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**A NEW TUNED VIBRATION ABSORBER BASED ON ONE-
DEGREE-OF-FREEDOM OF TRANSLATIONAL MOTION**

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Abstract. A new tuned vibration absorber (TVA) based on one degree-of-freedom (1DOF) of translational motion is proposed. The absorber mass takes the shape of a tank filled with fluid. This device is attached to the fluid circulating system which plays a role in changing the absorber mass. It fills the tank or drains it depending on the data received from the frequency force sensor. The type of TVA used is similar to the simple damped TVA. A numerical procedure is used to test the TVA and compare the performance with the other TVAs. This method is simple to carry out the performance of the TVA as it is adequate for real-life problems.

Key words: Damping, Degree of Freedom, Tuned Vibration Absorber, Varies Mass System.

Introduction.

A conventionally tuned vibration absorber (TVA) is a single-degree-of-freedom (SDOF) system, which is used to attenuate the vibration of a structure at a particular forcing frequency. The SDOF-TVA is a vibration device that has received the attention of many researchers, which was initially proposed by Frahm about a century ago [1] to configure a mass-spring system. The damper has been introduced in the TVA by Ormondroyd and Den Hartog [2] as they show its effect and how it can broaden the range of frequency. Different aerospace, automotive, and civil engineering sectors have been studied for some decades since its instigation. In its most generic form, a TVA is an auxiliary system whose parameters can be tuned for suppressing the host structure's vibration. The vibration at its point is suppressed by the TVA to host structure using the interface force application as the auxiliary system is majorly considered as a spring-mass-damper system. The undamped natural frequency with its base blocked is represented by the tuned frequency of the TVA.

Den Hartog [3] introduced the two-step design technique, which starts with the selection of resonance frequency and then the determination of optimal level damping. It is based on the observation of two invariant (fixed) points in the frequency response function of the TVA and the host structure. The two fixed points are the same when changing the damping. The optimal value of damping can be obtained by averaging the two damping coefficients obtained by setting the derivative at the two fixed points to zero [3, 4]. This approach gives a good performance, but this approach gives an optimal design for specific range vibration. Many researchers have been conducted in the TVA that allows changing its stiffness so it can broaden the frequency range. Wu and Shao [5] introduced a virtual vibration absorber in which the virtual spring stiffness was tuned depending on the difference of the primary body and the acceleration based on the algorithm. Another active dynamic vibration absorber has been developed using the voice coil motor in [6, 7].

The vibration energy of the vibrating main system has been widely researched through TVA or TMD. Its concept is to segregate the vibration energy part of the main system throughout the secondary system. Generally, an additional mass-spring-damper system was used in TVA or TMD associated with the main system and considered as active vibration control. The design of TVA is similar to that of the operating frequency of the main system. Recently, the progression of TVA emerges as an active and adaptive TVA, additionally to

passive TVA. Den Hartog [3] introduces initially the fundamental concept of passive TVA while investigating the method of reducing the influence of ship rolling. It has been observed that plenty of problems to man-made structures as well as devices are rose due to vibrations. It can be annoying for individuals in a vehicle or a building or it might even lead to structural collapse or metal fatigue. The least expensive and most efficient way to reduce vibration issues is to instigate damping, which is a mechanism for mitigating the energy from the vibrations [8].

There are several advantages of using TVA; for instance, easy mounting, effective to reduce low-frequency band, and simple design. TVA is effectively used at a specific-defined operating frequency as an anti-resonance [9]. When the operating or excitation frequency is constant, the best performance of TVA is shown in reducing vibration. In contrast, the performance of undamped TVA is low when the operating frequency is differentiated. A study has developed a damped TVA to reduce vibration of the main system in a wide frequency range [8]. Another study has instigated an adaptive vibration absorber for minimizing transient vibrations and steady-state vibrations [10]. To validate the control and robustness performances, a study uses a hybrid dynamic vibration absorber of multi-degree freedom structure using experimental work. Another study has shown the effectiveness of active vibration control for builders subjected to vertical and horizontal large seismic excitation [11]. A study conducted on vibration control of slab breaker machines improves the effectiveness of the TVA by using a passive dual-mass tuned vibration absorber [12].

In this regard, the study has described a new type of tuned vibration absorber (TVA) one degree-of-freedom (1DOF) of translational motion. The system has been mathematically and numerically modeled and simulated. The simulation results were experimentally verified and analyzed. Due to the variation of the vibration source distance and TVA distance from the COG, this model was appropriately utilized for simulating the vibration responses and vibration reduction of the 1DOF main system. The type of TVA used in this study was similar to the normal spring-damper mass system, but tuning will be the act of changing in mass and not changing in stiffness.

Methodology.

In this study, the MATLAB Simulink has been used to create a simple model of an SDOF with the translational move from the basic equations. The TVA has been studied initially by deriving the equations of the system as the exciting force was applied on a primary mass. Afterward, a procedure design was conducted to find out the parameters of the TVA. Finally, the effect of changing the absorber mass was carried out. However, the idea of variable massed system has been presented later.

Model.

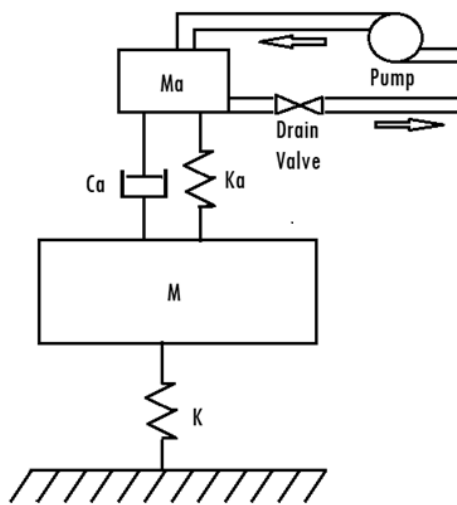


Fig. 1

The TVA used in this study is shown in fig. 1. The TVA consists of spring and damper in parallel k_a , c_a , and mass m_a , which was connected to a host structure of mass m_1 and stiffness k_1 using the spring and damper. A circulation system has been attached to supply and drain the fluid, which plays an effective role in changing the absorber mass. The TVA may translate vertically as the host structure was subjected to base excitation to be moved in the vertical direction.

Analysis.

The equation of motion will be:

$$m_1 \ddot{x}_1 + k_1 x_1 + c_a (\dot{x}_1 - \dot{x}_2) + k_a (x_1 - x_2) = F. \quad (1)$$

$$m_a \ddot{x}_2 + c_a (\dot{x}_1 - \dot{x}_2) + k_a (x_1 - x_2) = 0. \quad (2)$$

The resonance equation has been analyzed to

$$\frac{x_1}{\delta} = \left(\frac{2\zeta g^2 + (g^2 - f^2)^2}{2\zeta g^2 (g^2 - 1 + u g^2)^2 + (u f^2 g^2 - (g^2 - 1)(g^2 - f^2))^2} \right)^{0.5}, \quad (3)$$

where

$$\zeta = \frac{c}{c_c}; \quad g = \frac{\omega}{\omega_n}; \quad f = \frac{\omega_a}{\omega_n}; \quad u = \frac{m_a}{m_1}. \quad (4)$$

Parameters.

The selection of absorber damping depends on absorber spring and mass, specifically on its machine or structure that needs to attenuate the vibration. The following parameters were selected for the simulation results. Design parameters for the model

Variable	Primary structure	Value
Ms		5
Ks		2000
	Absorber structure	
Ma		1
Ca		0,8
ka		300

Results and Discussion.

The codes and the Simulink model were shown in the Appendix. For the first experiment, the natural frequency was set to be constant while the exciting force applied on the primary mass where the changes in frequency force effect were shown in fig. 2. By putting a value for ca, the vibration changes as shown in fig. 3.

The working of TVA was effectively observed in unique structures that possess similar natural frequencies with a specific force-frequency range. To use this absorber in different structures, a change in absorber spring or mass was applied. Since this paper was focusing on the mass, the absorber spring will be set as constant, although a circulating fluid system was attached. Fig. 4 shows the effect of changing mass while the frequency force was set to be equal to natural frequency.

There was a low peak when changing the absorber mass where it should fall in. When the fluid fills in the tank, the mass will change as long as the weight. To take in response to the other conditions of frequency force, the ratio of w/w_a has been studied in fig. 5.

Fig. 5 shows that the increase or decrease in the absorber frequency is equal to the force-frequency, which will attenuate the vibration located on point 1 over the w/w_a axis. Another case was studied

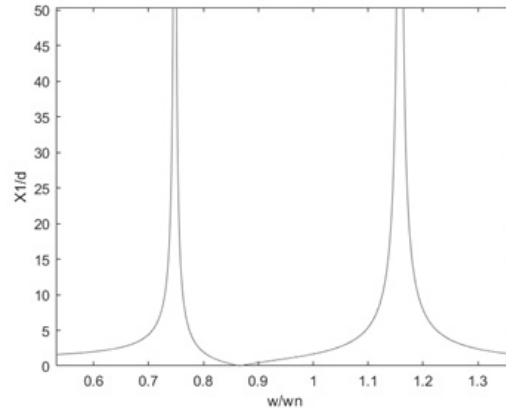


Fig. 2

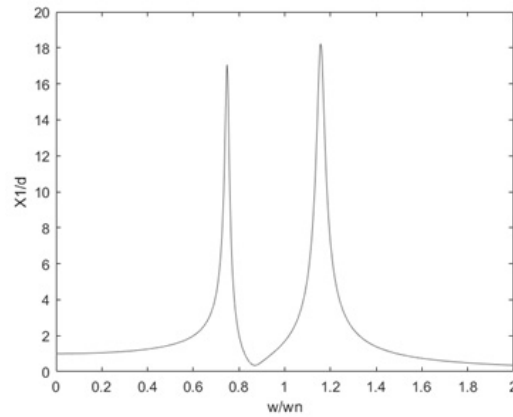


Fig. 3

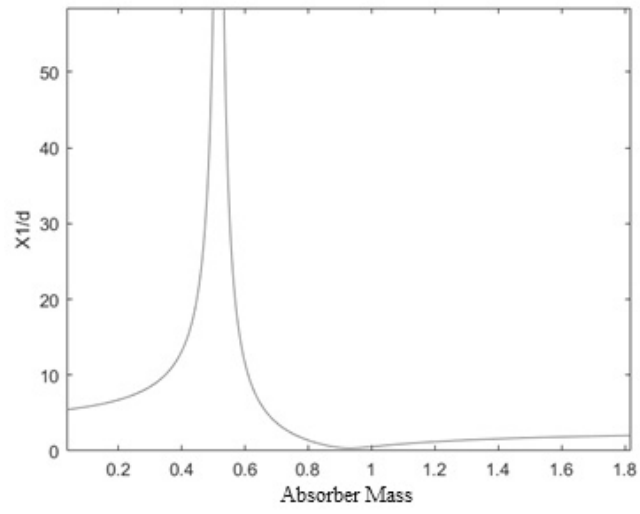


Fig. 4

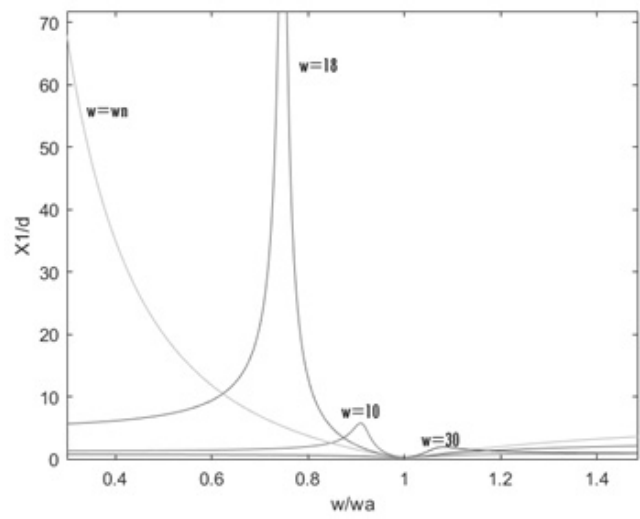


Fig. 5

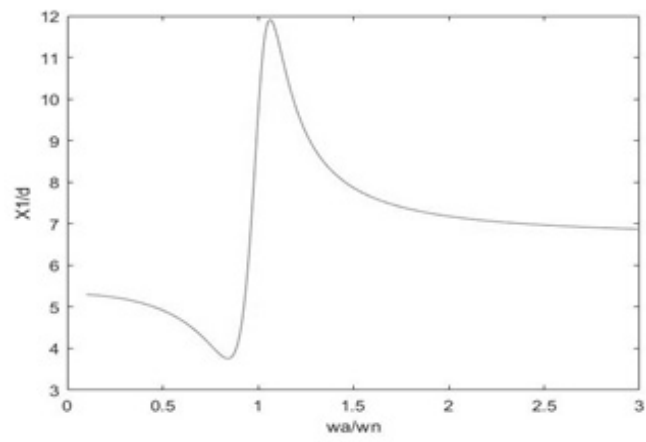


Fig. 6

when all the three frequencies (Natural, Force, and Absorber) were equalized and a sudden change in force-frequency occur. Fig. 6 and 7 show clear low peaks where the absorber frequency must fall in. These figures were obtained from a sensor attached to the system.

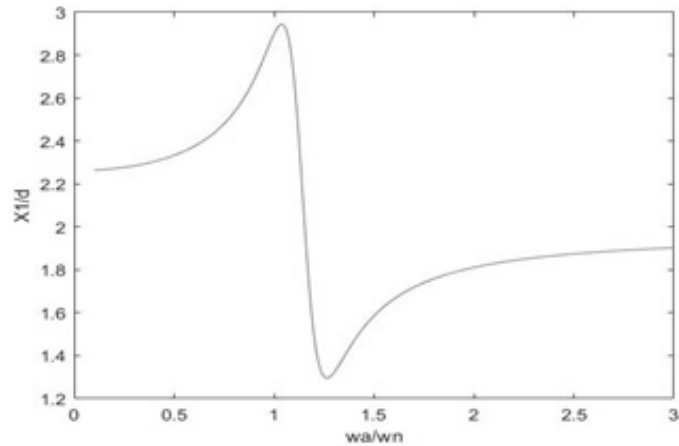


Fig. 7

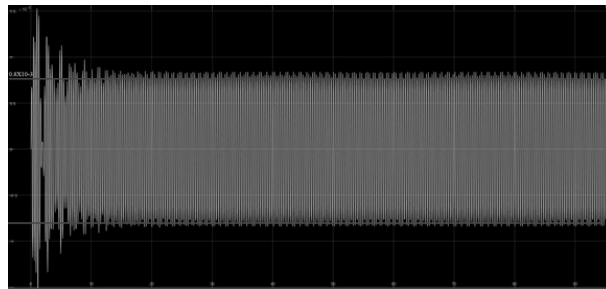


Fig. 8

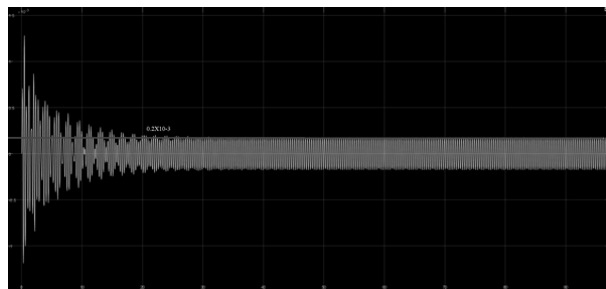


Fig. 9

The result of Simulink model was compatible with the previous results where parameters from the Table were inserted and the force-frequency was set to be equal to the natural frequency. Fig. 8 presents the resonance concerning time in which the absorber frequency was computed to be less than the force-frequency.

The sensor must measure the resonance and the circulation system must act to reduce the mass by draining the fluid using the drain valve. By reducing the absorber mass, its frequency will increase and it will stop draining when the two frequency matches. Fig. 9 shows the final result. From fig. 8 and 9, it was obvious that the effect of the TVA reduced the peak of the resonance from $0,8 \times 10^{-3}$ to $0,2 \times 10^{-3}$.

Conclusion.

In this study, a SDOF TVA has been developed by adding the variable mass system, using the circulation fluid system to change the mass as per the system required. A model graph was extracted using MATLAB software. It was shown from the graphs that by increasing or decreasing the force-frequency, a low peak was developed in the resonance graph. Therefore, to attenuate the vibration, the frequency ratio must fall in that low peak. In particular, reducing the force-frequency will lead to reducing the absorber frequency and will require adding fluid to absorber mass. The size of the tank (absorber mass) can be obtained by less force-frequency when applied to the system. The tank weight (empty) should be applied based on the maximum force. Further research is needed to study the time required for the system. The presented control algorithm was devised for prioritizing the frequency control of the TVA 1DOF by precise stiffness emulation as damping and stiffness emulations were computed with semi-active damper quantities. Thereby, it was essential for future studies to emphasize on the progression of damping and stiffness correction strategies for improving the damping emulation accuracy to maintain precise stiffness emulation.

РЕЗЮМЕ. Запропоновано новий резонансний поглинач коливань (РПК), оснований на поступальному русі з однією степінню свободи. Поглинальна маса приймає форму резервуара, наповненого рідиною. Цей пристрій приєднаний до системи циркуляції рідини, яка відіграє роль у зміні маси поглинача. Він заповнює резервуар або зливає його залежно від даних, отриманих від датчика величини амплітуди коливань. Тип РПК, що використовується, схожий на простий амортизуючий РПК. Чисельна процедура використовується для тестування РПК та порівняння продуктивності з іншими РПК. Цей метод є простим для роботи РПК, оскільки він адекватний реальним проблемам.

КЛЮЧОВІ СЛОВА: демпфування, степінь свободи, резонансний поглинач коливань, система зміни маси.

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