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Study on the Dynamic Stability of the Excavator Working Device

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Abstract

Objectives: The wide variety of possible work excavator, a large vibration occurs in small movements. The main purpose is to damp the vibrations of the excavator of the working device. **Methods/Statistical Analysis**: In order to verify the damping performance of the excavator of the working device, using the vibration analysis of the working device model, it was to understand the dynamic characteristics. In addition, this study sought a way to reduce vibration caused by the excavator working device using TMD (Tuned Mass Damper). In this study, it conducted a study on the dynamic stability of the excavator of the working device. **Findings**: It was determined the dynamic characteristics (natural frequencies and mode shapes) of the excavator working device through the modal analysis. In addition, using a numerical analysis, it was possible to determine the vibrational response from the working device. By using this, comparative analysis for vibrational response of the applied the TMD excavator and apply to not excavator. Depending on the vibration damping, and proposed that it is possible to design a more stable excavator. **Improvements/Applications**: It is suitable for the design of the excavator working device in consideration of the dynamic stability and vibration reduction. In addition, to reduce the operator's fatigue caused by the vibration of the excavator.

Keywords: Damped Vibration, Dynamic Stability, Excavator, Tuned Mass Damper, Working Device

1. Introduction

An excavator, which is one of heavy construction equipment, is widely used for works that men have difficulty handling. Since the working devices of an excavator is bulky and long, its small motion can cause much vibration through a working device and thus it prevents precision work. So excavator can be dynamic stability problems in the work relatively sophisticated maneuverability is required[1,2].

This study was conducted on the dynamic stability at the tip of excavator working device under various motions to minimize its vibration. As a solution, the study suggests using TMD for the working device to reduce vibration.

The Figure 1 shows a model of 30-ton excavator and Table 1 shows the material properties of the excavator. Utilizing the design model, it is possible to realize various movements of the excavator. Of many motions an excavator takes, Max. Digging Radius is one made where the tip of the excavator working device is located farthest from an operator's seat, and the vibration on the tip is the greatest. In this study, it was to study the dynamic stability of the movement of the Max. Digging Radius.

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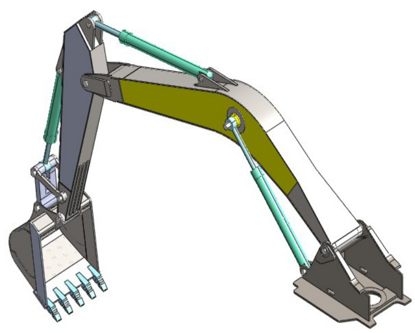

Figure 1. 30-Ton excavator (design model)

Table 1. Properties of materials

Material	Young's modulus [N/m^2]	Density [kg/m^3]	Poisson's Ratio
Steel	**2.0×10^{11}**	**7850**	**0.3**

2. Subject

2.1 Vibration Analysis

In order to verify the damping performance of the excavator working device, it is necessary to understand the dynamic characteristics. The modal analysis is used to find out the vibration characteristics (natural frequencies and mode shapes) of a structure or a machine component in the process of its design[3,4]. The following Table 2 shows the dynamic properties of excavator working device through modal analysis. It is natural frequency (or frequency of vibration of normal mode) by mode calculated with boom and arm of an excavator whose boundary condition was set to free-free[5]. Free-free boundary conditions, the boundary conditions for the experimental model, it verifies the reliability of the finite element model.

Table 2. Dynamic behavior of excavator working device.

Working Device	Mode Shape			
	Rigid Mode	1	2	3
Boom	0 [Hz]	0.9504 [Hz]	74.472 [Hz]	133.76 [Hz]
Arm	0 [Hz]	102.33 [Hz]	165.96 [Hz]	226.62 [Hz]

2.2 Analysis Model of Working Device

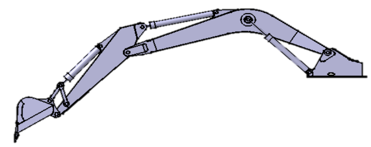

Figure 2. Max. digging radius position.

The Figure 2 is an implementation of the Max. Digging Radius motion. A simple mathematical model can be derived like Figure 3 [2].

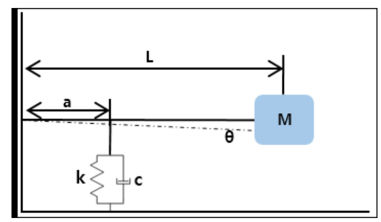

Figure 3. Lumped mass mathematical model.

The dynamic equations of the mathematical model are expressed in (1) and (2).

$$mL^2\ddot{\theta}(t) + ca^2\dot{\theta}(t) + ka^2\theta(t) = f(t) \quad (1)$$

$$M\ddot{\theta}(t) + C\dot{\theta}(t) + L\theta(t) = F \quad (2)$$

m : Mass of the working device modeled at one degree of freedom.

c : Damping constant of boom cylinder.

k : Spring constant of boom cylinder.

L : Distance from boom fixture to the center of total mass of the working device modeled at one degree of freedom.

a : Distance from boom fixture to the mounding spot of boom cylinder.

When applying numerical integration to the second order differential equations simplified at 0 degree of freedom in Equation (1) and (2), the vibration response of the working device can be calculated. Numerical integration method is using a Runge-Kutta 4 Equation, damping ratio and natural frequency of the excavator working device was assumed as shown in the following Table 3. Numerical analysis results are as follows in Figure 4.

Table 3. Natural frequency and damping ratio X

Natural Frequency (w_n, rad/s)	**15**
Damping Ratio (ζ)	**0.03**

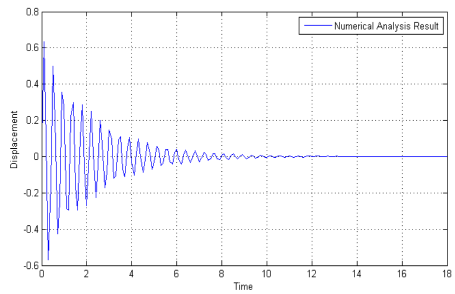

Figure 4. Numerical analysis result.

2.3 Tuned Mass Damper (TMD)

TMD is a sort of vibration damper mounted to a main device to absorb vibration. Therefore, it is a protective device for a main device. At large, it is added to a main device to absorb vibration.

In this study, TMD was designed for an excavator working device and dynamic equations at simplified two degrees of freedom were formulated to confirm that applying designed TMD to the working device reduced vibration response[6]. Figure 5 shows a mathematical model that is simplified applying the TMD.

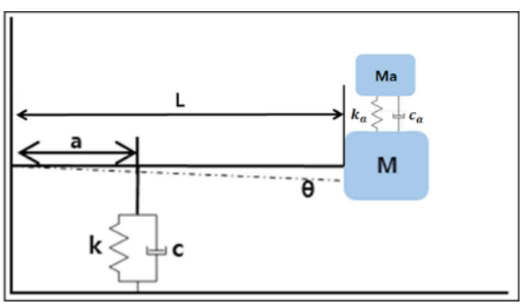

Figure 5. Tuned mass damper model.

TMD system is composed of a main system and the damping system, and the dynamic equations of the mathematical model are expressed in (3) and (4).

$$m\ddot{\theta}(t)+(c_a+c)\dot{\theta}(t)-c_a\dot{\theta}_a(t)+(k_a+k)\theta(t)-k_a\theta_a(t)=f(t) \quad (3)$$

$$m_a\ddot{\theta}_a(t)-c_a\dot{\theta}(t)+c_a\dot{\theta}_a(t)-k_a\theta(t)+k_a\theta_a(t)=0 \quad (4)$$

m_a: Mass of absorber
c_a: Damping of absorber
k_a: Stiffness of the absorber
θ_a: Magnitude of the displacement of the absorber mass

To summarize the two equations, as the following (5), it is possible to obtain the dimensionless original expressions of the amplitude of the main system[6].

$$\left|\frac{\theta K}{F}\right|=\sqrt{\frac{\left(-r^2+\beta^2\right)^2+4r^2\zeta^2}{\left(\left(-r^2+1+\beta^2\mu\right)\left(-r^2+\beta^2\right)-4r^2\zeta\zeta_a-\beta^4\mu\right)^2+4r^2\left(\left(-r^2+1+\beta^2\mu\right)\zeta+\left(\zeta_a+\mu\zeta\right)\left(-r^2+\beta^2\right)-2\beta^2\mu\zeta\right)^2}} \quad (5)$$

r: Frequency ratio = $\frac{w}{w_n}$
β : The ratio of the decoupled natural frequencies = $\frac{w_a}{w_n}$
μ: The ratio of the absorber mass to the primary mass = $\frac{m_a}{m}$
ζ: The ratio of the primary damping and $2mw_n$
ζ_a: The ratio of the absorber damping and $2m_a w_n$
w_a: Natural frequency of absorber = $\sqrt{\frac{k_a}{m_n}}$

It is possible to determine the parameters (r, β, ζ_a, μ) in the (5). Assume that the mass ratio (μ = 0.2) and zeta ($\zeta = \zeta_a = 0.01$). If the region of the $(0 \le \beta \le 1)$ and '$0 \le r \le 2$' can be obtain graph as following Figure 6.

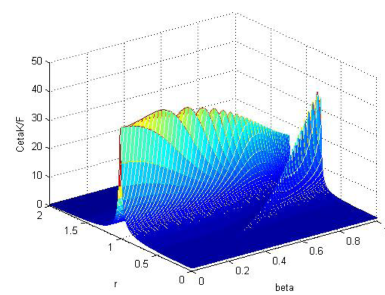

Figure 6. Stiffness, β Frequency ration r.

Beta value ($\beta = 0.84$) that fits comfortably in the above regions are shown in the following Figure 7.

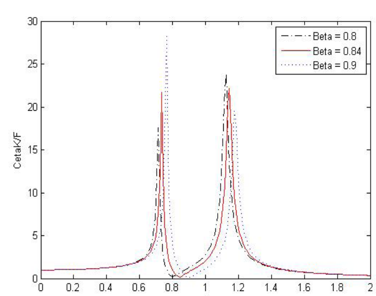

Figure 7. Optimal beta value ($\beta = 0.84$).

Assume that the mass ratio (μ = 0.2) and apply the beta value (β = 0.84). Determine the optimal zeta_a (ζ_a) value in the '$0 \le r \le 2$' region. Value of the result ($\zeta_a = 0.3$) is as follows in Figure 8.

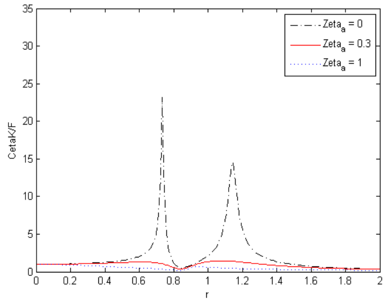

Figure 8. Optimal Zeta_a Value (ζ_a= 0.3).

Accordingly, the vibration response of an excavator equipped with TMD is as follows in Figure 9[7]. And Figure 10 is a comparison graph of the vibration response of the excavator is not applied with TMD applied excavator.

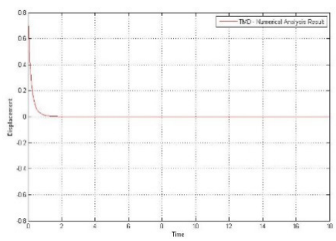

Figure 9. TMD_Numerical analysis result.

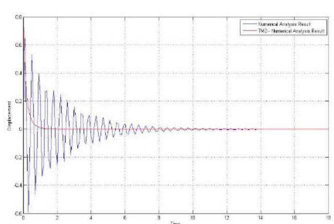

Figure 10. Compare TMD excavator and non excavator.

3. Conclusion

Dynamic stability of the excavator equipped with TMD through Figure 10 can be relatively stable. Excavator equipped with TMD was confirmed to be stably maintained within 2 seconds by disturbance. In this study, since the application of the ideal damping ratio, it is possible to know the good stability visibly. In order to design the actual TMD must be considered to making and cost. Since accordingly an ideal damping ratio can't be obtained so, would not be equivalent to the results of the present study. However, by designing the TMD, could be dynamic stability confirms that better.

The present study examined the dynamic characteristics of excavator working devices by using modal analysis and determined the dynamic stability of them using simplified mathematical modeling of excavator motion (Max. Digging Radius). In addition, TMD was designed for the excavator working devices to estimate that it can more secure the dynamic stability of the excavator working devices by reducing vibration.

From the findings above, this study demonstrated a method to secure dynamic stability and vibration reduction in designing an excavator working device.

4. Acknowledgement

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