrations in the main system. This research begins with building a prototype consisting of the main system and vibration dampers. The main system is in the form of beams, which are connected to cantilever rods as a substitute for damper springs. On both sides of the beam arms, electric motors are provided as a source of excitation, and unbalanced masses are rotated by electric motors with a phase difference of 90° to obtain translational and rotational movements. As a vibration damper, an absorber mass (Ma) with a weight of 1/10 of the main system weight (Ms) is used and also has a certain absorber mass inertia (Ia).

2. RESEARCH SIGNIFICANCE

The existence of excess vibration in a system, especially vehicles, is not expected when designing. Because excessive vibration can affect the comfort level of passengers. It is necessary to develop the use of DVA in damping vibrations from translational and rotational movements. In addition, the results of this study can be used as a reference for the development of DVA in the case of translational and rotational movements.

3. RESEARCH METHODS

This study uses simulation, so it is necessary to determine the dynamic modeling of the main system. Figure 1 is a simplification of the prototype model, with a description of the figure 1 in Table 1 below.

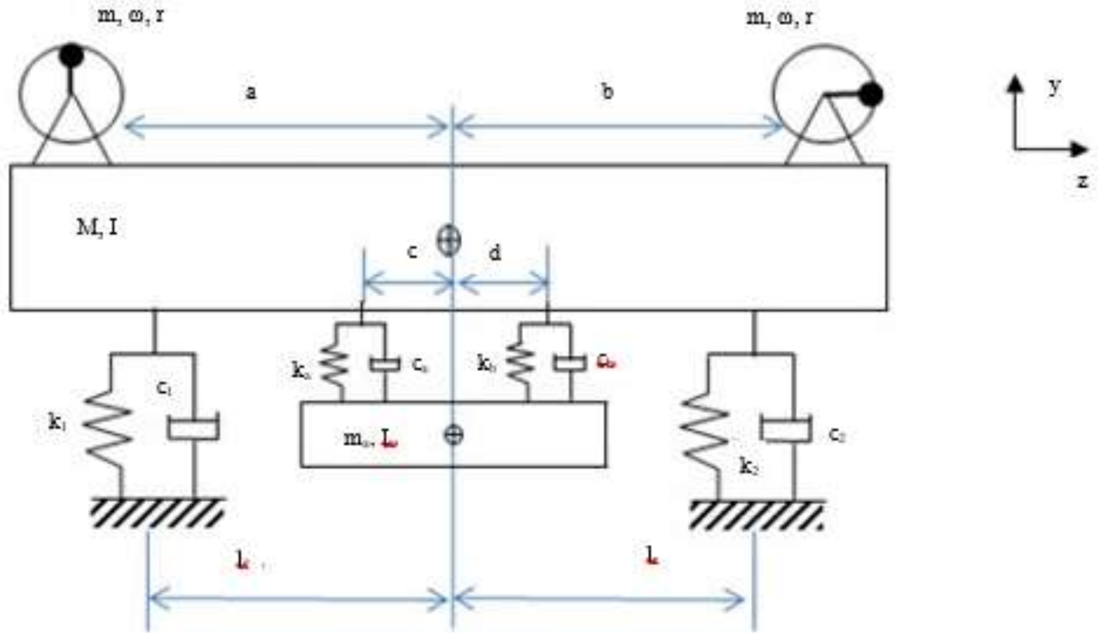

Fig 1. Mass dynamic modeling with Dual-DVA

From the simplified model in figure 1, degrees of freedom (DOF) are used to figure out the equations of motion. Because there were four degrees of freedom in this study, four equations of motion were obtained. From the equations of motion, state variable equations are made to be used in building simulation block diagrams.

Table 1. Symbols and descriptions

Symbol	Descriptions
m	unbalance mass
M	block mass
m_a	Absorber mass
r	radius of rotation of the unbalanced mass
ω	Angular velocity of motor
a & b	The distance of excitation source to CG beam
I	mass inertia of block
I_a	mass inertia of absorber
c & d	The distance of cantilever absorbers to CG beam
k_1 & k_2	stiffness of cantilever beam
k_a & k_b	stiffness of the cantilever absorber
c_1 & c_2	damping of beam
c_a & c_b	Damping of cantilever absorber
L_f & L_r	The distance of catilever beam to CG beam
L	Total length of beam

The parameters used were obtained from measurements and tests directly on the DVA prototype. The main system stiffness constant value can be determined by conducting tests and measurements directly on the test prototype. To carry out this test, several variations of the load and calipers are needed to measure the deflection in the cantilever due to the application of the load. From the results of this test, the average deflection value (Δx) is sought which has been measured from both sides of the cantilever as Figure 2. The damping constant of each cantilever beam is obtained by testing. The test is carried out by giving an initial deviation to the main system and allowing it to vibrate freely. The vibration response that occurs from the main system mass is then displayed and taken using an accelerometer and oscilloscope to be processed using a logarithmic decree.

In determining the stiffness constant of the cantilever absorber, a comparison of the stiffness of the main system is carried out. Where each cantilever absorber stiffness depends on the magnitude of the total stiffness of the main system and also the ratio of parameters used. The parameter ratio used is 1/20, where this ratio refers to previous studies. So that the stiffness of the absorber used is (k1 + k2)/20. With constant values of absorber stiffness ka and kb respectively 4480.272 N/m.

As with the damping constant of the main system, the damping constant of each absorber is also obtained by carrying out the test. The test is carried out by giving an initial displacement to the absorber mass and allowing it to vibrate freely. The vibration response that occurs from the absorber mass is then displayed and taken using an accelerometer and oscilloscope to be processed using a logarithmic decree. Data processing is done by finding the average amplitude of an adjacent peak and valley (an) and then finding the average of a peak and a valley that is next in sequence (an+1). The average of the first amplitude and the average of the second amplitude are then entered into the logarithmic decree (δ) equation.

From the measurements and tests. following are the simulation parameter values used:

M	: 13,884 Kg	k_1, k_2	: 44802,72 N/m
m	: 0,14 Kg	c_1, c_2	: 60,199 N.s/m
I	: 0,323 Kg	c_a	: 2,1979 N.s/m
a,b	: 0, 26 m	L	: 0,53 m
l_f, l_r	: 0, 145 m		

Fig 2. Cantilever Beam Height Measurement

For the mass weight of the absorber, use a ratio of 1/10, and for the stiffness of the absorber, use a ratio of 1/20 [10]. As for the mass inertia of the absorber, a change is made from IS/40 to IS/10. The excitation force is a periodic excitation with a phase difference of 90°. The excitation frequency used ranges from 0 Hz to 30 Hz.

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4. RESULTS AND DISCUSSION

In this study, the main system has 2-DOF, namely translational and rotational directions. The translation direction system response is represented by the system acceleration response and the rotation direction system response is represented by the system angular acceleration response. This system is given an excitation force from a motor which is given an unbalanced mass with a 90o phase difference to create vibrations in the system. The main system model without the addition of dual DVA-dependent is used as a comparison to the system with the addition of dual DVA-dependent. System analysis without the addition of dual DVA-dependent is done by analyzing calculations and simulations with Simulink Matlab.

Figure 3 shows an increase in the acceleration value along with an increase in the excitation frequency value, namely at an excitation frequency of 0 Hz to 12.89 Hz. The acceleration value decreases with increasing the excitation frequency value up to 17.8 Hz. In this condition a peak is formed in the acceleration response of the system. This peak indicates an increase in acceleration at the excitation frequency of 12.89 Hz, followed by a decrease in acceleration at the next excitation frequency. At excitation frequencies above 17.8 Hz, the acceleration value increases with increasing excitation frequency values. This is because the input value of the excitation force is a function of the square of the frequency so that the acceleration response has a tendency increase with increasing excitation frequency.

In system modeling, the input used is the excitation force from the motor and ballast due to unbalanced masses which have a phase difference of 90°. The excitation force from the motor and ballast is in the form of a sinusoidal (harmonic) input. From the sinusoidal input, the response displayed is the response of the RMS acceleration and RMS angular acceleration to variations in the excitation frequency. The motion response in the direction of translation of the system is indicated by the response to the system acceleration and the response in the direction of rotation of the system is indicated by the response to the angular acceleration of the system. The response is shown in figures 3 and 4.

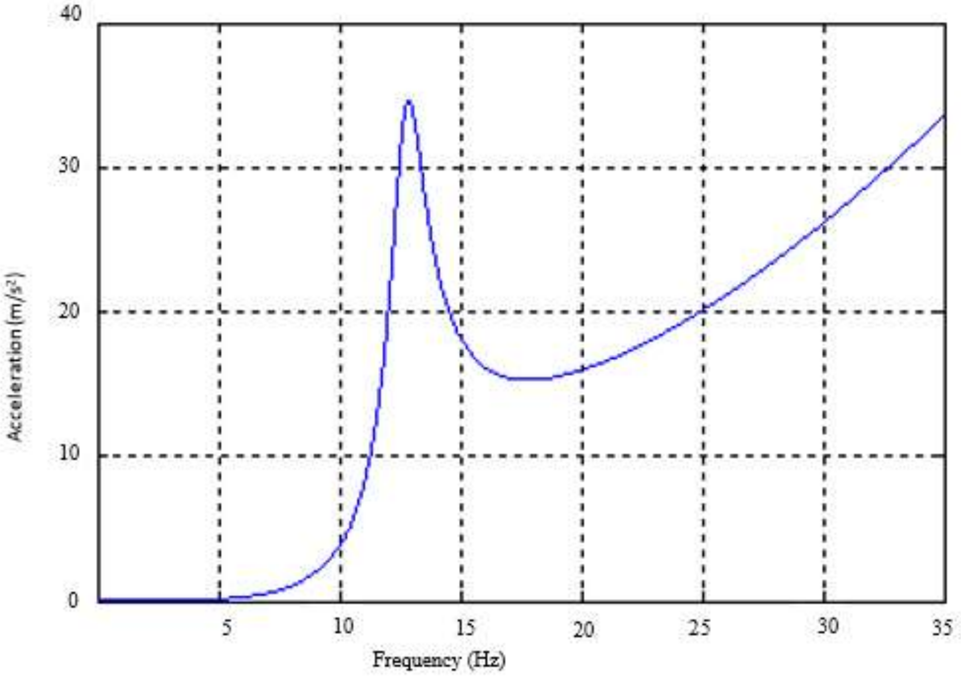

Fig 3. Graph Response of Acceleration to Frequency Ratio in Systems without Addition of DVA

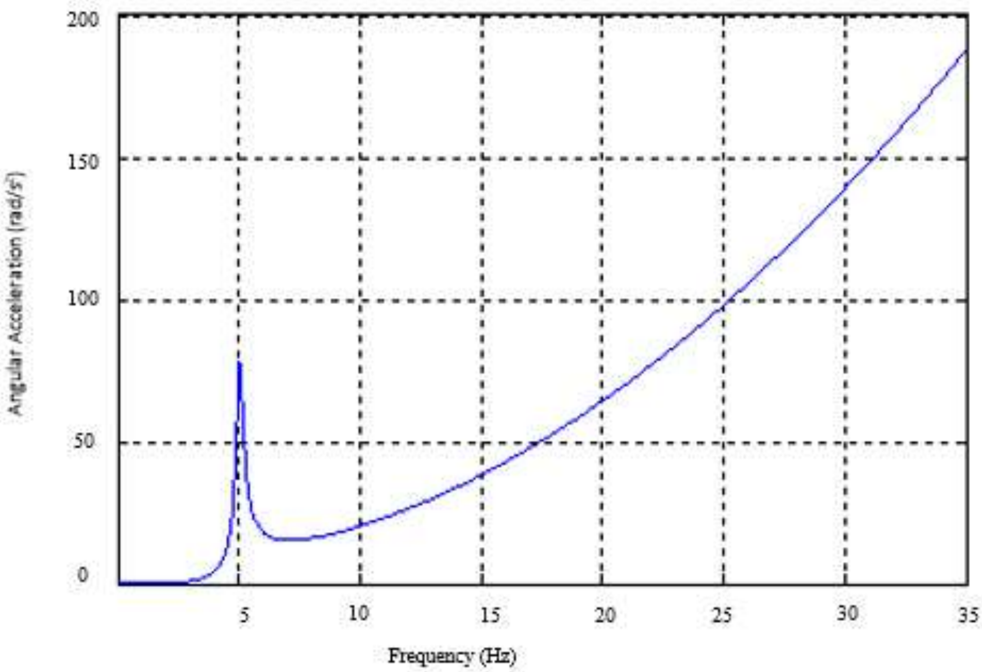

Fig 4. Graph Response of Angular Acceleration to Frequency Ratio in Systems without Addition of DVA

The angular acceleration value increases with the increase in the excitation frequency value as shown in figure 4. This phenomenon occurs at an excitation frequency of 0 Hz to 5.04 Hz. Then the value of the angular acceleration decreases at the value of the excitation frequency up to 7 Hz. Under these conditions a peak is formed in the angular acceleration response of the system. This peak indicates an increase in angular acceleration at the excitation frequency of 5.04 Hz, followed by a decrease in angular acceleration at the next excitation frequency. At excitation frequencies above 7 Hz, the acceleration value increases with increasing excitation frequency values. This is because the input value of the excitation force is a function of the square of the frequency so that the angular acceleration response tends to increase with increasing excitation frequency Based on the results of the simulation, the natural frequency of the system is shown by peaks in the system's acceleration response and angular acceleration. With an excitation frequency of 12.89 Hz and an acceleration of 34.6645 m/s^2, the natural frequency in the direction of translation is 34.6645 m/s^2. In the direction of rotation, the system's natural frequency happens at an excitation frequency of 5.04 Hz and a value of 78.5069 rad/s2 for the angular acceleration. The natural frequency of the system differs between translation and rotation in that the translational and rotational motions of the main system are unrelated.

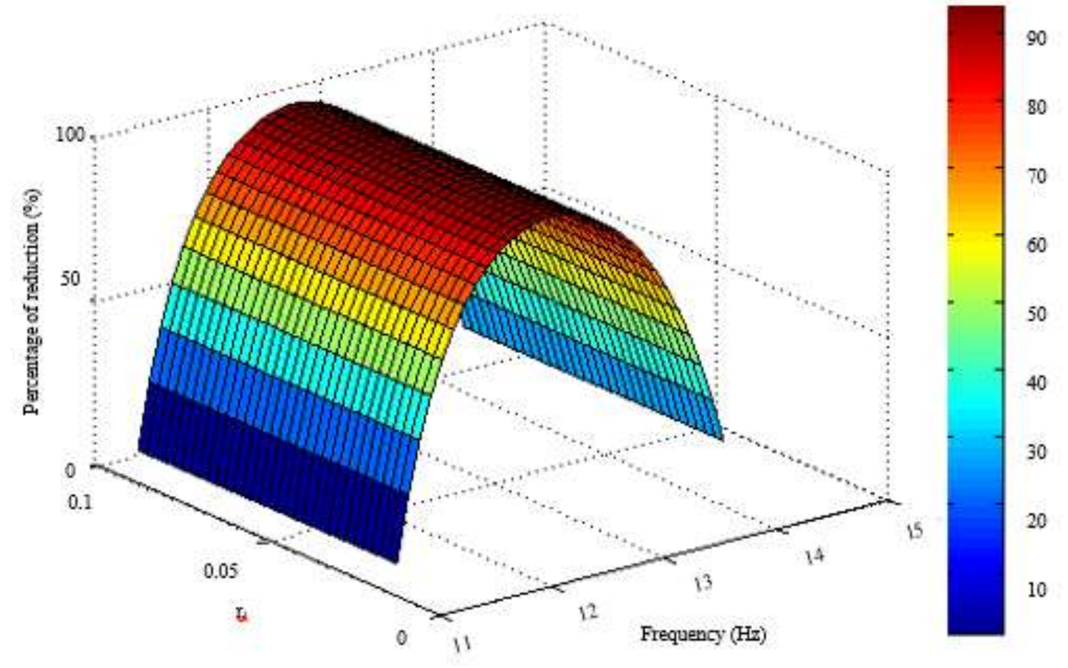

Fig 5. Decreased vibration in the direction of translation

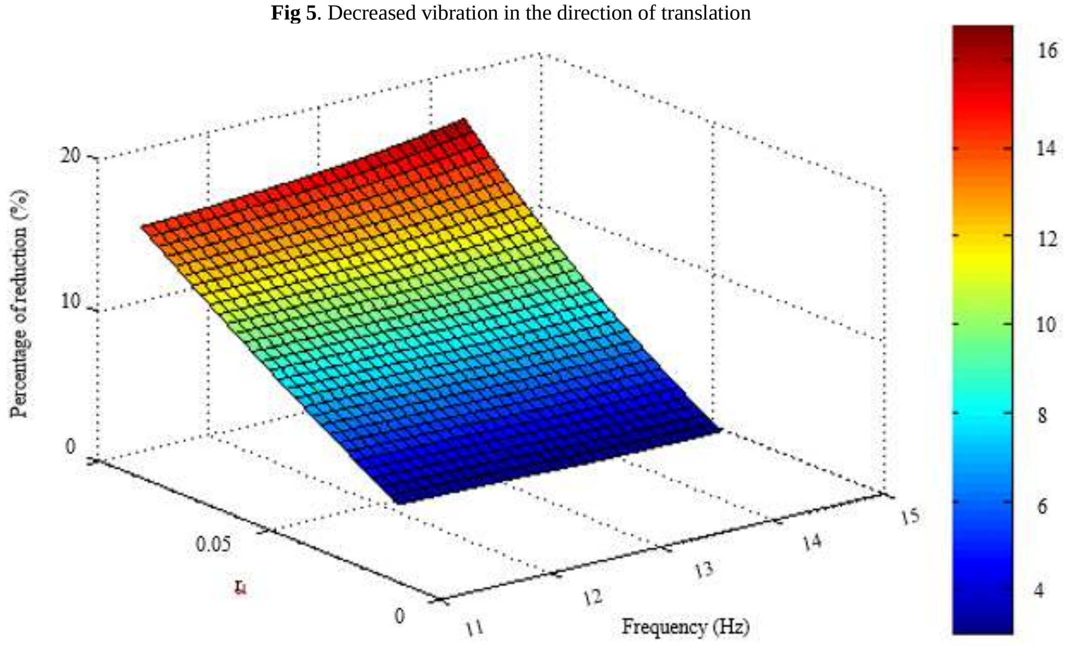

Fig 6. Decreased vibration in the direction of rotation

System modeling with dual DVA-dependent addition has 4 DOF. This movement includes translational movement of the main system, rotational movement of the main system, translational movement of the absorber, rotational movement of the absorber. In this modeling, the DVA parameter is determined by using a comparison to the main system parameters.

Variation of parameters in Dual DVA-dependent was carried out to obtain parameters that are able to dampen vibrations in the direction of translation and rotation and are optimum at a certain frequency range. The simulation of decreasing acceleration and angular acceleration is carried out by giving the DVA parameter for absorber mass inertia of Ia = Is/10 and in the frequency range 11.38–14.28 Hz, because in this frequency range for the translation direction there is a very good decrease in vibration. Graphs of response results from the simulation can be seen in Figures. 2 and 3.

Vibration reduction simulation for variations in absorber mass inertia is carried out by providing a distance of RL = 1 on the cantilever absorber. The excitation frequency is given in the range of 11.38 Hz–14.28 Hz. The response results are shown in Figure 4. Figure 2 shows that the decrease in acceleration increases with increasing excitation frequency and tends to be constant with changes in the ratio of inertial mass (RI). In the direction of rotation (Figure 3), variations in inertia greatly affect the decrease in the angular acceleration of the main system. The greater the mass inersia ratio of the absorber, the greater the decrease in angular acceleration. The magnitude of the percentage reduction in acceleration and angular acceleration due to variations in absorber inertia is summarized in Table 2.

Based on table 2, the ability of DDVA to damper vibrations in the translation direction tends to be the same at different absorber mass inertias. The mass-inertia ratio of the absorber does not affect the ability of the DVA to reduce vibrations in the translation direction. The biggest percentage reduction is obtained when the DVA is at a frequency of 12.78 Hz, which is 94.1684%; this is because that frequency is the natural frequency of the main system. As for the direction of rotation, variations in the inertia of the absorber mass have an effect on reducing vibrations in the direction of rotation. The greater the ratio of rI given, the greater the reduction in vibration. The greatest vibration reduction is obtained when the ratio rI = 1/10. At this mass-inertia ratio, the greater the excitation frequency given, the greater the percentage reduction in vibration. The percentage of maximum vibration reduction given is 16.7781% at a frequency of 14.28 Hz.

Table 2. Percentage of vibration reduction

Freq. (Hz)	**Percentage of vibration reduction on r_L = 1**					
	r_I = 1/40		**r_I= 1/20**		**r_I=1/10**	
	a (%)	**α (%)**	**a (%)**	**α (%)**	**a (%)**	**α (%)**
11.38	0.9213	3.2917	0.9219	6.8525	0.9234	14.7493
11.58	36.6319	3.2696	36.6323	6.833	36.6332	14.8142
11.88	67.5826	3.2404	67.5828	6.8114	67.5832	14.934
12.18	83.7749	3.2121	83.775	6.7944	83.7752	15.077
12.48	91.8784	3.1699	91.8785	6.7677	91.8786	15.2309
12.78	94.1684	3.1044	94.1684	6.7219	94.1685	15.3878
13.08	91.1029	3.0464	91.1029	6.6864	91.103	15.5757
13.38	83.3219	3.0211	83.322	6.6851	83.3222	15.8174
13.68	69.6844	3.0153	69.6845	6.7052	69.6849	16.1028
13.98	47.4777	3.0168	47.478	6.7349	47.4787	16.4228
14.28	12.825	3.0241	12.8254	6.7725	12.8266	16.7781

5. CONCLUSIONS

In the change in mass inertia ratio (rI) for the frequency range 11.38 Hz–14.28 Hz, the ability of DDVA to dampen vibrations in the translation direction tends to be the same for different absorber mass inertias. The mass-inertia ratio of the absorber does not affect the ability of the DVA to reduce vibrations in the translation direction. The biggest percentage reduction was obtained when the DVA was at a frequency of 12.78 Hz, which was 94.1684%. As for the direction of rotation, changes in mass inertia (rI) have an effect on reducing vibrations in the direction of rotation. The greater the ratio of rI given, the greater the reduction in vibration. The greatest vibration reduction is obtained when the ratio rI = is/10. The percentage of maximum vibration reduction given is 16.7781% at a frequency of 14.28 Hz.

6. ACKNOWLEDGEMENTS

I would like to express my deepest gratitude to all the participants who took part in this study. Their willingness to share their time and experiences has been essential in providing the data needed to carry out this research.

7. AUTHOR CONTRIBUTIONS

- Conceptualized the study: Talifatim Machfuroh, Zakiyah Amalia, Fica Aida N.A.
- Designed the research methodology: Talifatim Machfuroh, Zakiyah Amalia, Fica Aida N.A.
- Conducted data analysis: Talifatim Machfuroh,
- Data collection and analysis: Fica Aida N.A.
- Manuscript preparation, reviewed, and approved the final manuscript: Talifatim Machfuroh, Zakiyah Amalia, Fica Aida N.A.

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