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A Nonlinear Damper with Dynamic Load and an Elastic Slit Membrane: Modeling and Interaction Analysis

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Abstract: This article presents research into the feasibility of applying a nonlinear damper of a new conceptual structure. The key component of the damper is a circular membrane with slits that can move in a cylinder filled with viscous fluid. When an external load is applied to the damper, the membrane deforms, opening the slits. The flow of viscous fluid through the slits generates a damping force. The phenomenological model of the damper is based on the notion that the slit membrane moves according to the fundamental axisymmetric vibration mode of a circular membrane. The slit membrane blocks the entire radius of the pipe in the state of equilibrium when all slits are closed. As the membrane moves, the opening area of the slits varies depending on its deformation. This gives a nonlinear damping characteristic. The damping constant depends on the input displacement and velocity, which is the reason for the nonlinearity of the damping characteristic. From the phenomenological model, the nonlinear characteristic of the drag force is obtained. The performance of the damper is simulated using a mass–spring–damper system. Two cases of harmonic excitation and impulse excitation are analyzed. The results show that, using the slit membrane damper, the suppression of dynamic loads is more effective compared to a conventional linear damper.

Keywords: damper; nonlinear; membrane; slit; viscous

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1. Introduction

Dampers play an important role in achieving the optimum performance of dynamic systems. They are key components that ensure the effective suppression of shocks and other dynamic loads. This research deals with the development of a nonlinear damper with an original structure. The proposed damper is a passive damper with a simple structure; therefore, there is no need to apply a sophisticated control system to achieve the nonlinearity of its damping characteristic. In order to achieve effective operation of the damper, the dynamic properties of its main components need to be studied in detail.

The most common type of damper is the hydraulic damper. Dampers are energy-dissipating elements that absorb kinetic energy, making them an effective means of controlling the rate of motion of mechanical components. The main elements related to the reduction in the rate of movement are as follows: reducing the vibration amplitude, preferably by reaching the critical damping ratio of the dynamic system; softening the impact effect by absorbing the impact energy; contributing to the precision of the movement by stabilizing the movement parameters, i.e., enabling even/smooth changes in the movement coordinates, velocities and accelerations; and regulating the movement velocity and acceleration, thus preventing the occurrence of high-inertia loads.

Vehicle suspension systems are the most common applications for dampers. Here, the dampers act in synergy with the elastic elements and masses of the vehicle body and other structural components to ensure proper vehicle handling and ride comfort. Another common application is in civil engineering, in the construction of high buildings, bridges and towers subjected to random seismic or atmospheric (wind) excitation. Dampers are also used for vibration reduction and shock absorption in aerospace and railway systems. In technological machines, including robotic systems, dampers serve as important components in achieving motion precision and regulating motion parameters. Research into damper performance is an ongoing and intensive process.

The simplest systems for modelling and simulation are linear systems, including linear dampers. However, better performance characteristics can be achieved by using nonlinear systems. Nonlinear dampers have variable damping characteristics; usually, their damping constant depends on the input displacement and velocity. This can be observed as a hysteresis loop in the damping force–displacement and damping force–velocity relationships.

An important direction for achieving nonlinearity in mechanical systems is the development of nonlinear passive dampers.

P. S. Balaji & K. Karthik SelvaKumar [1] carried out a thorough review of passive vibration control with the aim of mitigating the response of dynamical systems to external excitation. Various configurations in mechanical systems that give geometrical nonlinearities are reviewed. Nonlinear dampers, including dry friction (Coulomb damping), viscous nonlinear dampers, and the application of smart materials for damping and energy harvesting are reviewed. It is concluded that linear vibro-isolation systems are effective in the narrow-frequency range, but for the broadband low-frequency spectrum (e.g., shocks and random excitation), nonlinear vibro-isolation systems are more effective. Zheng Lu et al. [2] present a comprehensive review of nonlinear dissipative devices for vibration control. The limitation of linear dampers is their sensitivity to variations in excitation conditions due to their narrow bandwidth. For nonlinear dampers, however, this is eliminated due to their broadband response. The authors classify nonlinear dampers into three groups: nonlinear stiffness dampers, nonlinear-stiffness nonlinear-damping dampers, and nonlinear damping dampers. In this article, cylinder-type dampers with a special configuration of the orifice path for fluid flow and shear-type dampers in the form of a viscous damping wall are studied. Typical applications of the damper in building structures to resist earthquakes and as a shock absorber in a vehicle are discussed. Pazooki et al. [3] proposed an analytical technique for the frequency analysis and design of nonlinear dampers to further improve vehicle dynamic performance in automotive applications. Using the Energy Balance Method (EBM), a methodology for determining the equivalent linear damping coefficient of any nonlinear damper was developed. Using the quarter car model, the design principles of position-dependent, velocity-dependent, and position–velocity-dependent dampers were proposed.

A number of attempts have been made to develop nonlinear dampers in which the gap for viscous fluid flow varies with the stroke of the piston. In this way, a nonlinear damping characteristic is obtained. H. Saber et al. [4] developed a nonlinear variable damping device called the variable damping viscous dashpot (VDVD) for use as a bridge vibration absorber. This device is insensitive to frequency detuning effects. The basic idea is that two conical surfaces with a common axis—inner and outer—form the gap for fluid flow. When the surfaces are axially shifted, the gap for fluid flow changes. Kalyan Raj and C Padmanabhan [5] proposed a nonlinear damper based on a cylindrical piston moving in a conical cylinder. The results of experimental research on a nonlinear damper for a twin-tube motorcycle shock absorber presented in [6] identify phenomena in the damper caused by external force, frictional forces in the main cylinder and compensation cylinder, and the compliance of the seal of the rebound chamber. In [7], a study of large-scale nonlinear viscous dampers for application in earthquake-resistant building structures is presented. The concept of a cylindrical damper with a stroke-dependent annual gap between the piston and the cylinder for fluid flow is presented by Weizhi Xu [8]. Hysteresis behavior under different excitation conditions and significantly higher energy dissipation compared to linear fluid dampers were proved.

Another intensively researched way to achieve the nonlinearity effect in damping is the development of control strategies for active/actively controlled systems. Min Cheng et al. [9] present an active damping method to minimize velocity overshoot and achieve an optimal damping ratio at shock loading by introducing a control valve into a hydraulic system with a symmetrical cylinder. Using a multi-objective optimization method, the control parameters (gain and time constant) were determined, which allowed the improved damping ratio and reduced dynamic impact. H. Farahpour and F. Hejazi [10] developed an Adjustable Bypass Fluid Damper (ABFD) with a pair of external fluid-flow tubes and flow-control valves to control the fluid flow. This allowed the damping function to be adjusted according to the displacement of the structure.

The other way to achieve damping nonlinearity is by using semi-active dampers based on the application of controllable parameter fluids, such as magnetorheological (MR) fluids. By applying a magnetic field, the viscosity of these fluids is changed, enabling the diversity of their application, e.g., for dampers, batteries, valves and brakes. H. Eshgarf et al. [11] present a thorough review of MR fluids, detailing their composition, preparation and operation principles, and multiple application cases, including MR dampers with controllable damping characteristics. Another review of MR technology for dampers was carried out by P. N. Vishwakarma [12], in which a cylinder-shaped MR damper with variable damping is discussed in detail.

Jiang et al. [13] developed a new phenomenological model to describe the nonlinear behavior of magnetorheological (MR) dampers with fluid deficiency. The damper analyzed is a multi-channel bypass damper with an MR valve that controls the damping modes of the damper by controlling the bypass flow. The typical application of MR dampers is in semi-active vehicle suspension systems. S. Kumbhar [14] characterized a magnetorheological fluid damper and performed a quarter car analysis with it. Research on the control of the MR damper for two-wheeler applications was carried out by Devikiran et al. [15].

An example of the application of MR dampers in technological machinery is given in the research work carried out by M. Emami et al. [16]. The effect on chatter vibration suppression in straight turning operation is investigated using Multiphysics simulation software (Version: COMSOL 5.5) for the process simulation. The tests proved the efficiency of the developed damper. The Multiphysics coupling simulation model and the circuit simulation model were used by G. Hu et al. [17] to develop MR dampers for anti-seismic buildings. The applied multi-objective structural optimization enabled the improvement of the output damping force by 6.6% and the adjustable range of the damper by 46.3%. Aziz et al. [18] integrated computational fluid mechanics (CFD) and finite element analysis (FEA) to investigate the dynamic characteristics and the controllable damping force of shear-mode MRP (magnetorheological plastomer) dampers.

A number of studies present the development of nonlinear dampers using shear thickening fluids (STFs). The viscosity of these fluids increases with the increase in shear rate. Article [19] presents an analysis of the structure and the phenomena of shear thickening in fluids, provides a phenomenological model of STFs, and gives examples of typical applications, including personal protection, preventing spike puncture, cutting knives, impact suppression and vibration control. M. Wei et al. [20] present a phenomenological model showing the relationship between complex viscosity and dynamical shear rate. The effectiveness of STF is demonstrated in a cylinder-shaped damper with STF. The particular features of the damper are that after reaching the critical value of shear rate, the damping force stops increasing and the damper is sensitive to the annular gap. P. Nagy-György & C. Hos [21] developed an analytical model to predict the damping characteristics of dampers with non-Newtonian fluids, including STF. An analysis based on laminar Poiseuille flow was performed to study the annual gap and holes in the piston of a cylinder-shaped damper. The analytical solution was validated by CFD analysis.

The effectiveness of the damper was demonstrated on a single-degree-of-freedom (DOF) mass–spring–damper system by plotting its amplitude frequency characteristics.

In summary, it can be stated that with the application of nonlinear dampers, damping effectiveness in dynamical systems can be achieved over significantly wider excitation frequency ranges than with linear systems.

Nonlinear systems are developed as passive and active (or semi-active) controllable systems. Typically, passive systems include several interacting components of complicated geometry. They can be very effective, but their development needs innovative solutions and a deep understanding of dynamical phenomena in order to obtain a precise matching of the structural, physical and mechanical parameters necessary to achieve effective performance characteristics. Active systems are sophisticated systems for which proper control strategies need to be developed.

In this research, the results of the development of a nonlinear passive damper of an original structure named the nonlinear slit membrane damper are presented. The structure is of simple geometry compared with the majority of reported developments of passive and active dampers. Its key component is the slit membrane. The behavior of the membrane under external dynamical loads is the reason for the nonlinear damping characteristic and damping effectiveness. The research results of the article are presented in the following order: the concept of the nonlinear cylinder-shaped slit membrane damper is presented in Section 2; the phenomenological model of the flexible membrane damper with slits, based on the drag force of the damper, is outlined in Section 3; and the performance effectiveness of the damper is analyzed using a case study with a single-degree-of-freedom mass–spring–damper system in Section 4.

2. Concept of Nonlinear Cylinder-Shaped Slit Membrane Damper

The conceptual structure of the slit membrane damper is shown in Figure 1. Like in conventional fluidic dampers, the damping force is obtained as the piston moves in a cylinder filled with viscous fluid. The piston is of axisymmetric structure. It consists of an outer ring, inner circular-shaped flexible membrane with slits, and a piston rod rigidly attached at the center of the membrane. When external excitation is applied to the damper, the membrane deforms causing the slits to open, allowing fluid flow through the slits. The slit opening area is dependent on membrane deformation, which in turn is dependent on external excitation load. The variation in the opening area during a dynamical process results in the condition of nonlinearity of the damping characteristic. Therefore, damping effectiveness depends on the dynamical properties of the membrane.

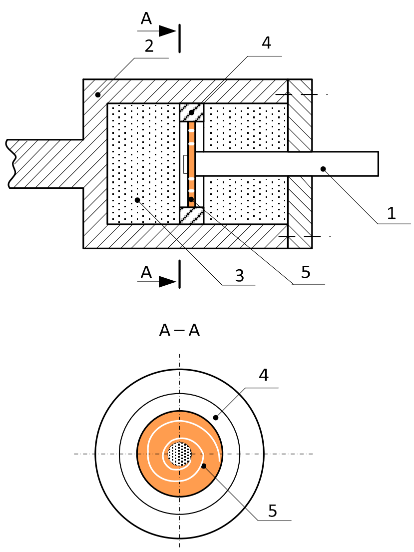

Figure 1. Conceptual structure of slit membrane damper: 1—rod, 2—cylinder, 3—viscous fluid, 4—piston's ring, 5—slit membrane.

3. Phenomenological Model of Flexible Membrane Damper with Slits

The phenomenological model of the flexible flat circular membrane with slits is derived based on the following assumptions:

1. The flexible membrane of the damper is flat and circular in the state of equilibrium.
2. The flat circular membrane with slits moves in accordance with the fundamental axisymmetric vibration mode.
3. When the flat circular membrane is deflected, the slits open. The area of the open slits is equal to the difference between the area of the deformed membrane and the area of the circular-shaped non-deformed membrane.
4. The flat circular membrane with slits blocks the whole cylinder in the state of equilibrium (all slits are closed).

Further, following the sequence of assumptions made, the modelling of the damper is carried out.

3.1. Vibrations of the Circular Membrane

Consider the vibrations of the circular membrane with radius a in the axisymmetric case. The deflection of the membrane from the state of equilibrium u does not depend on the angle, and the wave equation reads as follows:

$$\frac{\partial^2 u}{\partial t^2} = c^2 \left(\frac{\partial^2 u}{\partial r^2} + \frac{1}{r} \frac{\partial u}{\partial r} \right) \tag{1}$$

where t is time; r is the radial coordinate; and c is the wave speed. The solutions in the separable variables $u(r,t) = R(r)T(t)$ read as follows:

$$T''(t) = Kc^2 T(t) \tag{2}$$

$$rR''(r) + R'^{(r)} - KR(r) = 0, \tag{3}$$

where K is a constant. The equation for $T(t)$ has periodic solutions for $K < 0$. Assuming $K = -\lambda^2$ yields the following:

$$T(t) = A\cos c\lambda t + B\sin c\lambda t \tag{4}$$

The equation for $R(r)$ is a special case of Bessel's differential equation. After dropping the non-physical part of the solution, $R(r)$ reads as follows:

$$R(r) = J_0(\lambda r) \tag{5}$$

where J_0 is a zero-order Bessel function of the first kind. The fixed boundary conditions (u must be zero on the boundary) yield the following:

$$R(a) = J_0(\lambda a) = 0 \tag{6}$$

For the fundamental natural frequency, we assume that $\lambda a = \alpha_{01}$, where $\alpha_{01} \approx 2.40483$ is the first root of J_0. Thus, finally, the fundamental axisymmetric vibrations of a circular membrane read as follows:

$$u_{01}(r,t) = \left(A\cos \frac{c\alpha_{01}}{a} t + B\sin \frac{c\alpha_{01}}{a} t \right) J_0 \left(\frac{\alpha_{01}}{a} r \right) \tag{7}$$

Note that $0 \le r \le a$.

3.2. Surface of the Deformed Membrane

According to the assumptions of the phenomenological model, the displacement of the circular membrane corresponds to the fundamental axisymmetric modal shape, as shown in Figure 2. The displacement can be expressed as follows:

$$u_{01}(r) = CJ_0\left(\frac{\alpha_{01}}{a}r\right) \quad (8)$$

where C is a real constant and defines the magnitude of displacement from the state of equilibrium.

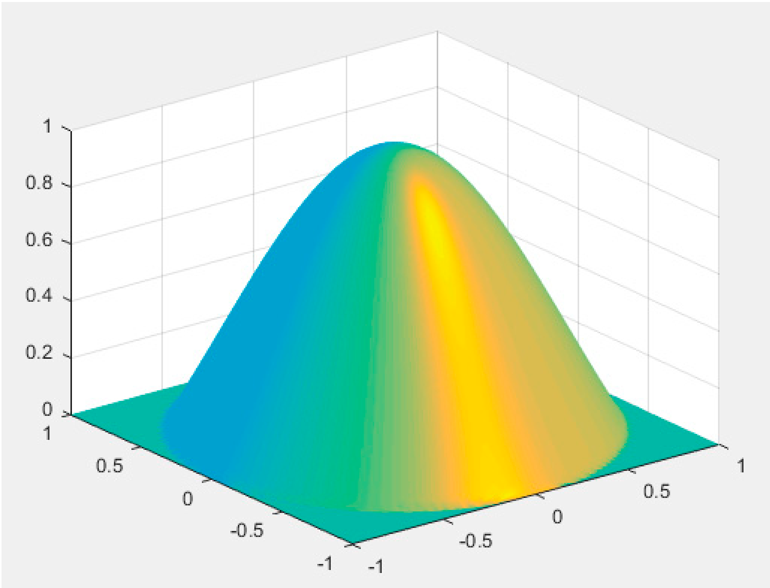

Figure 2. The fundamental axisymmetric modal shape of the circular membrane at $a = 1$ and $C = 1$.

The surface of the deformed membrane (the surface of the revolution) reads as follows:

$$S(C,a) = 2\pi\int_0^a z\sqrt{1+\left(C\frac{dJ_0\left(\frac{\alpha_{01}}{a}z\right)}{dz}\right)^2}dz \quad (9)$$

Note that $\frac{dJ_0\left(\frac{\alpha_{01}}{a}z\right)}{dz} = -\frac{\alpha_{01}}{a}J_1\left(\frac{\alpha_{01}}{a}z\right)$. Therefore,

$$S(C,a) = 2\pi a^2\int_0^1 z\sqrt{1+\left(\frac{C\alpha_{01}}{a}\right)^2 J_1^2(\alpha_{01}z)}dz \quad (10)$$

The area of the membrane in the state of equilibrium equals the area of the cross-section of the tube:

$$S(0,a) = \pi a^2 \quad (11)$$

3.3. Area of the Open Slits

The area of the open slits of the flat circular membrane reads as follows:

$$\delta(C,a) = S(C,a) - S(0,a) \quad (12)$$

The dependence of the area of the open slits on the deformation of the membrane is shown in Figure 3.

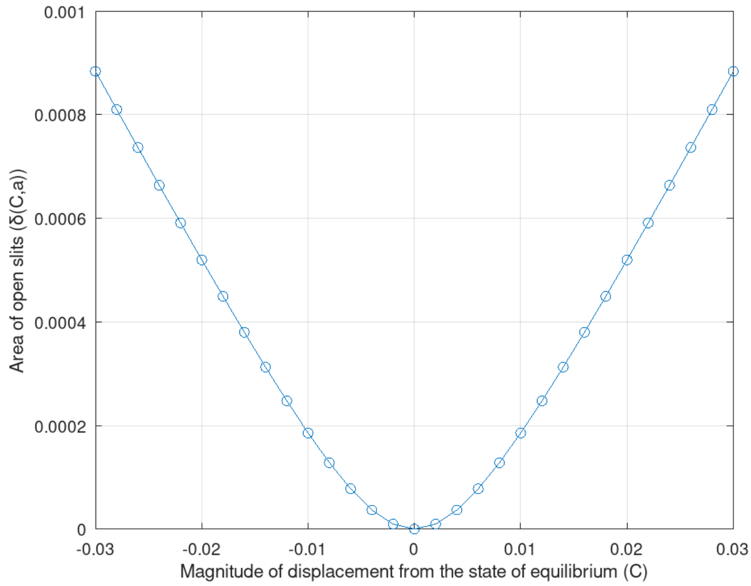

Figure 3. The relationship between the area of open slits $\delta(C, a)$ and C at $a = 0.01$.

4. Mathematical Model of the Damper

4.1. Structure of the Drag Force

The phenomenological model of the drag force is based on several assumptions:

1. If the area of open slits $\delta(C, a)$ is fixed, the drag force is assumed to be proportional to the square of the velocity (due to the high viscosity of the fluid).
2. The velocity determines the shape of the flat circular membrane with slits. The higher the velocity, the larger the area of open slits (the larger the parameter C).
3. The higher the area of open slits, the smaller the drag force.

Note that $S(C, a)$ is an even function in respect to C. Therefore, according to Assumption 2,

$$C = \Omega \dot{x} \tag{13}$$

where Ω is the coefficient of proportionality between C and $\dot{x}$.

According to Assumption 3, the drag force is inversely proportional to the area of open slits. Combining all three assumptions results in the following expression of the drag force D:

$$D(\dot{x}) = h \cdot sign(\dot{x}) \cdot (\dot{x})^2 \cdot \frac{1}{\delta(\theta \dot{x}, a)} \tag{14}$$

This structure of the drag force has a singularity point at $\dot{x} = 0$. Therefore, we add the following fourth assumption:

4. The drag force is not infinite when all the slits are closed (when the flat circular membrane is in the state of equilibrium).

Combining the fourth assumption yields the final phenomenological model of the drag force:

$$D(\dot{x}) = h \cdot sign(\dot{x}) \cdot (\dot{x})^2 \cdot \frac{1}{\delta(\theta \dot{x}, a) + \varepsilon} \tag{15}$$

where ε is a small positive number determining the drag force when all slits are closed.

First, using the above assumptions, properties of the damper were researched. The force–velocity characteristic of the slit damper in comparison with a conventional linear damper is shown in Figure 4. An increased damping efficiency can be observed in the lower velocity range.

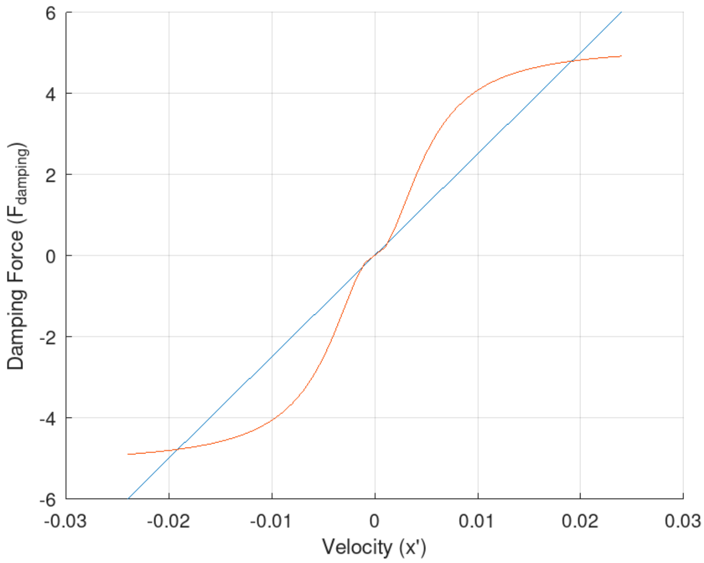

Figure 4. Relationship between drag force and x' at h = 0.02, θ = 0.04, ε = 0.0000001 and a = 0.01. Red line the slit membrane damper, blue line for linear damper.

4.2. Governing Equations of Motion

For the analysis of the damping effectiveness of the slit membrane damper, the single-degree-of-freedom mass–spring–damper model, e.g. the quarter car model (Figure 5) was used.

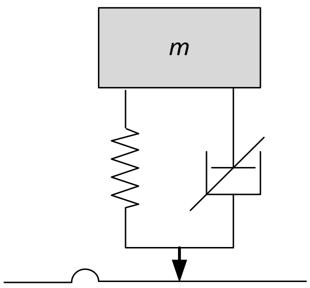

Figure 5. Single DOF mass–spring–damper dynamical model.

Finally, the governing equation of motion for the system shown in Figure 5 reads as follows:

$$m\ddot{x} + D(\dot{x}) + kx = f(t) \tag{16}$$

5. Results and Discussion

Using governing Equation (16) of the system shown in Figure 5, the damping effectiveness of the system with the slit membrane damper was analyzed. The drag force derived in Section 4.2 (Equation (15)) is used in the model.

The typical system parameters for the quarter car model were used: mass m = 250 kg; ε = 0.0000001; a = 0.01 m; h = 0.02; Θ = 0.04; k = 25000 N/mv.

Two cases were analyzed where harmonic excitation and impulse excitation were applied.

The results of harmonic excitation in the form of f = sin wt (f = 100×sin(2π0.5t)) are presented in Figure 6.

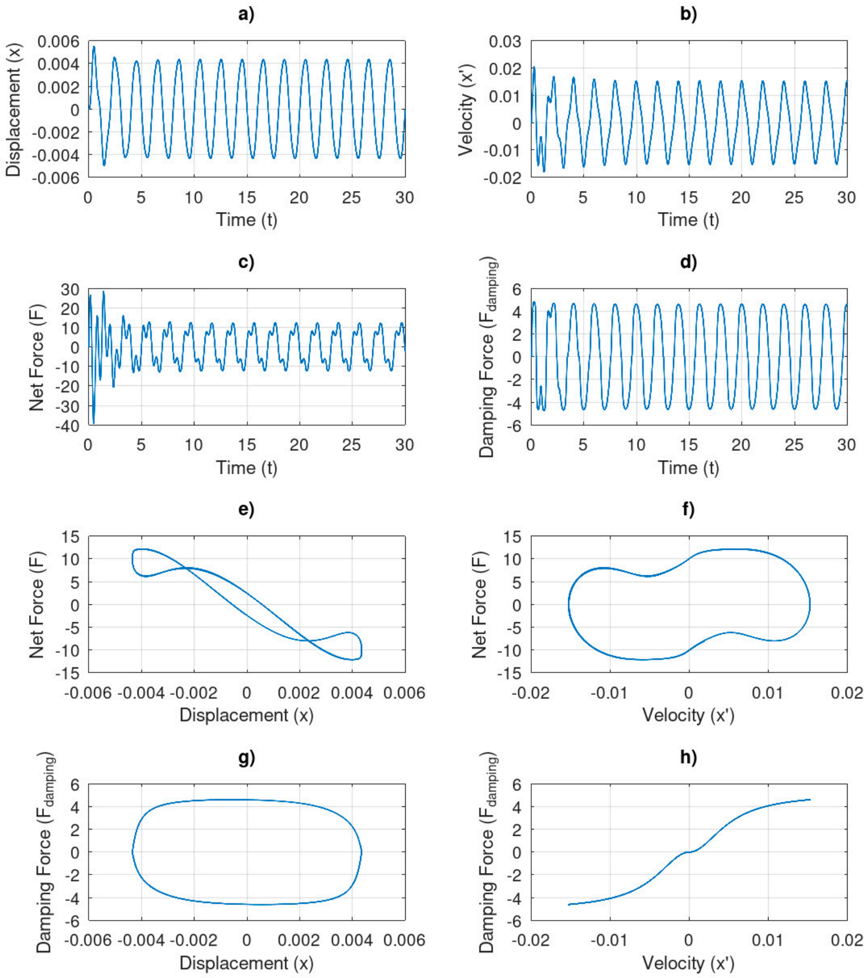

Figure 6. Motion characteristics of the mass–spring–damper system: (**a**) displacement of mass x; (**b**) velocity of mass x′, (**c**) net force acting on mass m; (**d**) damping force; (**e**) net force acting on mass m versus displacement; (**f**) net force acting on mass m versus velocity; (**g**) damping force versus displacement; (**h**) damping force versus velocity.

The nonlinear effect can be observed. We can see clear hysteresis in the graph of damping force displacement (Figure 6g). This can be explained by motion of the membrane due to external excitation. The motion of the excited membrane is described in Sections 3.1 and 3.2. This leads to the slit opening area given by Equation (12). The flow of viscous fluid through the variable-area openings of the slits, associated with the drag force expressed by Equation (15), gives a nonlinear effect in the form of hysteresis. Then, the resultant force acting on mass m is shown in Figure 6e,f.

Also, it is worth noting that at lower displacements an increased damping force is obtained as the opening area of the slits is smaller. The increase in damping force at lower velocities (Figure 6h) can also be explained by the smaller opening area of the slits.

In the second case, the effectiveness of the system with the slit damper was analyzed when shock (impulse) excitation was applied. The results are presented in Figure 7. Here, the system response with a linear damper and the system response with the nonlinear slit membrane damper are compared. The results are consistent with the obtained relationship of drag force versus velocity (Figure 4). At lower velocities, the drag force is higher compared to the linear damper; therefore, we see a lower velocity amplitude (Figure 7b) and an increased damping force (Figure 7c). Thus, conditions are created for more effective shock suppression. A shorter time is necessary for the oscillations to decay. Or, we can say that the nonlinearity of the damping characteristic allows a shorter transient process to be achieved.

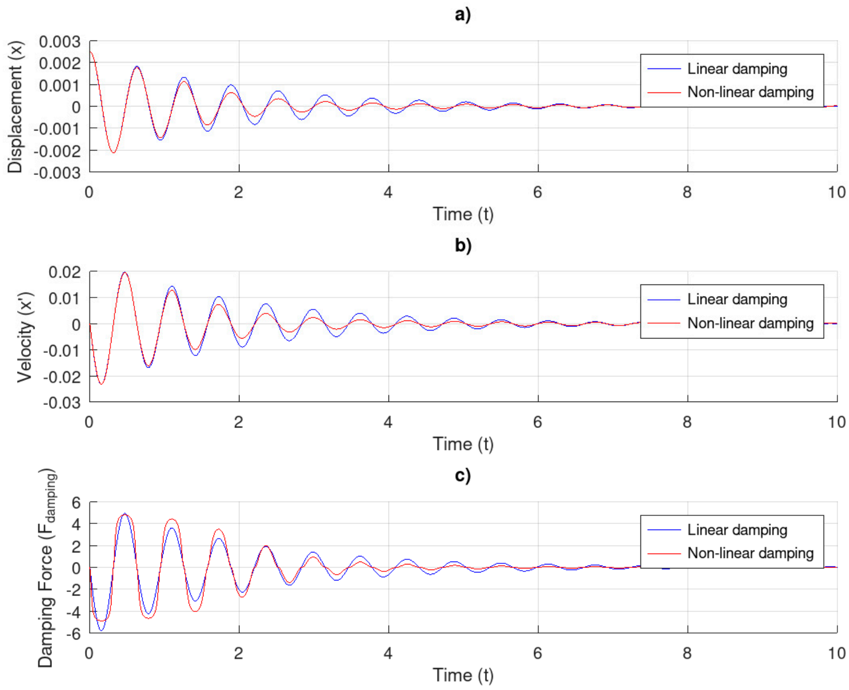

Figure 7. Shock-suppression process: (**a**) displacement of mass m; (**b**) velocity of mass m; (**c**) damping force.

An innovative side of the research is that the nonlinear damping characteristic was achieved with a damper with a simple structure. It is much simpler than the passive dampers presented in [4–6,8], as no parts with complicated geometrical form are necessary. It is simpler than active dampers [9,10,13,15,17] because no sophisticated control system is necessary. But on the other hand, the dynamics of its main component, the slit membrane, have a great influence on damping effectiveness. Therefore, the modelling and simulation of membrane behavior is an important step in developing a slit membrane damper for particular dynamic loads and system parameters.

In further research, this property can be used to control the reaction of the mechanical system to dynamic effects, as shown by the principal diagram in Figure 8.

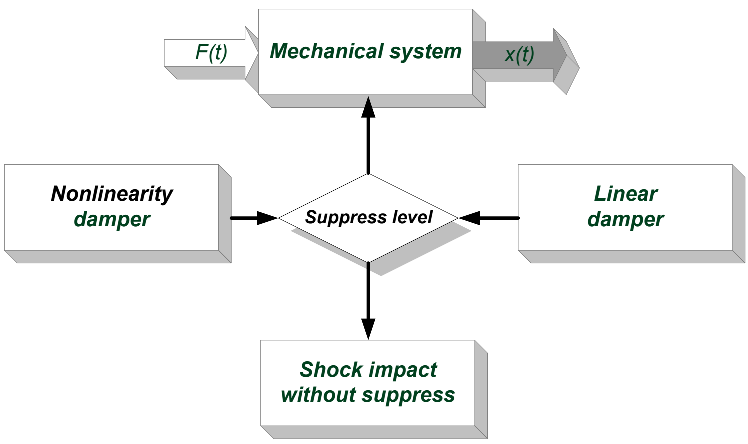

Figure 8. Fragment of suppression-control process in a mechanical system.

In a mechanical system, an input dynamical load can be transmitted directly without suppression (without a damper), using linear suppression (linear damper) and using nonlinear suppression (nonlinearity damper), which is the fastest. Thus, the strength of the reaction will depend on the logical command used in the mechanical system.

6. Conclusions

A nonlinear damper with an original structure is proposed. The key component of the damper is its membrane with slits. The damper has a simple geometrical form compared with most passive or active dampers. The motion of the circular membrane according to the fundamental axisymmetric vibration mode allows a nonlinear damping constant to be achieved. The simulation results show that with the application of this damper, the effectiveness of shock and vibration suppression increases.

Based on the research findings, the following can be concluded:

1. The application of the main component—the flat circular slit membrane—enabled the development of a nonlinear damper with a simple geometrical form.
2. Excited by dynamical load, the slit membrane moves according to the axisymmetric vibration mode, and the deformation-dependent area of the slit openings is obtained.
3. As the velocity determines the shape of the flat circular membrane with slits, the velocity-dependent drag force of the damper is obtained.
4. The nonlinearity of the damping characteristic enables the effective suppression of vibration and shock loads similar to passive and active dampers of sophisticated structures.

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References

1. Balaji, P.S.; SelvaKumar, K.K. Applications of Nonlinearity in Passive Vibration Control: A Review. *J. Vib. Eng. Technol.* **2021**, *9*, 183–213. [CrossRef]
2. Lu, Z.; Wang, Z.; Zhou, Y.; Lu, X. Nonlinear dissipative devices in structural vibration control: A review. *J. Sound Vib.* **2018**, *423*, 18–49. [CrossRef]
3. Pazooki, A.; Goodarzi, A.; Khajepour, A.; Soltani, A.; Porlier, C. A novel approach for the design and analysis of nonlinear dampers for automotive suspensions. *J. Vib. Control* **2018**, *24*, 3132–3147. [CrossRef]
4. Saber, H.; Samani, F.S.; Pellicano, F. A novel nonlinear variable damping device and its application for the systems with uncertain parameters. *Proc. Inst. Mech. Eng. Part K J. Multi-Body Dyn.* **2022**, *236*, 660–671. [CrossRef]
5. Raj, A.H.K.; Padmanabhan, C. A new passive non-linear damper for automobiles. *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.* **2009**, *223*, 1435–1443. [CrossRef]
6. Cossalter, V.; Doria, A.; Pegoraro, R.; Trombetta, L. On the non-linear behaviour of motorcycle shock absorbers. *Proc. Inst. Mech. Eng. Part D J. Automob. Eng.* **2010**, *224*, 15–27. [CrossRef]
7. Dong, B.; Sause, R.; Ricles, J.M. Modeling of nonlinear viscous damper response for analysis and design of earthquake-resistant building structures. *Bull. Earthq. Eng.* **2022**, *20*, 1841–1864. [CrossRef]
8. Xu, W.; Wang, Y.; Guo, H.; Du, D.; Wang, S. Theoretical and experimental investigation on the seismic performance of a novel variable-damping viscous fluid damper *J. Build. Eng.* **2022**, *53*, 104537. [CrossRef]
9. Cheng, M.; Luo, S.; Ding, R.; Xu, B.; Zhang, J. Dynamic impact of hydraulic systems using pressure feedback for active damping, *Appl. Math. Model.* **2021**, *89 Pt 1*, 454–469. [CrossRef]
10. Farahpour, H.; Hejazi, F. Development of adjustable fluid damper device for the bridges subjected to traffic loads. *Structures* **2023**, *47*, 1295–1322. [CrossRef]
11. Eshgarf, H.; Nadooshan, A.A.; Raisi, A. An overview on properties and Applications of magnetorheological fluids: Dampers, batteries, valves and brakes. *J. Energy Storage* **2022**, *50*, 104648. [CrossRef]
12. Vishwakarma, P.N.; Mishra, P.; Sharma, S.K. Characterization of a Magnetorheological fluid damper a review. *Mat. Today Proc.* **2022**, *56 Pt 5*, 2988–2994. [CrossRef]
13. Jiang, R.; Rui, X.; Wei, M.; Yang, F.; Zhu, H.; Gu, L. A phenomenological model of magnetorheological damper considering fluid deficiency. *J. Sound Vib.* **2023**, *562*, 117851. [CrossRef]

14. Kumbhar, S.; Puneet, N.; Kumar, H. Characterization and quarter car analysis with magnetorheological fluid damper using modified algebraic model (mAlg). *Mater. Today Proc.* **2022**, *56 Pt 2*, 749–754. [CrossRef]
15. Devikiran, P.; Shravya, P.; Puneet, N.; Kumar, H. Design, characterization and control of MR damper for two-wheeler applications. *Mater. Today Proc.* **2022**, *62 Pt 4*, 2056–2063. [CrossRef]
16. Emami, M.; Nasab, V.H.; Akar, S.; Batako, A. Experimental investigation into the effect of magnetorheological fluid damper on vibration and chatter in straight turning process. *J. Manuf. Process.* **2023**, *99*, 825–847. [CrossRef]
17. Hu, G.; Ying, S.; Qi, H.; Yu, L.; Li, G. Design, analysis and optimization of a hybrid fluid flow magnetorheological damper based on multiphysics coupling model. *Mech. Syst. Signal Process.* **2023**, *205*, 110877. [CrossRef]
18. Aziz, M.A.; Aminossadati, S.M.; Leonardi, C. Load response of magnetorheological (MR) plastomer dampers under applied magnetic fields. *J. Magn. Magn. Mater.* **2022**, *547*, 168930. [CrossRef]
19. Wei, M.; Lin, K.; Sun, L. Shear thickening fluids and their applications. *Mater. Des.* **2022**, *216*, 110570. [CrossRef]
20. Wei, M.; Lin, K.; Guo, Q.; Sun, H. Characterization and performance analysis of a shear thickening fluid damper. *Meas. Control* **2019**, *52*, 72–80. [CrossRef]
21. Nagy-György, P.; Hős, C. Predicting the damping characteristics of vibration dampers employing generalized shear thickening fluids. *J. Sound Vib.* **2021**, *506*, 116116. [CrossRef]