Post-earthquake Dynamics of Bridge Structures using New Particle Dampers – A Case Study of the Nujiang River Bridge

Gong Lei* and Tang Miao

CCCC Second Highway Consultants Co., Ltd., Wuhan 430056, Hubei, China
E-mail: gonglei660218@126.com
**Corresponding Author*

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Abstract

In this study, a new mechanical model, named particle damping mechanics model (PU_SPD), is proposed to study the damping problem of bridge structures. The model takes the Nujiang River Bridge as a case study, and explores the mechanism of force action by analyzing the time domain vibration characteristics and frequency domain of the excitation force, vibrating body (bridge structural properties) and particle damping. the PU_SPD model and its calculation method can intuitively and scientifically describe the damping dissipation characteristics of a vibrating beam under the action of particle damping, avoiding the tedious process of parameter iterative solution and improving the computational efficiency. In addition, the damping influence law of particle damping on the beam structure is derived through the analysis of transfer function and damping level. The study also proposes an optimal design method for PU_SPD damping parameters under dynamic loading of the bridge, and its performance parameters are analyzed and verified, and compared and validated with the time-domain analysis method. The results

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show that the PU_SPD mechanical model based on time-frequency domain analysis can intuitively reflect the damping dissipation mechanics with high accuracy, clear solution process and reasonable and accurate parameter optimization analysis method. PU_SPD has a wide frequency range, good effect and stability, and has a good prospect of application in engineering vibration and noise reduction.

Keywords: Particle dampers, mechanical dynamic model, damping dissipative mechanics, mechanical modulation, frequency domain analysis, dynamics optimization.

1 Introduction

Particle dampers use the momentum exchange and damping effect generated by the impact force and friction between particles and cavities to Dynamic acceleration response in impeded vibration body excitation mode [1–3], which has the advantages of good damping effect, wide action band, robustness, distributed arrangement, and high-cost performance [4]. The particle dampers can be divided into single-particle dampers [5] and multi-particle dampers [6] based on the number of particles. Yan Weiming et al. [7] made a detailed analysis of the differences in the damping dissipative mechanics and performance of these two types of particle dampers and clarified the advantages and shortcomings of both. Using mechanical theoretical analysis, numerical simulation and experimental investigation, the damping dissipative mechanics, performance and damping effect of particle dampers have been studied in depth. The analysis is based on the phase trajectory in the time domain. Lu Zheng et al. [10, 11] explored the necessity of considering the effect of friction between particles and controlled structures based on the method; Yan Weiming [12] proposed a segmental computational model to solve the impact problem of single particle dampers by using the time course analysis method. Zhao Wenli [13] considered the inelastic impact as an equivalent spring-damping model action. The discrete element coupled finite element approach is a good treatment and is most appropriate for granular damping [14, 15], which determines the position relationship between the particles and Vibrating body in the time course and solves dynamic behavior characteristics and state response time course of the particles and Vibrating body by using the time domain analysis method. For the experimental study of particle dampers [16], the main purpose is to obtain Structural Dynamics response of the Force Structure by inputting the dynamic time course load to the test object, and then analyze its damping performance and mechanism. In terms of frequency domain analysis, researchers generally study the performance of particle dampers in an indirect way, which mainly includes two ways: one is to perform the Fourier transform based on the time domain analysis results of the particle dampers to obtain the frequency domain analysis results and further study the damping performance, for example, the steady-state acceleration response of each measurement point on the particle damped composite plate was collected and the corresponding transfer function was obtained by Su Junzui et al. [17]. The results showed that the particle dampers have good damping characteristics over a wide range of frequencies; the other is to equate the particle dampers to TMDs [18, 19] or DTMDs [20] and study their performance by using the frequency domain analysis methods on TMDs or DTMDs. However, the theoretical study of the damping dissipative mechanics and performance of particle dampers involves impact problems and has nonlinear characteristics. In addition, particle dampers are essentially force (acceleration) dependent dampers, which do not have a fixed vibration frequency of their own. Regarding the optimization analysis of the performance of particle dampers, although the mechanical model of complex dampers and the optimization analysis method are established by the time domain analysis method, the method is complicated and the calculation efficiency is low. Although intelligent algorithms [21, 22] can also be used to analyze the optimized performance of particle dampers, the calculation is tedious and lacks a theoretical basis.

The methodological difficulties in the performance analysis of damping structures with additional particle dampers lie in the mechanical characterization of the impacts between the particles and Vibrating body. It is difficult to solve dynamic behavior characteristics and state response in a impact by using time domain or frequency domain analysis methods alone, so the system with impact problems can be solved by combining time domain analysis with frequency domain analysis. For example, Zhao Dengfeng [23–26] studied the impact model of a linear system and performed the dynamic analysis and analytical solution of the model based on the transfer function (frequency domain analysis) and the time domain response.

In summary, the method of parametric coupling analysis in the time domain is relatively complex, for example, the mechanism of the structural vibration frequency and the influence of the excitation frequency and amplitude on its performance cannot be directly obtained, and the calculation process is lengthy, more importantly, the theoretical and numerical models established by the time domain analysis method are mainly based on the particles and Vibrating body. The theoretical and numerical models established by the time domain analysis are mainly based on the observation of dynamic phenomena and the constraints of Vibrating body, which cannot visualize and fully reflect the vibration reduction mechanism of the particle dampers. The frequency domain analysis method has the obvious advantage that it is not necessary to solve the differential equations, and it is easy to solve the structural response of some special excitation forms (step excitation, impulse excitation) [27, 28]. In addition, for tuned dampers such as TMDs, their performance is closely related to the excitation frequency, and the relationship between the damping performance and dynamic behaviors of Vibrating body and the input load frequency can be established directly by using the frequency domain method, while particle dampers are force related dampers. And their performance is also closely related to dynamic behaviors of Vibrating body, the excitation frequency and the excitation amplitude [29, 30]. However, it is extremely difficult to solve dynamic behavior characteristics and state response of Vibrating body of particle dampers by using frequency domain analysis alone.

During the design and construction of the Mengnuo Nujiang River Special Bridge as part of the Ruili-Menglian Expressway, it is essential to adhere to seismic design standards and codes. These standards encompass methodologies for calculating seismic loads, material selection, structural connections, and seismic damping measures. They ensure that the bridge possesses adequate strength and resilience during earthquakes. In light of existing research, this paper introduces a Parallel Unidirectional Single Particle Damper (PU_SPD) to address the limitations and applications of particle mechanics dampers. Figure 1 illustrates a mechanical analysis comparing the PU_SPD with two other structures. The study delves into the mechanical mechanism of PU_PSD, conducting an in-depth analysis of its damping dissipative mechanics. By pulsating the excitation force, the energy consumption of the damper's particle impact is simulated. A combined frequency domain and time domain approach is employed to analyze and solve the mechanical model, proposing a parametric analysis method to achieve optimal damping effectiveness for PU_PSD under dynamic loads. This approach differs from the simple time domain analysis. Lastly, a comprehensive comparison of the damping dissipative mechanics and performance between PU_PSD and TMD verifies the characteristics and advantages of the PU_PSD mechanical model and its solution method. Figure 2 illustrates the manufacturing process of particle damping as explored in this study.

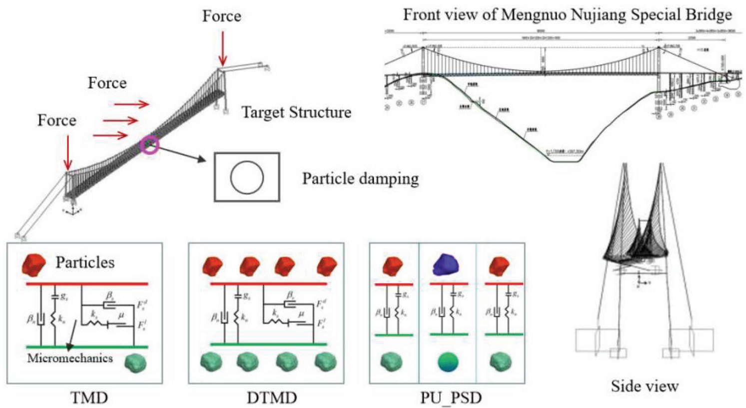

Figure 1 Macro-mechanical and micro-mechanical illustrations of particle dampers.

Figure 2 Macro-mechanical and micro-mechanical illustrations of particle dampers.

2 Establishment of PU_SPD Mechanical Model

Based on the experimental test results of the bridge, we can obtain the ground vibration acceleration response spectrum and the seismic time domain curves of the project site with different absorption stiffness. As shown in Figures 3 and 4, it can be found that the vibration of the New River Bridge is influenced by the structural stiffness, structural aging process, etc. Among them, the peak vibration acceleration produces significant differences due to the degradation of the stiffness and damping structure in different periods within 0–20 Hz. In the stiffness range of 10∼70 KN/mm, the vibration velocity of the beam gradually decreases. Therefore, PU_SPD can be combined with structural stiffness and damping optimization to better control the vibration of the vibrating body.

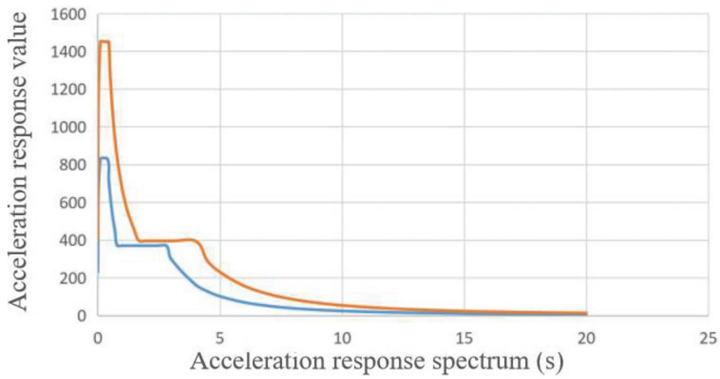

Figure 3 Vibration acceleration response spectrum under different working conditions.

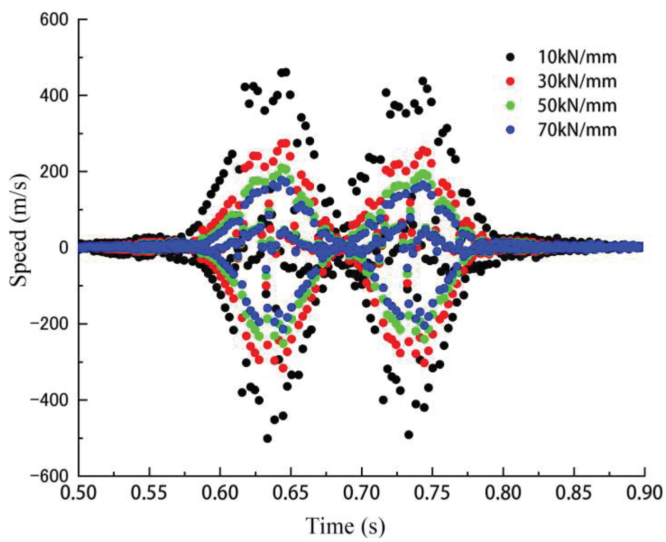

Figure 4 Earthquake time range function.

Based on the study of particle motion characteristics, damping dissipative mechanics, and performance in PU_PSD structure, the mechanical model of PU_PSD is established as shown in Figure 5. Where m,c,k are the mass, damping coefficient, and stiffness of Vibrating body, mp is the particle mass, ag is the excitation acceleration, are the displacement, velocity, and acceleration of Vibrating body, are the displacement, velocity, and acceleration of the particle, and d is the particle motion spacing. The effect of particle

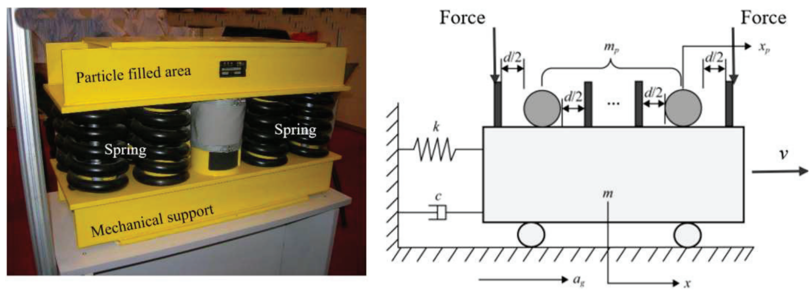

Figure 5 Macro-mechanical and micro-mechanical illustrations of PU_PSD dampers.

damping mainly depends on parameters such as the type and nature of the particle material, the filling density of the particles, the distribution mode of the particles, and the size and shape of the particles. The reasonable selection and control of these parameters can maximize the damping effect of the bridge structure.

Although there is a high degree of nonlinearity between the particle and Vibrating body in the process of impact in PU_PSD, if the impact mechanism between the particle and the vibrating body and the non-complete friction are taken as the external forces. And the vibrating body is in the elastic-plastic state. As the external force acting on the vibrating body excited by the dynamic light behavioral characteristics and state response can be continuously superimposed, where the cumulative process of displacement and acceleration is shown in Figure 6. Here the excitation or impact contact time between the particle and the vibrated body is very short. Then the impact effect between the two can be equal to a certain period of the particle motion effect due to the particle radiation spacing. Therefore, the equivalent pulse force has a certain delay time. The specific action process is shown in Figure 6. In the figure, S_0 is the pulse volume when the particle collides with Vibrating body, which characterizes the amplitude of the pulse force when the particle collides with Vibrating body, and the value depends on Vibrating body, the mass of the particle and the velocity before the impact. $\delta(t)$ is the unit pulse function, T is one period of the simple harmonic motion set by the particle, τ is the phase of this applied excitation force and j is the j_{th} period. Choosing the optimum spacing of the particles to be twice the distance from the outside of the particles to the wall aims to maximize the damping effect and to establish proper energy transfer and dissipation paths between the particles. This configuration can help reduce the vibration response of the structure and provide more effective vibration control and noise mitigation.

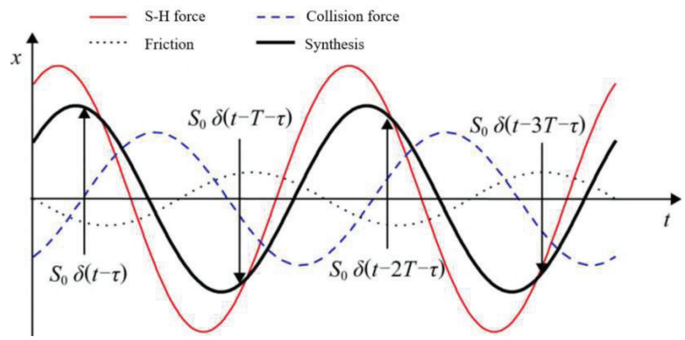

Figure 6 Superposition curve of PU_PSD mechanical motion.

Through the above analysis, the vibration formulas of Vibrating body system with PU_PSD attached is

$$\begin{cases} m\dfrac{\partial^2 x}{\partial t^2} + c\dfrac{\partial x}{\partial t} + kx = -ma - S_0 \displaystyle\sum_{j=0}^{\inf} \delta(t - jT - \tau) - F_s \\ m_p\dfrac{\partial^2 x}{\partial t^2} = S_0 \displaystyle\sum_{j=0}^{\inf} \delta(t - jT - \tau) + F_s \end{cases} \tag{1}$$

Where e is the impact recovery coefficient between the particle and Vibrating body; Fs is the force required for the particle to overcome the rolling friction. r is the radius of the particle, g is the acceleration of gravity, and n_p is the number of particles. The advantage of the PU_PSD mechanical model constructed in the formula is that it is not only easy to solve dynamic behavior characteristics and state response of PU_PSD, but also can reveal the vibration reduction mechanism of PU_PSD. Which is not only the improvement of the mechanical model solution method but also the full embodiment and deep theoretical investigation of the damping dissipative mechanics of PU_PSD. In the following section, the mechanical model will be solved to illustrate its characteristics of easy solving.

When considering the external load simple harmonic excitation response condition. The response of Vibrating body under simple harmonic excitation contains two parts: its first term represents the decaying vibration of the structure, which decays after some time; its second term represents the forced vibration of the structure, and generally only the Vibration in base modeof the structure under simple harmonic excitation is considered, whose value can be characterized by the displacement amplification factor, i.e.

$$|H(\omega)| = \frac{1}{\sqrt{(1-\lambda^2)}} \tag{2}$$

3 Transformation and Solution of Dynamical Equations

For the characteristics of the vibration equations of the constructed PU_PSD mechanical model, it is difficult to solve them by the time domain analysis method, and it is more convenient to solve them by the frequency domain method. To facilitate the frequency domain solution, After the transformation of the equation, which has the following form

$$\begin{cases} ms^2 + 2\zeta\omega sX + \omega^2 X = -A_g - \dfrac{F_s}{m} \\ m_p s^2 + \dfrac{S_0}{m}\displaystyle\sum_{j=0}^{\inf} e^{(t-jT-\tau)} = \dfrac{F_s}{m} \end{cases} \tag{3}$$

In the above results, the pulse amount S_0 is not known because although the masses of Vibrating body and the particle are known, the operating power of the two before the impact are not obtained. Although it is difficult to obtain the velocities before the impact when the structure and the particle are not in a steady state. The dynamic particle state vector that can be derived by time-frequency transformation analysis. The derivation of Equation (3) yields Vibration speed power of Vibrating body as

$$s(x,\omega) = \frac{L}{2\pi}\int_{-\frac{\pi}{L}}^{+\frac{\pi}{L}}\int_0^L \widetilde{F}(\widetilde{x},\omega)H(\omega)\exp(t-jT-\tau)d\widetilde{x}_p d\kappa \tag{4}$$

The velocity composition of a vibrating body is determined by its dynamic behavior, the amplitude and frequency of the excitation, and the impulse effect caused by particles colliding with the vibrating body. Figure 7 illustrates the value of E, which represents the cumulative impact of the particle and vibrating body on the vibrating body's velocity. While all other terms are constants, E is a time-dependent variable that represents the magnitude of the vibrating body's velocity prior to the collision.

The magnitude of E depends on the period of the pulse force generated by the particle's impact, which, in turn, is determined by the collision between the particle and the vibrating body. Hence, the value of E can be used to

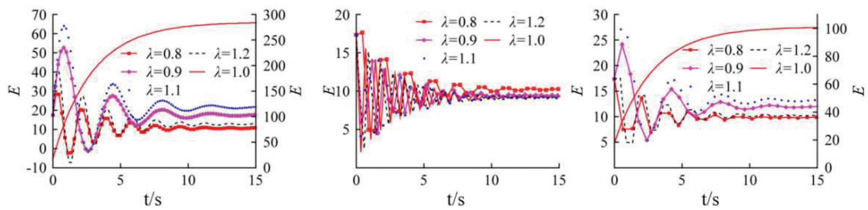

Figure 7 Calculation results of E value of mechanical parameters.

characterize the damping dissipative mechanics of PU_PSD as a result of momentum exchange through impacts. Furthermore, the value of E also indicates the intensity of dynamic energy exchange.

By substituting the calculated parameters of the coupled model, the trend of the thought closing of E in the dynamic impact time domain can be derived. The schematic diagram of the results is shown in Figure 8. After analysis, it can be seen that when the pulse period is taken as a half multiple of the structure period, the E value keeps growing at the resonant frequency. But there is a limit. When the pulse period is taken as a multiple of the excitation period, the E value shows a fluctuating trend. The fluctuating trend indicates that there is a chaotic motion state between the particle and Vibrating body in the impact time course, which is not considered in this paper. When the particle arrives at steady state motion from rest, the particle velocity will reach the maximum, so when seeking the impact velocity of the structure, E should take the maximum value E_{Max}, and it can also be found from the figure that the damping effect is best when T is taken 1/2 times T_1.

Based on the above analysis, when the PU_PSD single-degree-of-freedom structural system reaches Vibration in base modeunder simple harmonic excitation, the velocity of Vibrating body after the impact is

$$\frac{\partial^2 x^+}{\partial t^2} = \frac{p_0\omega}{\sqrt{C^2\omega^2 + D^2}} - \frac{S_0 E_{\max}}{m\beta\omega_n} \tag{5}$$

The classical mechanics method can characterize the non-exact plastic impact process between the particles and Vibrating body in PU_PSD. From the definition of the recovery coefficient. Equation (5) can be used to obtain the velocity of the PU_PSD single-degree-of-freedom structure system before and after the impact between Vibrating body and the particle in the steady state under simple harmonic excitation. The displacement response of the uncontrolled structure in the steady state vibration under simple harmonic excitation is obtained in Equation.

$$x_{st}(x,\omega) = p_0(\omega)H(\omega)\exp(2\pi\omega t) \tag{6}$$

Then Vibration speed power of the uncontrolled structure during Vibration in base modeunder simple harmonic excitation results in

$$x_{st}(x,\omega) = 2\pi\omega p_0(\omega)H(\omega)\exp(2\pi\omega t) \tag{7}$$

Since the impact damper only suppresses the vibration of the structure by providing periodic pulse force to Vibrating body, it does not change the spectral characteristics of the structure, so the moment when the vibration velocity of Vibrating body and the uncontrolled structure have the same maximum value, both are

$$t_0 = \frac{(2k+1)\pi - 2\alpha}{2\omega} \tag{8}$$

Therefore, it can be obtained that the damping effect of PU_PSD is best when the pulse period T is taken as 0.5T1. This is because the peak velocity appears twice at each cycle of the forced cycle, i.e., the particle impact with the vibrating body is generally characterized twice. The period of the pulse $T = \pi/\omega$ The energy dissipation is most adequate in this case. Let k = 0, we can obtain $\tau = (\pi - 2\alpha)/2\omega$ For the response of Vibrating body generated by friction, the rolling friction coefficient μ can be determined from the material of the particle, the material of the damper cavity, and the roughness of the contact surface between the particle and Vibrating body, and then the response time of this part can be calculated by combining dynamic behaviors of Vibrating body. The relevant parameters are shown in Table 1. Based on the above parameters, the synthetic response of Vibrating body and the particle in the steady state under simple harmonic excitation can be obtained, and the above single-layer steel structure is still used as an example for analysis in this paper.

The actual response superposition results under the working conditions of this example, as shown in Figure 8. This result verifies the correctness

Table 1 Table of key parameters of mechanics

Parameters	Numerical Value	Parameters	Numerical Value
μ	0.02	e	0.08
μ_f	0.05	r	45
λ	0.95	P0	0.25

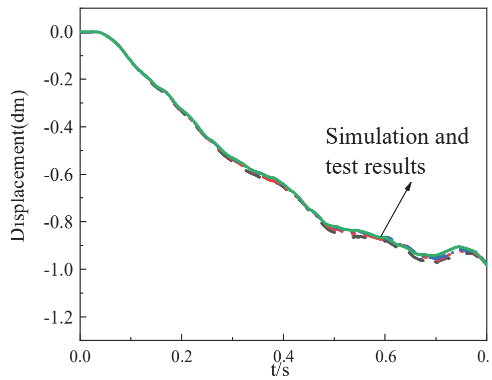

Figure 8 Comparison of test and simulation results.

and scientificity of the conclusion of the superposition of the vibrating body state vectors. It also verifies the reasonableness and efficiency of the vibration damping model involved in this paper.

The time domain displacement response of Vibrating body and the particle in Vibration in base modecan be obtained by analyzing the above-mentioned partial parameters, and the difference between the two displacements can be used to obtain the motion spacing of the particle under the optimal damping effect of PU_PSD, i.e., the optimal spacing of the particle is two times the distance from the exterior of the particle to the wall, as shown in Equation (9).

$$d_{op} = 2\max(x - x_p) \tag{9}$$

4 Parametric Analysis of the Vibration-Damping Effect

Although dynamic behavior characteristics and state response of the PU_PSD structure in steady state is solved by combining the frequency domain and time domain analysis methods. The results obtained are the optimal damping effect of PU_PSD, it still cannot visualize how the parameters affect the damping performance of PU_PSD. The influence law of each parameter can be clarified only by the change of the transfer function of Vibrating body with each parameter. Specifically, the pulse phase, mass ratio, recovery coefficient, frequency ratio. excitation amplitude variation need to be investigated. The influence of the parameters on the transfer function of the displacement response of Vibrating body.

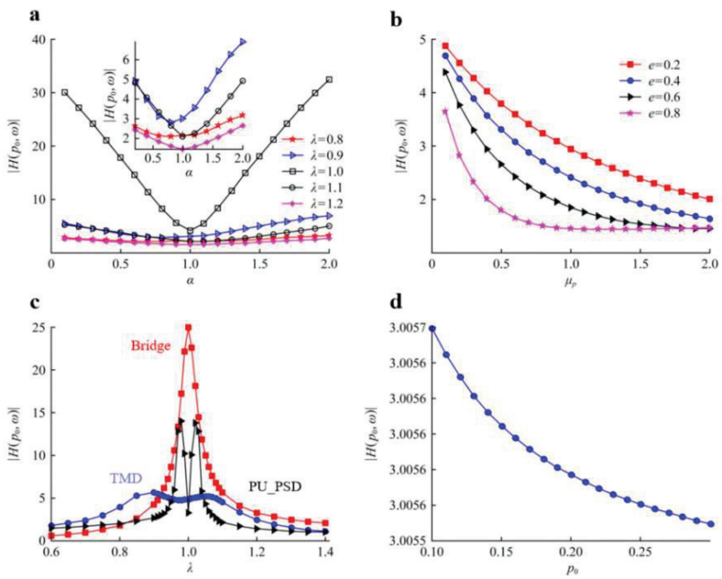

Figure 9 Effect of mechanical parameters on the displacement amplification factor (a) phase (b) mass ratio (c) frequency ratio (d) amplitude.

From Figure 9 it can be seen that the displacement amplification coefficient of Vibrating body is minimized when $\alpha = 1$, i.e., there exists an optimal pulse phase of τ, which has been described in the previous section. In Figure 9, the displacement amplification coefficient of Vibrating body decreases with the increase of Particle quality, and finally, it smooths out. It can be seen that the relationship between the velocity and mass of the particle is inversely proportional, and the expression of S_0 shows that its value is proportional to the product of the velocity and mass of the particle, so S_0 has a limit value with the increase of particle mass, so the curve finally smooths out. When other conditions are certain, the displacement amplification coefficient of Vibrating body decreases with the increase of the recovery coefficient, because the larger the recovery coefficient, the larger the impulse force between Vibrating body and the particle in the impact process, which also shows that the momentum exchange between Vibrating body and the particle dominates the damping dissipative mechanics of PU_PSD. According to the comparison of the damping effect of PU_PSD and TMD, it can be seen that the damping rate of TMD is 72.4% and that of PU_PSD is 24.07% when the frequency ratio λ is 0.95∼1.95, and the damping rate of TMD is better than that of PU_PSD. When λ is 0.8∼0.95 and 1.05∼1.4, the damping rate of PU_PSD is up to 60% and down to 12.5%, while the damping rate of TMD is low, and the response of Vibrating body is amplified when λ is 0.8$\sim$0.95. A comprehensive analysis of Figure 9(c) shows that PU_PSD has the advantage of a wide damping band compared with TMD. The main reason for this phenomenon is that there is an essential difference between the damping dissipative mechanics of PU_PSD and TMD-tuned damping device, the particles have no fixed frequency, and the impact with Vibrating body will transfer its energy, thus reducing the vibration of Vibrating body, and thus the damping band is wide, while the suppression effect of TMD mainly depends on the tuning relationship between the structure frequency and the external load frequency, and the damping band is narrow. In addition, the random dynamic load (wind load and ground vibration) on the structure usually has the characteristics of strong randomness and rich spectrum, and the possibility of resonance is small. Compared with TMD, PU_PSD has stronger robustness in practical damping applications. Under the optimal parameters, the damping effect of PU_PSD increases with the increase of the excitation amplitude, as shown in Figure 9(d), but the increase of the damping effect is small, which indicates that the damping performance of PU_PSD is stable.

In summary, through the parameter analysis, it is proved that when the impact between the particles and Vibrating body in PU_PSD is equivalent to the pulse force and its damping effect is optimal, the pulse phase is reasonable, and the damping dissipative mechanics of PU_PSD is clearer, the transfer function of Vibrating body of PU_PSD can reflect the change law of the damping effect of PU_PSD with each parameter.

5 Numerical Validation of the Mechanical Analytic Model

To verify the accuracy of the above mechanical model and the correctness of the optimal parameters of the damping effect, it is necessary to numerically verify its performance by numerical analysis. The more detailed analysis process will not be discussed in this paper. After verifying the accuracy and optimality of the mechanical model, the optimized analysis method in ground vibration will be further proposed.

It is obvious from Figures 10 and 11 that the structure designed in this paper has a good damping effect. The relative error values of the peak displacements calculated in the frequency and time domains and the peak

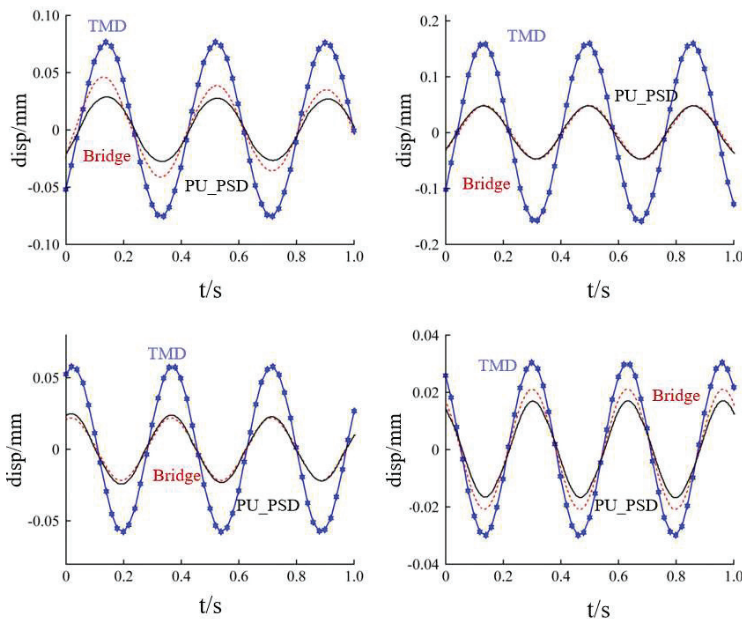

Figure 10 Comparison of dynamic damping characteristics under four operating conditions.

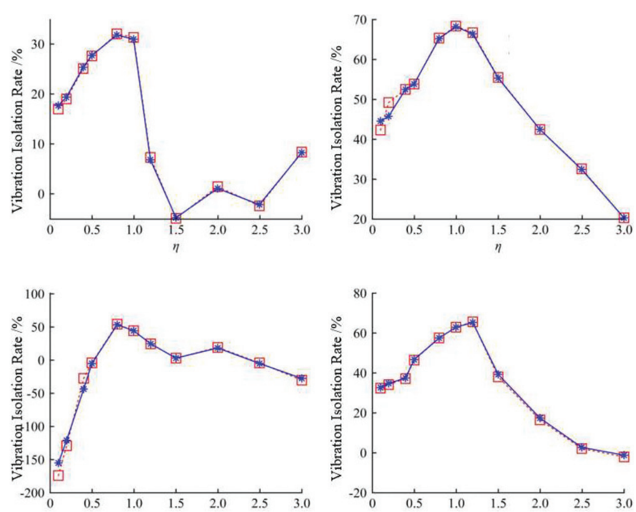

Figure 11 Root mean square comparison of vibration isolation rates for four operating conditions.

displacements of the numerical simulation are further obtained. This shows that dynamic behavior characteristics and state response of the two overlap at resonance. It can be seen that the numerical results are slightly higher than the theoretical results. (2) theoretical analysis assumes that the velocity of the particle before and after the impact does not change, but the direction is opposite to that before the impact, while the velocity of the particle after the impact with Vibrating body in the PU_PSD numerical simulation is obtained by the conservation of momentum and recovery coefficient, and the velocity of the particle after the impact in the numerical simulation is slightly less than that before the impact. (3) The theoretical analysis assumes that the impact between the particle structure and the vibrated body usually occurs at the peak velocity. When in addition for the most numerical aspects, the difference is small.

Since dynamic behavior characteristics and state response of the PU_PSD damping structure is subject to periodic bifurcation and chaos, the damping effect is evaluated by using the root mean square damping rate as the evaluation index, where the damping rate is the difference between the root mean square of the displacement (velocity) before damping and the root mean square of the displacement (velocity) after damping divided by the root mean square of the displacement (velocity) before damping. The variation law of the damping rate of Vibrating body with η under four operating conditions is shown in Figure 11. The variation of the damping rate with η for the four operating conditions is shown in Figure 11. Where the blue line is displacement and the red block is velocity.

To further analyze the damping dissipative mechanics of PU_PSD and compare the damping effect with that of TMD in the frequency domain, the frequency power spectrum analysis of uncontrolled structure, controlled structure and particle displacement, and ground vibration acceleration is carried out, and the analysis results are shown in Figure 11. The analysis shows that the particles can transfer and dissipate the energy of Vibrating body in a wide range of frequency bands, which verifies that PU_PSD has the advantage of wide vibration-damping frequency bands. It is also found from Figure 11 that the damping effect of PU_PSD on the displacement and velocity peaks is better than that of TMD when the additional mass ratio is greater than 0.4 under the action of S1 wave, while the displacement peak of TMD on S2 wave is amplified. This is because the spectral components of the S1 wave are richer than those of S2 and S3 waves, and the damping frequency band of TMD is narrow, while the damping frequency band of

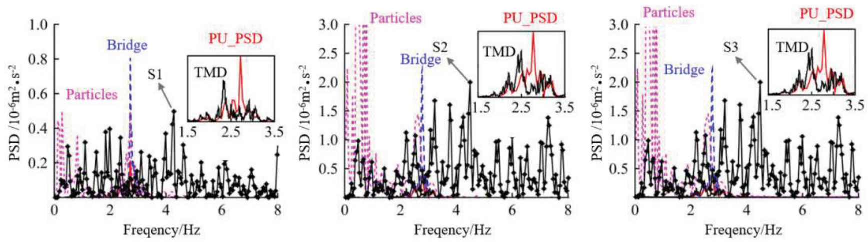

Figure 12 Frequency domain displacement-power spectrum under kinetic effects.

PU_PSD is wide and the power load spectrum is rich, which is more favorable to its damping performance. Since PU_PSD is a force (acceleration) dependent damper, which is different from the mechanism of tuned dampers. The damping performance of this damper in the multi-degree-of-freedom damping system is more dependent on the interlayer acceleration time course and less influenced by the multivibrator type.

6 Conclusion

(1) Analysis of PU_PSD Damped Dissipative Mechanics: The bridge body and particle absorber are simulated as micro/macro models, demonstrating that the PU_PSD mechanical model effectively visualizes the damped dissipative mechanics with practical computational results.
(2) Parameter Analysis of PU_PSD: By analyzing parameters such as pulse phase, mass ratio, recovery coefficient, frequency ratio, and excitation amplitude, the optimal damping effect of PU_PSD is determined. The transfer function of the vibrating body with PU_PSD accurately reflects the damping effect variation with each parameter.
(3) Parameter Optimization Analysis Method: A parameter optimization analysis method is proposed for PU_PSD under simple harmonic excitation and ground shaking, validated through numerical simulations and compared with pure time-domain analysis, confirming its rationality, feasibility, and accuracy.
(4) Comparative Analysis with TMD: The damping performance of PU_PSD and TMD is extensively analyzed under simple harmonic excitation and ground vibration. The study elucidates the reasons for the difference in damping effects and demonstrates that PU_PSD, as a force-dependent damper, offers a wider damping band than TMD, making it advantageous for practical engineering applications.

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Biographies

Gong Lei born in Wuhan, Hubei Province, graduated from the Department of Road, Bridge and River Crossing Engineering of Wuhan University of Technology. He is now a senior bridge designer of CCCC Second Highway Survey, Design and Research Institute. He has participated in the construction of many sea crossing bridges, such as Hong Kong Zhuhai Macao Bridge and Ningbo Xiangshan Port Bridge. He has also served as the project leader and technical director of several design projects, such as Wuhan Shenzhen Expressway, Sichuan Wenma Expressway, G5011 Wuhe Hefei Expressway Reconstruction and Expansion, G40 Hefei Expressway Reconstruction and Expansion, G5 Chengdu Mianyang Expressway, G5615 Astronomical Expressway, etc, He has rich experience in expressway design, construction and construction.

Tang Miao born in jingzhou, Hubei Province, graduated from the Department of Bridge Engineering of HU NAN University. He is now a senior bridge designer of CCCC Second Highway Survey, Design and Research Institute. He has served as the project leader and technical director of several design projects, such as Wuhan Shenzhen Motorway Project, Sichuan Wenma Motorway Project, Peshawar – Karachi Motorway Project (PKM) of PAKISTAN, Southern Coastal Highway Improvement Project of JAMAICA (SCHIP), Montego Bay Perimeter Road Project of JAMAICA, etc, He has rich experience in expressway design.