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Vibration control of fluid-conveying pipes: a state-of-the-art review*

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Abstract Fluid-conveying pipes are widely used to transfer bulk fluids from one point to another in many engineering applications. They are subject to various excitations from the conveying fluids, the supporting structures, and the working environment, and thus are prone to vibrations such as flow-induced vibrations and acoustic-induced vibrations. Vibrations can generate variable dynamic stress and large deformation on fluid-conveying pipes, leading to vibration-induced fatigue and damage on the pipes, or even leading to failure of the entire piping system and catastrophic accidents. Therefore, the vibration control of fluid-conveying pipes is essential to ensure the integrity and safety of pipeline systems, and has attracted considerable attention from both researchers and engineers. The present paper aims to provide an extensive review of the state-of-the-art research on the vibration control of fluid-conveying pipes. The vibration analysis of fluid-conveying pipes is briefly discussed to show some key issues involved in the vibration analysis. Then, the research progress on the vibration control of fluid-conveying pipes is reviewed from four aspects in terms of passive control, active vibration control, semi-active vibration control, and structural optimization design for vibration reduction. Furthermore, the main results of existing research on the vibration control of fluid-conveying pipes are summarized, and future promising research directions are recommended to address the current research gaps. This paper contributes to the understanding of vibration control of fluid-conveying pipes, and will help the research work on the vibration control of fluid-conveying pipes attract more attention.

Key words fluid-conveying pipe, vibration, passive control, nonlinear energy sink (NES), active control, semi-active control

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1 Introduction

Pipes are generally utilized to transmit oil, water, sewage, and natural gas from one place (location) to another. Fluid-conveying pipes play a vital role in the transmission and distribution of various fluids. Different fluid-conveying pipes can be found in various engineering applications, such as water pipes in civil and infrastructure engineering, oil supply pipes in mechanical and aerospace engineering, natural gas pipelines in gas transmission systems, and hydraulic pipes in large engineering systems. Figure 1 shows the schematic of pipes and images of fluid-conveying pipes widely used in engineering[1–5]. They not only play a crucial role in the operation of the systems such as aircraft and nuclear power plants, but also are crucial to any process plants such as oil, gas, and water plants.

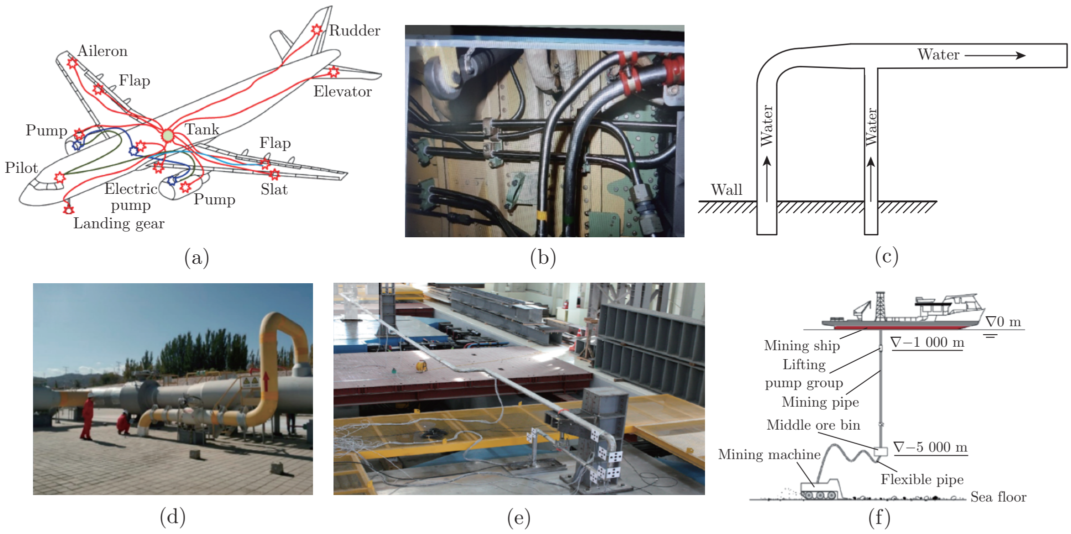


Fig. 1 Schematic of the piping layout and images of piping systems in practical applications: (a) schematic of the layout of aircraft hydraulic pipe system[1], (b) hydraulic pipe image, (c) water supply pipe image[2], (d) natural gas station pipe image[3], (e) piping tested to simulate actual nuclear power plants[4], and (f) schematic of pipes of deep sea mining system[5] (color online)

In transporting fluids, the flow of internal fluids produces a Coriolis force on fluid-conveying pipes[6–7]. The instability of the flow velocity may cause parametric vibrations of pipes[8]. High flow velocities may lead to dynamic instability of fluid-conveying pipes[9]. In addition, the excitations from the external environment may also induce severe vibrations on pipeline systems. The external excitations may act on pipes through the temperature field[10–11], the magnetic field[12], and the external flow field[13–14]. They may also be transmitted to pipes through the supporting structures of pipeline systems[15–16], generating vibrations of the pipes. Vibrations can induce variable dynamic stress and large deformation on the fluid-conveying pipes, leading to vibration-induced fatigue and damage on the pipes, or even leading to failure of the whole piping system and catastrophic accidents as well as significant economic losses. The vibrations of fluid-conveying pipes are a big risk to the integrity and safety of pipeline systems. Take nuclear power plants as an example. A failure in one of the multiple interconnected piping systems can induce severe damage, potentially leading to radiation leakage and diffusion accidents[4]. The damage of pipes caused by seismic waves has always been the focus of the protection of chemical and petrochemical industries and nuclear power plants[4,17]. Therefore, a deep understanding of the vibration mechanism of the fluid-conveying pipes and the effective control of pipe vibrations have always been vital and necessary to ensure reliable operation of pipeline systems in various applications[1,18–20].

1.1 Vibration analysis of fluid-conveying pipes

The understanding of the vibration characteristics of piping systems lays the basis for developing effective vibration control methodologies. Although there is an increasing engineering demand for controlling pipe vibrations, a large amount of existing research on fluid-conveying pipes has been mainly focused on the first stage, i.e., understanding the vibration characteristics of pipes. In comparison, there are relatively few studies focusing on the second stage, i.e., controlling the pipe vibrations. The vibration characteristics of fluid-conveying pipes are extremely complex, and demonstrate rich nonlinear dynamic phenomena. There are many research challenges in the vibration analysis of fluid-conveying pipes. First, fluid-conveying pipes include macroscopic and microscopic pipes in different engineering applications. The pipe shapes, materials, supports, and working conditions are diverse in different pipeline systems. Therefore, it is difficult to develop a unified vibration analysis theory[21–23]. Second, the fluid-structure coupling behaviors between multiphase fluids and pipes[24–25] and between fluids and control valves[26] are often difficult to identify. It is hard to accurately characterize the coupling behavior in the modeling. Third, in studying the dynamics of fluid-conveying pipes, the constraints of the pipes are complicated, and cannot be described accurately to represent the real practical situations, which is still a challenging problem in setting boundary conditions[27–28]. Lastly, the time-varying, aperiodic characteristics of fluid motion, and geometric nonlinear characteristics of piping vibrations need to be considered in the vibration analysis of fluid-conveying pipes[6–7]. However, modeling these factors and studying their effects on the vibration features of fluid-conveying pipes would need more research efforts. In general, the theoretical study on the vibrations of fluid-conveying pipes is gradually moving to deal with the actual situation from the idealized assumption[29–30]. Shao and Ding[31] evaluated the effect of gravity on the nonlinear vibration of a fluid-conveying pipe. The critical impact factors of the gravity of three metal pipes were proposed. Deng et al.[32] studied the effects of the retaining clip stiffness and position on the critical flow velocity, equilibrium configuration, and natural frequencies of pipes. Cao et al.[33] studied the natural characteristics of fluid-conveying pipelines and the effect of the connecting hose. Fan et al.[34] studied the coupling vibration of a hydraulic pipe system consisting of two pipes restrained by clips to one bracket at their middle points.

The vibration analysis of the fluid-conveying pipes has received considerable attention for many years, which mainly aims to understand the vibration characteristics of fluid-conveying pipes under the action of fluid and external excitations. In the context of mechanical vibrations of a fluid-conveying pipe, the pipe is a continuum vibration system with infinite degrees-of-freedom. The research focus has been on dynamics modeling, analytical methods for linear and nonlinear dynamic equations, vibration characteristics, static and dynamic instability, and nonlinear dynamic behavior after instability. Take Ref. [27] as an example to illustrate the vibration analysis of fluid-conveying pipes. Figure 2 shows the mechanical model of a fluid-conveying pipe with both ends constrained by retaining clips. The dynamic model describing the vibration of the pipe is a partial differential equation by considering the pipe as a continuum system. According to the generalized Hamilton principle and the Euler-Bernoulli beam theory, the nonlinear dynamic model and boundary conditions could be derived for the transverse

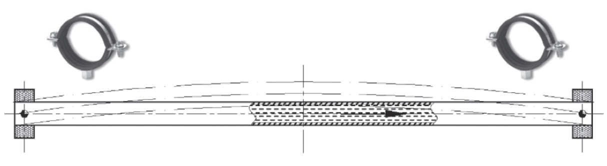


Fig. 2 Model of a fluid-conveying pipe restrained by clips[27]

bending vibration of the fluid-conveying pipe shown in Fig. 2[27], and can be expressed as

$$\begin{aligned}
&\underbrace{(\rho_{\rm p}A_{\rm p}+\rho_{\rm f}A_{\rm f})U_{,TT}}_{\text{Inertia force term}}\underbrace{+2\rho_{\rm f}A_{\rm f}VU_{,XT}}_{\text{Coriolis force term}}\underbrace{+\rho_{\rm f}A_{\rm f}V^2U_{,XX}}_{\text{Centrifugal force term}}\underbrace{-P_0U_{,XX}}_{\text{Axial pressure term}}\underbrace{-pA_{\rm f}U_{,XX}}_{\text{Fluid pressure term}}\\
&\underbrace{+EIU_{,XXXX}}_{\text{Restoring force term}}\underbrace{+cU_{,T}}_{\text{External damping term}}\underbrace{+\alpha IU_{,XXXXT}}_{\text{Material damping term}}\\
=\;&\underbrace{F(X,T)}_{\text{External force item}}+\underbrace{\frac{EA_{\rm p}}{2L}U_{,XX}\int_0^L U_{,X}^2{\rm d}X+\frac{\alpha A_{\rm p}}{L}U_{,XX}\int_0^L U_{,X}U_{,XT}{\rm d}X}_{\text{Geometrically nonlinear term of cube}},
\end{aligned}\tag{1}$$

$$\begin{cases}
\underbrace{EIU_{,XX}(0,T)-M_{\rm B}(0,T)=0}_{\text{Boundary conditions of bending moment}}, & \underbrace{EIU_{,XXX}(0,T)+F_{\rm B}(0,T)=0}_{\text{Boundary conditions of shear force}},\\
\underbrace{EIU_{,XX}(L,T)+M_{\rm B}(L,T)=0}_{\text{Boundary conditions of bending moment}}, & \underbrace{EIU_{,XXX}(L,T)-F_{\rm B}(L,T)=0}_{\text{Boundary conditions of shear force}},
\end{cases}\tag{2}$$

where L denotes the distance between the two clips. $\rho_{\rm p}$ is the density of the pipe. $A_{\rm p}$ represents the cross-sectional area of the pipe. $U(X,T)$ represents the transverse vibration displacement of the pipe. The subscript comma in front of X or T represents partial differentiation with respect to X or T. E and I, respectively, are the modulus of elasticity and the moment of inertia of the pipe. $\rho_{\rm f}$ represents the density of the fluid. $A_{\rm f}$ denotes the cross-sectional area of the fluid. V represents the velocity of the fluid inside the pipe. P_0 stands for the axial force applied to the pipe. α represents the material damping of the pipe. c represents the external damping. $F_{\rm B}$ and $M_{\rm B}$, respectively, are the reaction force and moment of the clips. Here, only the geometric cubic nonlinearity is retained.

In the literature, the boundary conditions of pipes are mainly based on the ideal boundary conditions of simple support, roller support, pinned support, and fixed support. The boundary condition (2) can be degenerated to the above-mentioned ideal boundary conditions by taking different values of the boundary force and moment. For the pipes with elastically constrained boundaries, the relationship between the boundary support force and moment and the displacement and rotation angle is usually assumed to be linear. The boundary support reactions of the pipe shown in Fig. 2 are set to be nonlinear[27]. Moreover, Dou et al.[28] proved that when calculating the natural frequencies of the pipe, ignoring the clip width or modeling the retaining clip as a boundary would cause a noticeable error.

Once a dynamic model is established, the dynamic analysis will be carried out to study the influence of boundary conditions on the resonance frequency, modal shapes of the pipe, and the vibration response, decoupling of the modal responses, the influence of system parameters on the vibration response, and complex nonlinear phenomena. For example, the vibration characteristics of fluid-conveying pipes were investigated by considering different boundary supporting conditions[27], different pipe materials[35], different pipe shapes[36], and different working environments[37]. Research progresses have also been made in dynamic analysis methods, including the development of numerical methods[38–39] and approximate analytical methods[40–41], and in exploring dynamic behaviors, including the dynamic responses under different excitation conditions[42–43] and complex nonlinear vibrations under different parameter conditions[44]. In the literature, attention has been mainly paid to the understanding of the vibration characteristics of fluid-conveying pipes, resulting in a relatively slow progress in the vibration control of fluid-conveying pipes, though there is an increasing demand on the vibration control from engineering application perspectives.

1.2 Existing review papers on vibration and control of fluid-conveying pipes

Over the last two decades, a number of literature reviews have been conducted on different aspects of fluid-conveying pipes. For example, Zhang et al.[45] reviewed the research on the dynamic models of fluid-conveying pipes in 2000. Ren and Jiang[46] outlined the dynamic modeling and analysis methods, test methods, vibration control, and dynamic strength design of pipes in 2003. Xu and Yang[47] considered the nonlinear vibration, stability, bifurcation and chaos, and vibration control of fluid-conveying pipes in 2004. Païdoussis[24] summarized three of the many unsolved or partially solved problems in fluid-structure interactions in 2005. Ibrahim presented an overview of the modeling, dynamic analysis, and stability of pipe mechanics for transporting fluids[7] and an overview of fluid elastic instability and related applications in single-phase and two-phase flows[6] in 2010 and 2011, respectively. Hong and Shah[18] focused on the dynamics and vibration control techniques of marine riser systems in 2018. In 2020, Tan et al.[20] discussed the seismic damage and passive seismic protection of building piping systems. Quan et al.[48] reviewed the research progress on fluid-structure interaction models and the passive, active, and semi-active vibration control of hydraulic pipes in 2015. Gao et al.[1] considered the vibration analysis and vibration control technology of aircraft hydraulic piping systems in 2021. Focusing on an experimental platform for a vertical cantilever pipe transporting fluids, Païdoussis et al.[13] reviewed the problem of pipes loss of stability due to flutter or static divergence under a sufficiently high flow velocity in 2021. Later, Païdoussis[49] reviewed the dynamics of cylindrical structures immersed in axially flowing fluids in 2021. The above-mentioned review papers have provided useful information on the vibration and control of some specific pipes. To the authors' best knowledge, there have been no thorough reviews on the vibration control of fluid-conveying pipes, though the research has been of growing interest over the last two decades.

1.3 Organization of this review paper

There are a large number of pipes widely found in engineering applications. These pipes are of different shapes, subject to multiple excitations from their operation environments, and are under complex support constraints, as well as transport multiphase fluids. These factors bring more challenges in the vibration control of fluid-conveying pipes. Research on the vibration control of fluid-conveying pipes is far less than that on vibration characteristics, though the vibration control is the ultimate goal of vibration analysis. Over the last 20 years, focusing on the design, optimization, and testing, the vibration control of fluid-conveying pipes has been investigated from different aspects. On the basis of the above review papers and the research work published in the past 20 years, this paper will conduct a comprehensive literature review on the vibration control of fluid-conveying pipes.

By distinguishing between energy input and non-energy input, the vibration control of fluid-conveying pipes can be categorized into three groups, i.e., active control, passive control, and semi-active control. In the family of passive control, it can be further subdivided into the use of vibration absorbers, vibration dampers, and vibration isolators for achieving the vibration control. In addition, based on theoretical analysis and experimental investigation, the control of pipe vibrations can also be achieved by optimizing the configuration, support conditions (or boundary constraint conditions), and flow velocity of fluid-conveying pipes. The remainder of this review paper is organized as follows. Section 2 gives an overview of research work related to the passive control. Section 3 presents the active control of vibrations of fluid-conveying pipes. Section 4 reviews the semi-active vibration control of pipes. Section 5 focuses on configuration optimization for pipes. In order to clearly show the research progress in the vibration control of fluid-conveying pipes, Section 6 summarizes the existing literature, and presents some concluding remarks on future promising topics.

2 Passive control

The fluid-conveying pipes in engineering applications have diverse material characteristics and different lengths and shapes, which lead to the natural resonance frequencies of the pipes falling in a wide range from a low frequency to a high frequency. Moreover, the installation space of fluid-conveying pipes in many engineering applications is generally small and compact. In addition, the hydraulic pipes that are widely used in large mechanical systems are characterized as being slender, and the internal fluid is subject to a high pressure and a high flow velocity.

The passive vibration control does not need to provide external energy, and has the advantages of simple structure, high reliability, and low cost. Common passive control methods mainly include adding a dynamic vibration absorber (DVA), increasing the system damping, and introducing a vibration isolation element. Generally speaking, adding a DVA can transfer the vibration energy of the fluid-conveying pipe to the absorber, increasing the system damping can dissipate the vibration energy of the pipe, and introducing a vibration isolation element can prevent or reduce the vibration transmission to the pipe, all of which can achieve the purpose of passively controlling the vibration of fluid-conveying pipes.

2.1 Vibration control using absorbers

Fluid-conveying pipes have both inertial and elastic elements that constitute a vibration system. Vibration absorbers are one of the widely used approaches for the vibration control due to their simple structure, high efficiency under specific conditions, and low cost[50].

2.1.1 Use of linear absorbers

Theoretically, the DVA is an additional vibration system attached to the main structure, which can generate a reaction force on the primary structure and thus reduce the vibration intensity of the main structure. Passive vibration absorption has been a popular vibration reduction approach for fluid-conveying pipes in engineering. Kunieda et al.[51] introduced the DVA, and confirmed its applicability to the pipe system through vibration tests by using a three-dimensional (3D) pipe model in 1987. Mani and Senthilkumar[52] developed an adaptive passive DVA with the help of a shape memory alloy spring, and used it to reduce the vibration amplitude of a fluid-conveying cantilever pipe in 2015. In 2017, Wang et al.[53] utilized the vortex DVA to control the vibration of the fluid-conveying pipe in the seawater environment. Kwag et al.[4] designed and tested a DVA configuration for nuclear pipes in 2021. The mass-spring-damper based vibration absorber was installed on the pipe through U-bolts. Vibration test results and numerical simulations confirmed that the application of DVAs was helpful to improve the seismic performance of nuclear pipes.

The linear DVA has a simple structure, but only works in a single resonance frequency. To overcome this problem, Yu et al.[54] theoretically designed a fluid-conveying pipe with the concept of phononic crystals by periodically attaching local resonance harmonic oscillators to the pipeline in 2008. Through the undamped linear oscillator, the vibration models of the resonant oscillator were established, and the low frequency bandgap was obtained. Nateghi et al.[55] used polymethyl methacrylate panels to make local resonance structures for vibration reduction and control of sound radiation in aluminum pipes in 2019. Experimental studies and numerical calculations showed that a distinct stop-band region was created by adding several rows of resonance structures. Wu et al.[56] used multiple DVAs to periodically connect to a pipe, as shown in Fig. 3, to form a system similar to locally resonant phononic crystals in 2021. The analytical and experimental results showed that the bending vibration of the pipe could be controlled more effectively by widening the attenuation bandwidth, when the DVAs were arranged periodically. In 2021, El-Borgi et al.[57] proposed a model of the resonator with a cantilever beam having a concentrated mass at the tip, and controlled some bending mode resonances of the pipe by adjusting the resonator, as shown in Fig. 4.

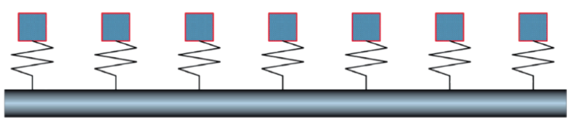


Fig. 3 Schematic of a fluid-conveying pipe with locally resonant oscillators[56] (color online)

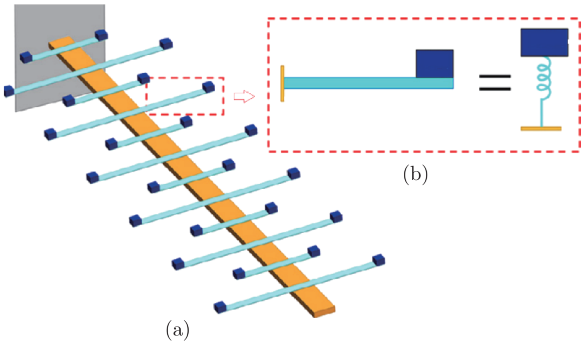


Fig. 4 (a) Schematic of a cantilever beam with double cantilever beam resonators and (b) simplification of a cantilever beam resonator into spring-mass oscillator[57] (color online)

Song et al.[58] designed an impact-type shock absorber and applied it to the vibration control of M-shaped pipes in 2016. The impact-type shock absorber was realized by L-shaped beams and mass blocks, as shown in Fig. 5. Jiang et al.[59] applied the shock absorber to the vibration control of an underwater pipe, and showed that the impact-type shock absorber could effectively control the free vibration and forced vibration of the pipe in 2017. Mlynarczyk et al.[60] designed a vibration damping device by combining a tuned mass damper and a shaped nozzle for pressure pulsation attenuation, as shown in Fig. 6, to control the vibration of the refrigerating compressor manifolds. The nozzle was mounted inside the pipe in the chosen position by using springs with tuned stiffness. The nozzle mass and spring stiffness created the tuned mass damper. It was shown that the use of a nozzle design with a tuned mass damper function could control the vibration of the refrigerating compressor manifolds more effectively than a fixed nozzle.

The above studies have shown that for a defined resonant frequency of a given fluid-conveying pipe, whether it is a periodic oscillator based on the local resonance theory[54,56], or a single

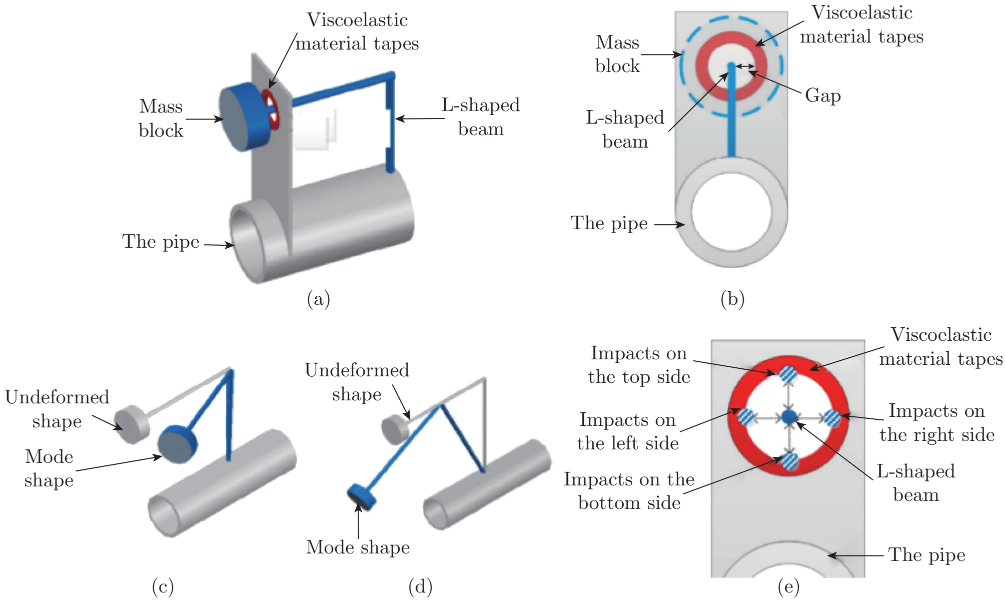
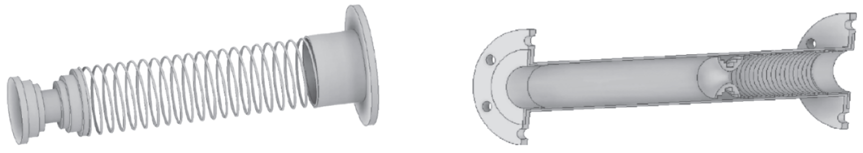


Fig. 5 The pounding tuned mass damper designed for an M-shaped pipe structure: (a) 3D diagram, (b) cross section diagram, (c) mode shape of tuned mass damper in the horizontal direction, (d) mode shape of tuned mass damper in the vertical direction, and (e) two-dimensional pounding[58] (color online)



(a) 3D diagram (b) Profile diagram

Fig. 6 The hybrid nozzle-tuned mass damper[60]

DVA[58–59], a good vibration suppression could be achieved. It is widely known that there is more than one resonance frequency in fluid-conveying pipes. The above-mentioned linear vibration absorber cannot perform adaptive vibration control on different resonance frequencies of the fluid-conveying pipes.

2.1.2 Use of nonlinear energy sinks (NESs)

The introduced linear DVA into the primary structure changes the vibration characteristics of the primary structure by adding a new resonance frequency. The elasticity of the structure includes a linear part and a nonlinear part. The linear elastic part of the DVA changes the natural vibration frequency and the mode of the primary structure. Usually, when a DVA is used to attenuate resonance, the restoring force provided by the linear elastic part is much larger than that of the nonlinear part. If the restoring force provided by the linear part is insignificant relative to the nonlinear part, then the nonlinear part plays a major role in suppressing the vibration[61]. In this situation, the DVA has little influence on the vibration characteristics of the primary structure, and exhibits an excellent vibration absorption performance[62]. Maciel et al.[63] composed a rotative nonlinear vibration absorber by a mass connected to the extremity of a rigid bar linked to the pipe by means of a hinge and a rotational damper. The research showed that the use of the rotative nonlinear vibration absorber is a more reliable solution when compared with linear passive suppressors. Furthermore, if a nonlinear stiffness with zero linear stiffness is designed to connect the main structure and the additional mass of the vibration absorber, a nonlinear dynamic vibration absorber with zero linear stiffness will be formed, which has been referred to as an NES in the literature. Since no new linear stiffness is introduced to the system, the NES can overcome the disadvantage of linear DVAs that increases the number of resonance frequencies[64]. Moreover, since the linear part of the connection stiffness between the absorber and the primary structure is zero, only the nonlinear part is included. Therefore, the vibration energy of the primary structure can be transferred to the absorber in a targeted manner under certain conditions. Thus, the vibration of the primary structure can be effectively suppressed. Moreover, the NES has adaptive characteristics to the resonance frequency of the primary structure, and can control multiple resonance peaks[65–66]. Therefore, the adaptive characteristics of the NES are appropriate for the fluid-conveying pipes whose resonance frequencies may change during the working process.

Yang et al.[67] proposed the use of an NES to attenuate the vibration response of simply-supported fluid-conveying pipes under impact in 2014. The nonlinear absorber was placed between two supports of the pipe, as shown in Fig. 7. The dynamic models of the fluid-conveying pipe and nonlinear absorber are given in Eqs. (3) and (4), respectively. The connection between the nonlinear absorber and the controlled pipe does not contain a linear stiffness component. It was shown that the introduction of NES does not change the original resonance frequency of the pipe, and the coupling system between the pipe and the NES does not add a new resonance frequency. In addition, the NES could efficiently absorb the vibration energy of the pipe.

$$\begin{aligned}&(\rho_{\mathrm{p}}A_{\mathrm{p}}+\rho_{\mathrm{f}}A_{\mathrm{f}})U_{,TT}+2\rho_{\mathrm{f}}A_{\mathrm{f}}VU_{,XT}+\rho_{\mathrm{f}}A_{\mathrm{f}}V^2U_{,XX}+EIU_{,XXXX}+\alpha IU_{,XXXXT}\\&+\underbrace{\Big(K(U(D,T)-Y(T))^3+C\Big(\frac{\partial U(D,T)}{\partial T}-\frac{\partial Y(T)}{\partial T}\Big)\Big)\delta(X-D)}_{\text{Coupling force}}=0,\end{aligned}\tag{3}$$

$$\underbrace{m_{\mathrm{NES}}\frac{\partial^2 Y(T)}{\partial T^2}}_{\text{Inertia force of the NES}} \quad \underbrace{+K(U(D,T)-Y(T))^3}_{\text{Nonlinear restoring force of the NES}} \quad \underbrace{+C\Big(\frac{\partial U(D,T)}{\partial T}-\frac{\partial Y(T)}{\partial T}\Big)}_{\text{Damping force of the NES}} = 0, \tag{4}$$

where K is the cubic nonlinear stiffness, m_{NES} is the mass of the NES, C is the damping coefficient of the NES, D indicates the position of the NES, and $Y(T)$ is the displacement of the NES.

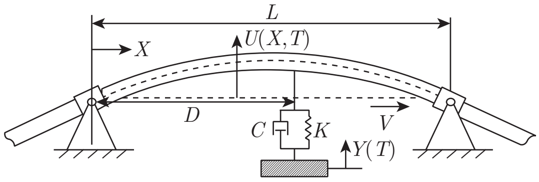


Fig. 7 A simply-supported pipe with an NES[67]

Mamaghani et al.[68] showed that under external periodic excitation, the midpoint of the pipe was the best position on the connection of the NES and the fluid-conveying pipe fixed at both ends for controlling the resonance of the first-order mode in 2016. Zhao et al.[69] optimized the parameters of an NES in the vibration reduction of fluid-conveying pipes in 2018. In 2019, Zhou et al.[70] introduced an NES to the vibration control of a cantilever pipe, and found that the NES could increase the critical flow velocity of the cantilever pipe, thereby improving the stability of the pipe. Liu et al.[71] introduced an inerter to enhance the vibration control of a fluid-conveying cantilever pipe in 2020. Yang et al.[72] utilized an NES with negative linear stiffness and cubic nonlinear stiffness for the vibration control of pipes, and showed that the maximum 98.19% suppression efficiency of pipe vibration could be achieved in 2019. Khazaee et al.[73] compared the vibration control of fluid-conveying pipes by two NESs, grounded and ungrounded, in 2020. They found that the stability of the pipes and the vibration control efficiency of the NES were reduced by increasing the fluid velocity inside the pipes. Furthermore, the effects of different configurations of multiple NESs on the vibration control of fluid-conveying pipes were compared[74]. Duan et al.[75–77] studied the stability of the NES-pipe conveying fluid, the inerter-enhanced NES-pipe, and the multiple parallel NES-pipe, respectively. Yang et al.[78] investigated the vibration suppression effect of an NES on the wind-vortex-induced vibration of deep-water jacket pipes in 2022. Mirhashemi et al.[79] investigated the dynamics of a harmonically excited nonlinear fluid-conveying pipe attached to a local NES in 2023. Philip et al.[80] employed the NES with nonlinear damping to control the vibrations in fluid-transmitting pipes. Tang et al.[81] integrated a piezoelectric energy harvester and the NES to investigate the vibration attenuation and self-powered sensing of 3D functionally graded fluid-conveying pipes under the end supports of complex constraints.

In the above-NES related literature, the NES was placed between two boundaries of fluid-conveying pipes, and was used for the control of the first-order modal resonance of the pipes. Theoretically, a fluid-conveying pipe has an infinite number of resonance frequencies, and the lower modal resonances could affect the fatigue life of the pipe[82].

Due to different lengths and flexibilities of fluid-conveying pipes, it is very difficult to configure a DVA for each pipe to effectively control the vibrations. This brings challenges in the design and application of DVAs for the vibration control of fluid-conveying pipes. Mao et al.[83] designed a novel torsional NES which was placed at the boundary of the pipe to control the flexural vibration of the pipe, as shown in Fig. 8. By using the torsional stiffness of the pipe boundary, the bending vibration energy was transferred to the vibration absorber at the boundary in a targeted manner. According to the generalized Hamilton principle and the Euler-Bernoulli linear beam theory, the dynamic model of the pipe was established, as shown in

Eqs. (5)–(7). The Taylor expansion of the connection expression between the vibration absorber and the pipe was conducted, and the nonlinear terms above the third order were discarded.

$$(\rho_{\mathrm{p}}A_{\mathrm{p}}+\rho_{\mathrm{f}}A_{\mathrm{f}})U_{,TT}+2\rho_{\mathrm{f}}A_{\mathrm{f}}VU_{,XT}+(\rho_{\mathrm{f}}A_{\mathrm{f}}V^2-P_0)U_{,XX}+EIU_{,XXXX}+\alpha IU_{,XXXXT}=0, \tag{5}$$

$$\begin{cases}\underbrace{EI_{\mathrm{p}}U_{,XX}(0,T)+\frac{k_1R^2r^2}{2(R-r)^2}(\theta_{\mathrm{L}}-U_{,X}(0,T))^3+\frac{3c_1R^2r^2}{2(R-r)^2}(\theta_{\mathrm{L}}-U_{,X}(0,T))^2(\theta_{\mathrm{L},t}-U_{,XT}(0,T))=0,}_{\text{Boundary conditions of bending moment}}\\ \underbrace{EI_{\mathrm{p}}U_{,XX}(L,T)-\frac{k_1R^2r^2}{2(R-r)^2}(\theta_{\mathrm{R}}-U_{,X}(L,T))^3-\frac{3c_1R^2r^2}{2(R-r)^2}(\theta_{\mathrm{R}}-U_{,X}(L,T))^2(\theta_{\mathrm{R},t}-U_{,XT}(L,T))=0,}_{\text{Boundary conditions of bending moment}}\\ \underbrace{U(0,T)=B\sin(\Omega t),\quad U(L,T)=0,}_{\text{Boundary conditions of displacement}}\end{cases} \tag{6}$$

$$\begin{cases}\underbrace{J\eta^2\theta_{\mathrm{L},tt}}_{\text{Inertial force term of the rotator}}\underbrace{+\frac{k_1R^2r^2}{2(R-r)^2}(\theta_{\mathrm{L}}-U_{,X}(0,T))^3}_{\text{Cubic nonlinear restoring force term of the rotator}}\\ \underbrace{+\frac{3c_1R^2r^2}{2(R-r)^2}(\theta_{\mathrm{L}}-U_{,X}(0,T))^2(\theta_{\mathrm{L},t}-U_{,XT}(0,T))=0,}_{\text{Nonlinear damping force term of the rotator}}\\ J\eta^2\theta_{\mathrm{R},tt}+\frac{k_1R^2r^2}{2(R-r)^2}(\theta_{\mathrm{R}}-U_{,X}(L,T))^3+\frac{3c_1R^2r^2}{2(R-r)^2}(\theta_{\mathrm{R}}-U_{,X}(L,T))^2(\theta_{\mathrm{R},t}-U_{,XT}(L,T))=0,\end{cases} \tag{7}$$

where θ denotes the rotational angle of the driven ring, and J is the rotational inertia of the rotator, which is expressed by m_0R^2. k_1 is the stiffness of the spring, and s_0 represents the initial length of the spring. At the left end, a basement excitation, $B\sin(\Omega T)$, provides a harmonic disturbance to the pipe. The length of the drive rocker is r. The radius of the rotator is denoted as R, and its mass is m_0. c_1 stands for the damping coefficient. The subscripts L and R of θ indicate the torsional angle of the absorber on the left and right sides of the pipe, respectively. The boundary conditions of the pipe and the governing equations of the torsional vibration absorber have no linear stiffness connection between the pipe and the absorber. It is shown that the torsional NES at the pipe boundary has three advantages. First, it has the characteristics of the NES, and does not change the resonance frequency of the pipe. Second, the torsional NES can control the first two resonances of the pipe, as shown in Fig. 9. Therefore, the resonance of the pipe can be automatically captured, indicating adaptive characteristics. Third, effective control of pipe vibration can be achieved with a very light additional mass.

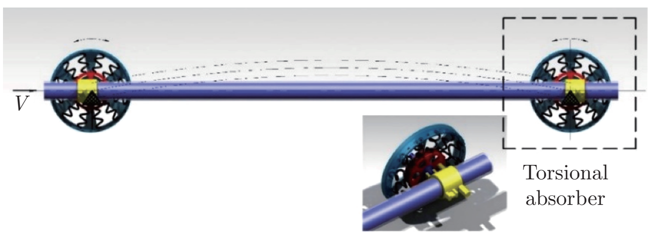


Fig. 8 The fluid-conveying pipes with torsional absorbers[83] (color online)

The above studies demonstrated that the NESs offer a promising opportunity for adaptively controlling multiple resonance in fluid-conveying pipes in a wide frequency range[68,72,83]. It is expected that controlling the complex dynamic responses of fluid-conveying pipes through NESs and implementing applied research into practice will still be active research topics.

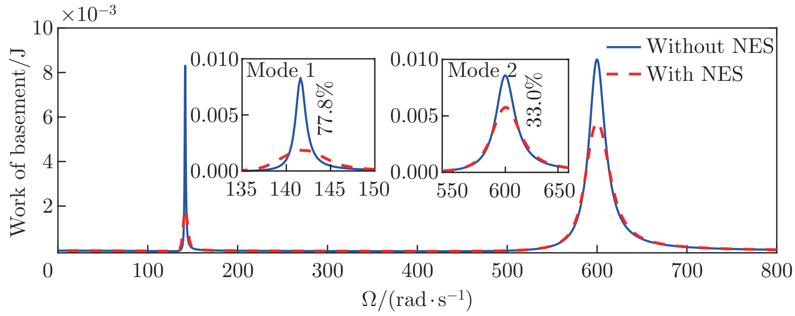


Fig. 9 Vibration response at one-fourth of the pipe with the fluid velocity 20 m/s[83] (color online)

2.2 Use of damper

Increasing the damping of the vibration system to enhance the vibration energy dissipation of the pipe system has been commonly used in engineering, and it is also an effective method for the vibration control of fluid-conveying pipes. In the 1980s, increasing the damping ratio to enhance the stability of nuclear power plant pipes was reported in several conference papers. In 1987, Kunieda et al.[51] introduced viscoelastic dampers and elastic-plastic dampers to suppress the vibration of piping systems. In 1990, Chiba and Kobayashi[84] developed visco-elastic dampers and elasto-plastic dampers for high damping supports for pipes. In 2005, Parulekar et al.[85] introduced elastoplastic dampers as supports for the seismic resistance of cantilever pipes and 3D piping systems. In 2006, Bakre et al.[86] introduced an X-plate damper. In 2007, Bakre et al.[87] installed a friction damping device between the fluid-conveying pipe and its support. In 2016, Chang et al.[88] introduced the Stockbridge damper, and fixed it in a definite position of the pipe to study its seismic performance. In 2017, Wang et al.[53] generated the damping force by the relative movement of a non-magnetic conductive metal through a magnetic field. In 2019, Cho et al.[89] introduced steel coil dampers into the earthquake resistance of pipes. Through finite element simulation and experimental studies, the vibrations of the controlled piping system were compared with the corresponding uncontrolled piping system vibrations to show that these dampers were very effective in reducing the seismic response of the piping system. In 2022, Hong et al.[90] developed a magnetorheological damping pipe clamp to suppress the low-frequency flow-induced vibrations of fluid-conveying pipelines.

In 2016, Kumar et al.[17] compared the control performance of several passive control devices, i.e., an X-shaped plate damper, a viscous damper, a viscoelastic damper, a single DVA, and multiple DVAs, on the seismic response of a piping system. They found that under certain optimal parameters such as the stiffness and damping, passive control devices without external energy input were very helpful for seismic response control of pipeline systems. In 2016, Bi and Hao[91] laid a viscoelastic material on the outer surface of the pipe, and numerically studied the vibration control of the pipe on the ground induced by earthquakes. A double-sided tape was used as the viscoelastic material. The outside was covered with a constraining layer. The fluid-conveying pipe was not completely covered by the viscoelastic material and the constraining layer, and a certain gap was introduced between the covering layers, as shown in Fig. 10. The results showed that even adding a thin layer of viscoelastic material could significantly increase the damping of the pipe.

Sabahi et al.[92] presented the vibration analysis of functionally graded porous fluid-conveying micro-pipes embedded in a viscous damping medium, and showed that decreasing the power-law exponent and the porosity volume fraction could decrease the nonlinear natural frequency of the vibration of fluid-conveying micro-pipes. Li et al.[93] studied the fluid-conveying pipes composed of graphene-reinforced composite layers and a viscoelastic interlayer.

Yano et al.[94] and Ishikawa et al.[95] attached silicone as a damping material to the pipe, as shown in Fig. 11. A modeling method for building a damping material with additional mass and damping was proposed, and viscoelastic materials were regarded as Voigt materials. It was found that the vibration of the pipe not only was controlled by the damping effect of viscoelastic materials, but also exhibited similar dynamic behavior to that controlled by a DVA. In 2023, Ma et al.[96] studied the nonlinear vibration characteristic of a straight pipe with a partially attached viscoelastic damping patch.

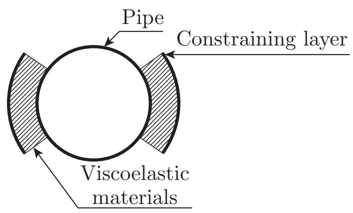


Fig. 10 Silicone material on pipe[91]

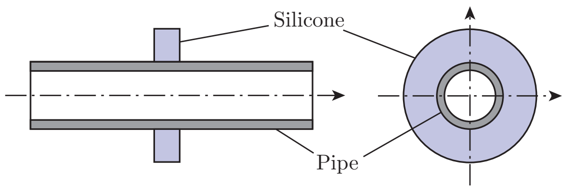


Fig. 11 Silicone material on pipe[94–95] (color online)

The flow of fluid in the pipes makes fluid-conveying pipes have some unique features that are different from static pipes. For example, the Coriolis force generated when the fluid is transported at a high speed in a pipe may cause the pipe to lose its stability[97–98]. However, the Coriolis force caused by the medium flow velocity may dampen the vibration of a cantilever pipe, but not for simply-supported or fixed fluid-conveying pipes at both ends[98]. Guo et al.[99] introduced a dry friction restraint to improve the stability of fluid-conveying cantilever pipes. The results demonstrated that the vibration of the pipe can be suppressed to some extent by setting reasonable dry friction parameters. Ambe et al.[100] proposed a stabilized controller for a jet-actuated two-dimensional cantilever pipe with a nozzle unit at the tip by using the damping effect of the internal flowing fluid, and verified the controller with a robot. A simple controller that can constantly decrease the energy function was proposed by utilizing the damping effect of the flowing fluid. They found that the vibrations were damped by the fluid effect, and the stability was improved by using the proposed controller. Wang et al.[101] presented a research on the multi-cavity particle damping technique for controlling the vibration of pipes.

In summary, for pipes, especially those made of metal and alloy materials, the damping in the material is relatively weak. Externally introduced damping, either various damping introduced at supports[17,51,84,86–87,95,102], or dampers suspended between the supports of pipes[17,89,101], or a damping coating attached to pipes[91,94–95], could significantly reduce the vibration response of the fluid-conveying pipes under seismic excitation and high-frequency excitation. However, for the resonance response in the low-frequency range of the pipe, how to design a damping device to control the vibration is a research topic that needs attention.

2.3 Use of isolators

Vibration isolation is realized by adding elastic and viscous vibration isolators between the excitation source and the controlled structure to reduce the transmission of vibration and achieve vibration control. Research on vibration isolation mainly focuses on the primary structure of single degree-of-freedom, and on the isolation of single resonance. Since the continuum has infinite resonance frequencies, and the vibration of the continuum is transmitted through more than one point, the isolation of multimodal vibration of the continuum is more complicated. There is little relevant research in this area. However, in order to protect high-risk engineering pipes such as chemical industry and nuclear power plant pipes[4] and natural gas pipes[3], the isolation of pipes has always been a subject of concern in engineering applications.

2.3.1 Isolation of transverse vibration of elastic beams

Taking the elastic beam without internal fluid as an object, Ding et al.[103] established a nonlinear elastic transverse vibration model supported by linear springs at both ends, as shown in Fig. 12, where the symbols $\omega_{\rm B}$ and F_0 are the excitation frequency and the amplitude, respectively. $K_{\rm L}$ is the linear support stiffness of the left boundary. $K_{\rm R}$ is the linear support stiffness of the right boundary. Based on the steady-state vibration response of the beam under harmonic excitation, two transmissivities for the transverse bending vibration of the beam to the ground were calculated[103], i.e., the resultant force transmissibility of the two supports acting on the ground (see Eq. (8)) and the pressure transmissibility (see Eq. (9)). Figure 13 compares the two transmissivities, where the symbols ω_1 and ω_2 are the first and second natural frequencies of the pipe, respectively. It was found that the two transmissivities are significantly different. In addition, the effect of support stiffness on the vibration transmissibility of the first two modal resonance of the elastic pipe was also studied, as shown in Fig. 14. The results showed that the effect of the support stiffness on the transmissibility of the first two modal resonance has an opposite trend, indicating that the isolation of continuum vibration is quite different from the isolation of single-degree-of-freedom structural vibration.

$$\eta_{\rm f} = \max\left(\underbrace{(K_{\rm L}U(0,T)+K_{\rm R}U(L,T))}_{\text{Resultant pressure on the boundaries}} \Big/ \underbrace{(F_0L)}_{\text{Static pressure on the boundaries}}\right), \tag{8}$$

$$\eta_{\rm p}(t) = \frac{1}{F_0L}\begin{cases} \max(K_{\rm L}U(0,T)+K_{\rm R}U(L,T)) & \text{if} \quad U(0,T)<0, \quad U(L,T)<0, \\ \max(K_{\rm L}U(0,T)) & \text{if} \quad U(L,T)>0, \\ \max(K_{\rm R}U(L,T)) & \text{if} \quad U(0,T)>0. \end{cases} \tag{9}$$

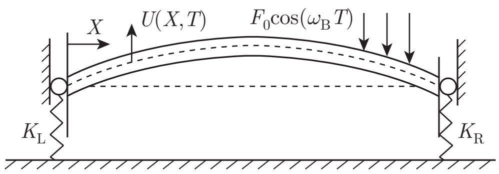


Fig. 12 An elastic beam with vertical spring support[103]

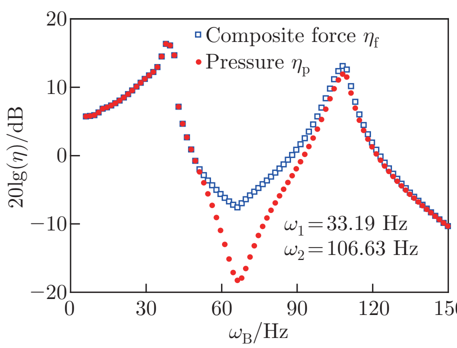


Fig. 13 Comparisons between the pressure and the composite force transmissibility[103] (color online)

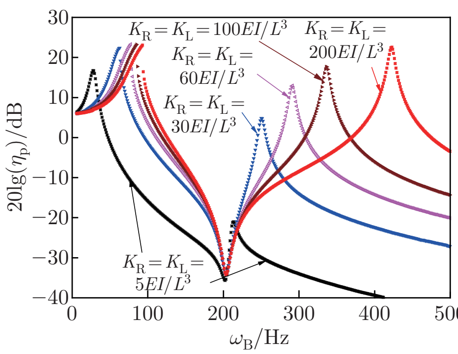


Fig. 14 Effects of the stiffness of the support springs on the vibration transmissibility[103] (color online)

The vibration modes of the elastic continuum would be difficult to obtain if the effects of nonlinear stiffness on the boundary supports were taken into consideration. This poses a challenge to perform perturbation analysis. Mao et al.[104] incorporated boundary conditions into the governing equations, and carried out the perturbation analysis based on the modified mode and the differential quadrature element method.

With the linear vibration isolation theory of single-degree-of-freedom systems, it has been found that an increase in the isolation stiffness deteriorates the vibration isolation of low-frequency excitation. However, a decrease in the isolation stiffness weakens the stability of the systems. In addition, an increase in the linear isolation damping deteriorates the isolation of the high-frequency excitation, while a decrease in the isolation damping increases the amplitude of the system resonance. Ding et al.[105] utilized a quasi-zero stiffness isolator to isolate the transverse vibration of an elastic beam. In addition, the force transmissibility through two boundaries separately in steady-state vibration was defined. In the steady-state response phase, the force transmissibility of the transverse vibration at the two ends of the pre-pressured beam can be defined as Eq. (10), where the linear support stiffness K_{Li} and the damping C_{S} of the isolators at the two boundaries are the same. K_{h} is the stiffness of the longitudinal springs. The results showed that the quasi-zero stiffness system has superior isolation performance over linear isolators for multimodal resonance of an elastic continuum. Moreover, the axial pressure at both ends of the elastic continuum could significantly improve the vibration isolation performance. Zhang et al.[106] proposed to introduce inertial NESs into the ends of the elastic supported beam. The results showed that this control scheme integrating vibration absorption and vibration isolation could effectively suppress the elastic vibration of the beam.

$$\begin{cases} \underbrace{\eta_{\mathrm{L}} = \max\left(\dfrac{\underbrace{K_{\mathrm{Li}}U(0,T) + K_{\mathrm{h}}\dfrac{L_0}{L_{\mathrm{h}}^3}U(0,T)^3}_{\text{Linear and nonlinear restoring forces}} + \underbrace{C_{\mathrm{S}}\dfrac{\partial U(0,T)}{\partial T}}_{\text{Damping force}}}{F_0L/2}\right)}_{\text{Vibration transmissivity of the left boundary}}, \\ \underbrace{\eta_{\mathrm{R}} = \max\left(\dfrac{K_{\mathrm{Li}}U(L,T) + K_{\mathrm{h}}\dfrac{L_0}{L_{\mathrm{h}}^3}U(L,T)^3 + C_{\mathrm{S}}\dfrac{\partial U(L,T)}{\partial T}}{F_0L/2}\right)}_{\text{Vibration transmissivity of the right boundary}}, \\ \underbrace{K_{\mathrm{Li}} = (K_{\mathrm{v}} - 2K_{\mathrm{h}}(L_0/L_{\mathrm{h}} - 1))}_{\text{Linear elastic part of boundary conditions}}. \end{cases} \tag{10}$$

2.3.2 Isolation of transverse vibration of pipes

Ding et al.[15] introduced nonlinear stiffness and nonlinear damping into the vibration isolation of fluid-conveying pipes, as shown in Fig. 15, under the harmonic vibration of the base. The transmission of the displacement excitation of the base to the fluid-conveying pipe was isolated by an isolator. The stiffness of the vibration isolation system was composed of two horizontal linear springs and one vertical linear spring, and the nonlinear damping was composed of two horizontally placed linear dampers, where L_0 and L_{h} are, respectively, the initial length and the assembly length of the longitudinal springs. C_{h} is the longitudinal damping coefficient. ω_0 and D are, respectively, the frequency and amplitude of the vibration of the base. K_{v} is the stiffness coefficient of the vertical support spring of the nonlinear isolator. For the isolation of the steady-state response of the pipe under the harmonic displacement excitation, the fluid flow in the pipe would deteriorate the isolation performance of the quasi-zero stiffness isolator on the transverse vibration of the pipe, as shown in Fig. 16. In 2021, Xu et al.[107] designed a multi-dimensional vibration isolation device combining viscoelastic core pads, viscoelastic plate dampers, and U-shaped dampers.

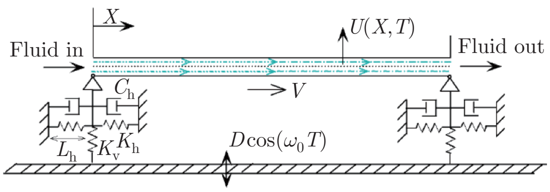


Fig. 15 A fluid-conveying pipe with nonlinear isolators[15] (color online)

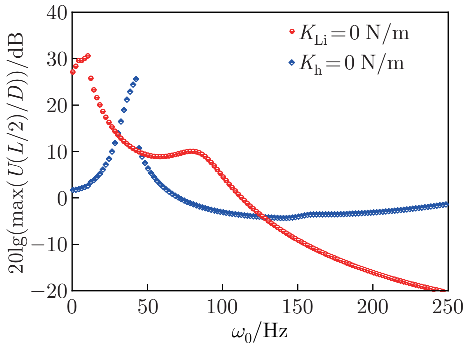


Fig. 16 Effects of the stiffness of the nonlinear isolators on the isolation performance of the fluid-conveying pipe[15] (color online)

The research showed that the quasi-zero stiffness system has a good vibration isolation performance on the transverse vibration of elastic beams[103,105]. In the vibration isolation of high-velocity pipes, the quasi-zero stiffness system was found to have a relatively good isolation performance in the local frequency range, and the first-order resonance peak could shift significantly to the low frequency region. However, due to the influence of fluid flow, the quasi-zero stiffness system did not show obvious benefits to isolate pipe vibration[15]. Although there have been preliminary studies on transverse vibration isolation of continua, including fluid-conveying pipes, related issues still need to be systematically studied.

Based on the description in this section, combined with the engineering characteristics of fluid-conveying pipes, the vibration control of pipes at the boundaries is a feasible control strategy for fluid-conveying pipes from a wide range of engineering application prospects. Boundary control includes designing supports or restraints that increase system damping, function as dynamic vibration absorbers, or provide better vibration isolation. However, viscous and nonlinear boundary conditions are introduced in the dynamic analysis of fluid-conveying pipes, which increases the difficulty of design and optimization analysis. The vibration control method of suspending the DVA or covering a damping layer between the boundaries of the pipe is possible to achieve good control performance for the designated pipes that suffer from vibration problems. However, if it is used in large piping systems, the cost of vibration control would be high.

3 Active control

Modern technology has high requirements for vibration control. Active control of structural vibration is a method to rapidly attenuate the structural vibration response by externally supplying energy. When the structure is excited to produce a vibration response, the active control method will apply a control force or change the vibration characteristics of the structure. Active control of the vibration of fluid-conveying pipes has been investigated by using piezoelectric actuators, magnetic devices, and memory alloy materials.

3.1 Use of piezoelectric actuators

In the research of active control of fluid-conveying pipes, the actuators are mainly composed of piezoelectric materials. In 1996, Lin and Chu[108] proposed the use of piezoelectric ceramics as actuators to control the flutter of fluid-conveying cantilever pipes at a high velocity. Two piezoelectric actuators were bonded to the top and bottom of the pipe, as shown in Fig. 17, to provide an equivalent bending moment acting on the fluid-conveying cantilever pipe. The

Bernoulli-Euler beam theory was applied to the dynamic modeling of a slender cantilever pipe that was discretized into finite elements to describe its dynamic behavior under flowing fluid loads. Using optimal independent modal space control, the piezoelectric actuator could effectively control the flutter of the cantilever pipe by inputting a moderately actuated control voltage.

In 2001, Lin and Chu[109] studied active feedback control of flow-induced vibration of fluid-conveying pipes by using the Timoshenko beam theory. In 2004, Lin et al.[110] investigated the optimal independent modal space control of a fluid-conveying pipe fixed at both ends with a supercritical flow velocity, and applied the control torque onto the pipe by two independent arms. The numerical results showed that the vibration of the pipe could be further suppressed by switching the control force to a value higher than the response mode. However, more control input and fast-response actuators were required to meet the control requirements. In 2013, Lin et al.[111] numerically analyzed the stability of independent modal vibration control of fluid-conveying pipes by using a piezoelectric inertia actuator placed at the free end.

In 1997, Tsai and Lin[112] used a model reference adaptive control approach in a modal space to control the flutter of fluid-conveying cantilever pipes. The obtained results showed that the model-referenced adaptive control was more robust than the optimal independent mode-space control in terms of flow velocity changes. In 1997, Lin and Tsai[113] proposed an instantaneous optimal closed-loop control approach for the vibration of fluid-conveying cantilever pipes, by considering the geometric nonlinearity caused by the supercritical flow of the fluid. It was shown that the position of the actuator was important for effectively controlling the vibration of the piping system. In 2003, in order to suppress bending propagating waves in infinitely long pipes, Variyart and Brennan[114] described the design of lead zirconate titanate (PZT) modal actuators for thin-walled fluid-conveying pipes. The expression of the radial motion of the pipe excited by the PZT actuator was deduced, and the transfer function of the PZT element and the theoretical model of the modal actuator were verified experimentally. In 2004, Variyart and Brennan[115] described an actuator and sensor arrangement that was theoretically demonstrated to suppress the vibrations of infinitely long fluid-conveying pipes. Experiments demonstrated the effectiveness of the control system.

In 2013, Shen et al.[116] proposed a passive/active hybrid control technique to control the vibration and in-structural acoustic transfer of fluid-filled cylindrical shells. The passive control method designed the shell as a periodic and discontinuous structure which is composed of epoxy pipes and aluminum pipes alternately, generating some frequency ranges in which harmonic elastic waves could not propagate freely. The active control method was realized by piezoelectric actuators pasted on the wall of the shell through the control strategies of inverse displacement feedback, velocity feedback, and acceleration feedback.

In 2014, Guan et al.[117] designed a piezoelectric direct-drive slide valve as an active vibration absorber for active control of fluid pressure pulsation. A basic adaptive optimal control method for single-frequency components and a stepwise optimal adaptive control method for dual-frequency components were proposed for the control of single-frequency and dual-frequency components of pressure pulsation, respectively. The amplitude and phase of the control signal were adjusted to minimize the mean square value of the pressure pulsation of the high-pressure and high-speed hydraulic pipe system. In 2015, Khajehpour and Azadi[118] used an adaptive robust control method to suppress the transverse vibration of a rotating cantilever pipe based on the local piezoelectric layer pasted on both sides of the pipe. Jin[119] presented the motion limiting nonlinear dynamics and frequency analysis of pipes reinforced with carbon nanotube agglomeration coupled with a piezoelectric actuator.

In 2020, Lyu et al.[120] proposed a tunable phononic crystal device based on an ultra-thin piezoelectric patch to control the excessive vibration of the fluid-conveying pipe. Piezoelectric patches are periodically pasted on the infusion pipeline with a rectangular cross-section, as shown in Fig. 18, to play the role of phononic crystals. The piping model is simplified to

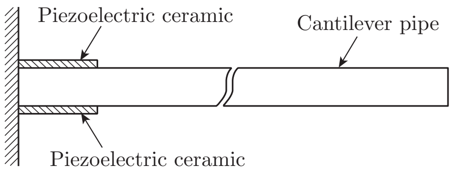


Fig. 17 A fluid-conveying cantilever pipe with surface mounted piezoelectric ceramics[108]

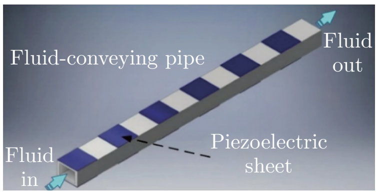


Fig. 18 A fluid-conveying pipe with periodic piezoelectric patches[120] (color online)

the Euler-Bernoulli beam, and the mechanical model is established as given by Eq. (11). The effective bandgap of phononic crystals was calculated by the transfer matrix method and the finite element method. The numerical results showed that the bandgap was hardly affected by the fluid flow velocity. However, a higher fluid density could produce lower and narrower frequency bandgaps. This ultra-thin vibration control method adhered to the pipe surface provided a promising solution for the vibration and noise control of long-distance fluid-conveying pipes.

$$\begin{cases} (\rho_{\mathrm{p}}A_{\mathrm{p}}+\rho_{\mathrm{f}}A_{\mathrm{f}})U_{,TT}+2\rho_{\mathrm{f}}A_{\mathrm{f}}VU_{,XT}+\rho_{\mathrm{f}}A_{\mathrm{f}}V^{2}U_{,XX}+EIU_{,XXXX} \\ \quad + \underbrace{M(X)U_{,TT}}_{\text{Inertia force of the PZT patch}} + \underbrace{D(X)U_{,XXXX}}_{\text{Restoring force of the PZT patch}} = 0, \\ M(X)=\begin{cases} 2E_{\mathrm{p}}I_{\mathrm{p}} & \text{for the pipe with the PZT patch,} \\ 0 & \text{for the pipe without the PZT patch,} \end{cases} \\ D(X)=\begin{cases} m_{\mathrm{p}} & \text{for the pipe with the PZT patch,} \\ 0 & \text{for the pipe without the PZT patch,} \end{cases} \end{cases} \tag{11}$$

where E_{p} is Young's modulus of the PZT patch. I_{p} is the moment of the inertia of the PZT patch. m_{p} is the PZT patch mass per unit length.

3.2 Use of magnetic devices

In 1996, Brennan et al.[121] designed a non-invasive, fluid wave hydraulic actuator driven by magnetostrictive elements for the active control of fluid-borne vibrations in a pipe. The actuator was mounted on a water-filled plexiglass pipe with integrated sensors. The study demonstrated the practical feasibility of non-invasive actuators and sensors to actively control fluid waves in pipes. In 2020, Amiri et al.[12] used an intelligent magnetostrictive layer as an actuator to implement the active control of the stability of a fluid-conveying cantilever micro-pipe. The intelligent magnetostrictive layer is laid on the upper and lower end faces of the rectangular section pipe, and the fixed layer is modeled as the Pasternak foundation, as shown in Fig. 19. The vibration characteristics and stability of the first three modes of the system were analyzed by eigenvalues, and the effect of the magnetostrictive layer on the critical velocity and frequency of the fluid-conveying cantilever pipe was studied. Numerical results showed that the added smart layer could well control the instability of the fluid-conveying micro-pipe.

In 2021, Chen et al.[122] studied the dynamic behavior of a fluid-conveying cantilever pipe made of hard-magnetic soft materials in an external magnetic field, as shown in Fig. 20. By considering the geometric nonlinearity caused by the bending deformation of the pipe, the dynamic equation of the pipe made of hard-magnetic soft materials was deduced based on the Hamilton principle. Through analyzing the stability, static deformation, and nonlinear vibration of the hard-magnetic soft material, it was found that the mechanical response of the fluid-conveying cantilever pipe made of hard-magnetic soft material could be effectively

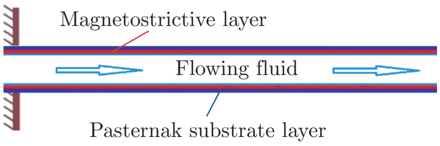


Fig. 19 A fluid-conveying micro-pipe with magnetostrictive layers[12] (color online)

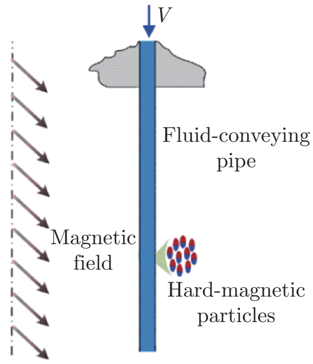


Fig. 20 A fluid-conveying cantilever pipe under a uniform external magnetic field[122] (color online)

controlled by applying an external magnetic field, including static deformation, instability, and vibration. In 2023, Dehrouyeh-Semnani[123] examined the nonlinear responses of a fluid-conveying cantilevered hard-magnetic soft pipe with uniform and non-uniform magnetizations under an actuating parallel magnetic field.

3.3 Use of other active control methods

Reasonably adjusting the flow velocity of the fluid in the pipe may be an effective way for controlling the vibration of the pipe. In 1996, Sugiyama et al.[124] conducted a cantilever pipe experiment to dissipate the vibration energy of the pipe by the flowing fluid inside the pipe whose velocity is smaller than the critical flutter velocity, to achieve the stabilization of the pipe. In 1999, Horiuchi et al.[125] proposed a compensation method to counteract the response delay of the actuator, and developed a real-time hybrid experimental system. The effectiveness of the compensation was demonstrated by the seismic experiment of the piping system.

In 2008, Yigit[126] discussed the applicability of active nodal vibration control into suppressing the flutter instability vibration of fluid-conveying cantilever pipes. The Euler-Bernoulli theory was used to model the pipe bending, and the finite element method was used to discretize the governing equations. The effectiveness of the active nodal vibration control was confirmed by comparing with the results obtained by the direct velocity feedback control. The active node vibration control method was essentially a model-based control method. Therefore, it could be affected by modeling error and model parameter uncertainty. In 2012, Bao[127] studied the active control of chaotic motion in a single-mode system with fixed supports at both ends of a fluid-conveying pipe under basic excitation. The simulation results showed that the single-mode chaotic motion of the pipe was successfully controlled as a periodic motion through the notch filter feedback control.

In 2015, Mani and Senthilkumar[52] proposed an adaptive passive control dynamic vibration absorber through shape memory alloy springs. Young's modulus of shape memory alloys was temperature-dependent, which was utilized to adaptively change the spring stiffness to control vibration. The elasticity of the shape memory alloy was changed by the current input to realize the adaptive control of the vibration absorber for different excitation frequencies. A microcontroller-based control system was developed to activate the shape memory alloy in time and provided the optimal current to the parallel shape memory alloy spring for vibration control of the pipe. In 2022, Shaik et al.[128] effectively controlled the parametric instabilities of flexible pipes conveying pulsating fluid by applying shape memory alloy actuation. The shape memory alloy wire actuator caused a significant reduction in the region of principal primary instability.

In 2019, Li et al.[129] proposed a feedforward vibration control method for fluid-conveying cantilever pipes to minimize the vibration displacement at the end. Experiments showed that the vibration displacement at the end of the fluid-conveying pipes of the concrete pump truck was reduced by more than 80%. In 2022, Zhang et al.[19] studied fractional proportional integral control to reduce the vibration of the typical pipeline system of a nuclear power plant.

From the above discussion, it is easy to notice that active control methods, whether based on piezoelectric actuators[108,117,119–120], by adjusting the magnetic field[12,122], or by other methods such as memory alloy materials[52,128], can effectively control the vibrations of fluid-conveying pipes. In engineering applications, pipes usually exist in piping systems, and many pipes are used together to achieve the purpose of conveying fluids. By considering different shapes of pipes, it would be very expensive and challenging to implement the active control of the vibration of each pipe.

4 Semi-active control

Semi-active control can change parameters such as stiffness or damping of the structure in real-time by using a small amount of energy to reduce the vibration intensity of the structure. It does not require a large amount of external energy input to provide the control force. To a certain extent, semi-active control may be able to achieve a similar control performance to that of active control. Semi-active control technology has also been adopted to control the vibrations of fluid-conveying pipes.

Semi-active variable stiffness dampers can dynamically switch the control valve of the damper on or off according to the vibration response of the pipe, change its stiffness, and increase the control effect on the pipe system accordingly. In 2013, Kumar et al.[130] studied the performance of semi-active variable stiffness dampers in switching control law, and modified switching control law for the vibration control of a pipe. Straight members in the piping system were modelled as 3D beam elements, and the bends were modeled as 3D elbows having six degrees-of-freedom at each node. According to the feedback information of the response of the piping system, the control device would close or open the control valve of the energy dissipation device connected to the support of the piping system. The vibration responses of the pipe with and without the control system were compared to demonstrate the performance of the semi-active variable stiffness damper on the vibration control of the pipe system under different seismic motions. The results showed that the semi-active variable stiffness damper with optimal parameters was very helpful for the seismic response control of the pipe system.

In 2017, Kavianipour[131] proposed an electromagnetic damper composed of a permanent-magnet direct current (DC) motor, a ball screw, and a nut. The designed semi-active electromagnetic damper was controlled by the on-off damping control strategy. The vibration energy was converted into electrical energy, while the pipe vibration caused by the fluid flow was controlled. It was shown that the designed semi-active electromagnetic damper had a better vibration control performance and had more energy conversion regeneration than the passive electromagnetic damper.

Electromagnetic devices of the transformer type have shown to be effective in active stabilization of non-conservative systems. In 2018, Pisarski et al.[132] studied the application of electromagnetic devices to improve the dynamic stability of air-conveying pipes. The movement of electromagnetic devices along the pipe would generate eddy currents in the plates and a drag force of viscous character, as shown in Fig. 21. The combined effect of the additional mass and the viscous electromagnetic force could significantly affect the dynamic characteristics of the fluid discharged from the cantilevered pipe. The control performance was evaluated in three speed ranges, i.e., low subcritical, high subcritical, and low supercritical, for pipe systems.

In 2019, Szmidt et al.[133] proposed eddy-current dampers to improve the dynamic stability of a cantilevered pipe discharging fluid. A closed-loop control system was designed, and the

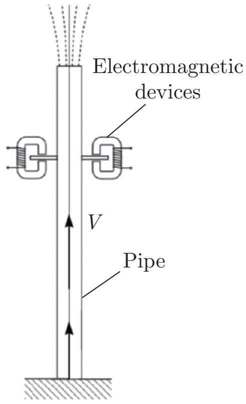


Fig. 21 A fluid-conveying cantilever pipe with electromagnetic devices[132]

experiments were conducted. It was shown that installing an actuator at approximately 60% of the length of a vertical cantilevered pipe could increase the critical flow velocity by 25%. In 2020, Ye et al.[134] designed a semi-active control method with additional valves, and conducted a series of experimental tests for the noise control of a scroll compressor in an air conditioning unit. It was found that noise levels could be reduced by using additional valves to adjust the rate of pressure change.

By summarizing the above research, it can be found that similar to the active control method, the semi-active control can achieve vibration control for different types of fluid-conveying pipes[130–134]. However, it would be difficult to realize the semi-active control of each pipe in the pipe system composed of pipes of different shapes.

5 Optimal design of structural configuration for vibration attenuation

For fluid-conveying pipes in engineering applications, their requirements are basically predetermined before the design and installation. During the operation, the range of the excitation frequency of a specified fluid-conveying pipe is basically fixed. Therefore, in order to change the natural frequency and modal characteristics of the vibration of fluid-conveying pipes, the pipe parameters, the conditions, and the location of supports can be modified by theoretical analysis and experimental methods. Avoiding possible resonance can be achieved, and the vibration control of the fluid-conveying pipes can be realized. In 2001, Biswas and Ahmed[135] used the minimum integral principle of Pontryagin to find the optimal flow velocity with minimal vibrations in a fluid-conveying pipe.

In 2014, Zang and Gao[136] conducted a series of experiments on the steady-current induced vibration of a piggyback pipe close to a plane seabed. The configuration parameters of the piggyback pipe were found to have a significant effect on the vortex-induced vibration suppression. In 2019, Wang et al.[137] proposed an irregular cone to control the ice-induced vibration of a vertical riser pipe. In 2020, Liu et al.[138] established a hoop support stiffness model with non-uniformly distributed springs and a hoop support damping model with uniformly distributed dampers. By taking the position of the annular hoop as the design variable as the optimization goal, the optimization of hoop layouts was realized based on the genetic algorithm to reduce the resonance of the pipe. In 2022, Xu et al.[2] discussed the influence of different arrangements of pressure pipes on the vibration of the pumping station pipe, and proposed an optimized design scheme. In 2021, Bamidele et al.[139] studied the effect of flow restricting orifices on pipe vibrations. In 2021, Liu et al.[3] conducted vibration analysis and vibration tests on the station pipe of a natural gas pipe system, and proposed a vibration reduction method for optimizing the pipe diameter. In 2022, Liu et al.[140] focused on the problem of abnormal vibrations within a compressor station pipe, and used a 3D calculation model of the launcher pipe of a gas pipe system to perform a simulation analysis. They found that the fluid pressure fluctuation in the pipe was the main factor for the abnormal vibration of the launcher pipe in the compressor station. The main causes of the vibrations were the excessive fluid flow and high flow velocity. Also, by comparing and analyzing the natural frequency of the pipe and the pressure fluctuation frequency of the vortex core in the pipe, they found that the pressure fluctuation frequency is close to the low-order natural frequency of the pipeline system, which is prone to resonance. Three vibration reduction schemes were suggested and verified.

In 1998, under the assumption that the main frequency components in the excitation load were known, Koo and Park[141] showed that the design of periodic supports could effectively reduce the vibrations of fluid-conveying pipes. In 2016, Wei et al.[142] designed a one-dimensional periodic composite structure for supporting the ship's hydraulic pipe, which was composed of metal and rubber and used to control the vibration transmission of the hydraulic pipe to the bulkhead. Theoretical and experimental results showed that more than 20 dB of attenuation was realized in the bandgap frequency range. Yu et al.[143] designed a periodic binary composite fluid-conveying pipe, which consisted of infinitely repeating alternating pipe elements, as shown in Fig. 22. Yu et al.[54,143] calculated the band structure of flexural waves, and studied the gap frequency range based on the Euler-Bernoulli beam theory in 2008 and based on the Timoshenko beam theory in 2011. In 2022, Iqbal et al.[144] presented a pipe supported periodically on a rack structure to design vibration control strategies. Li et al.[145] proposed the clamps' locations optimization to minimize the system impedance at vibration frequencies. In 2023, Wu et al.[146–147] studied the vibration bandgap characteristics of a periodic composite pipeline and a pipe with periodic torsional support. Zhang et al.[148] proposed a measurement-based layout optimization method. The proposed method was used to optimize the support layout of an L-shaped pipeline system for solving the resonance problem.

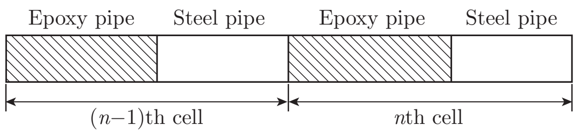


Fig. 22 A periodic binary pipe[143]

In 2019, based on the frequencies corresponding to the main vibration modes of the pipe, Nateghi et al.[55] experimentally studied the aluminum pipe with periodic resonance structure. A resonant structure was fabricated by a polymethyl methacrylate panel, as shown in Fig. 23. If the excitation source was known, better vibration control would be achieved with a smaller number of resonators. In 2020, Lyu et al.[120] investigated the control of the vibration and sound transmission of a pipe by periodically pasting ultra-thin piezoelectric sheets on the surface of the pipe. In 2021, El-Borgi et al.[57] developed dynamic modeling of the resonator with a cantilever beam with a concentrated mass at the tip, and controlled certain bending mode resonances of the pipe by tuning the resonant element. In 2022, Liang et al.[149–150] proposed a novel fluid-conveying phononic crystal pipe model of a pipe composed of different materials arranged alternately. An axially spinning motion was considered. It was found that different pseudo Bragg bandgaps existed in the two transverse directions, while the effective Bragg bandgaps were actually located in their coupled regions, in which the vibration was self-suppressed. It was also found that the spinning motion could reduce the effective Bragg bandgap regions of the periodic pipe. In 2021, combining the Bragg scattering bandgap mechanism and the inertial amplification mechanism, Shoaib et al.[151] proposed a strategy to enhance the bandgap characteristics of periodic inhomogeneous fluid-conveying pipes. It was shown that the piping system with periodic piping structure design could effectively reduce vibration. The fluid velocity was found to have little influence on the bandgap characteristics of a non-uniform fluid transport pipe. In 2022, Fernandes et al.[152] investigated the effects of fluid flow velocity on the edge frequencies of the bandgap in an acoustic metamaterial pipe.

Pipe-in-pipe arrangements are mostly used in subsea pipes, consisting of concentric inner and outer pipes. The outer pipe provides mechanical protection to withstand the pressure of external seawater. In order to enhance the seismic performance of subsea pipes, Bi and Hao[153] proposed an improved method for pipe-in-pipe design in 2016. Adding springs and dampers in the annular space between the inner pipe and the outer pipe would make the pipe-in-pipe system into a dynamic vibration absorber, with the outer pipe as the primary system and the inner pipe as the dynamic absorber mass, as shown in Fig. 24. Numerical results showed that the improved pipe-in-pipe system significantly reduces the vibration of the outer and inner pipes. In 2017, Nikoo et al.[154] used the improved pipe-in-pipe design to control the vortex-induced vibration of the pipe. In 2018, Bi et al.[155] utilized rotational friction hinge dampers with springs to connect the inner and outer pipes. The effects of the bolt preload, the friction coefficient, and the spring stiffness on the seismic response of the pipe-in-pipe system were studied.

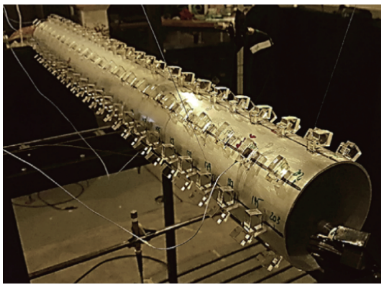


Fig. 23 Test set-up of a pipe with periodic resonant structures[55] (color online)

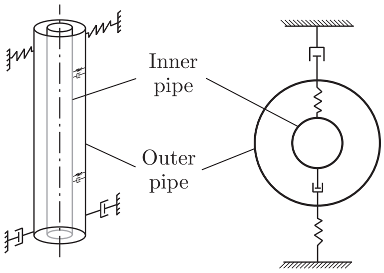


Fig. 24 Test set-up of a pipe with periodic resonant structures[153–154]

To summarize, for the specified pipe systems, the vibration of the fluid-conveying pipes can be controlled through the optimization of pipe layout based on dynamic analysis and experiment under the conditions of known working environment[2–3,32,140–141,148]. On the other hand, when installation conditions allow, the vibration of fluid-conveying pipes can also be controlled by carefully designing the shape, material, and composition of the pipe[53–54,146,151,153].

6 Concluding remarks

This section will be organized in three parts to summarize the research advances in the vibration control of fluid-conveying pipes and to recommend the potential future research directions. First, the main research advances in the vibration control of fluid-conveying pipes are summarized. Then, the literature on the vibration control of fluid-conveying pipes is classified and discussed. Finally, based on the research results and progress, some key issues for future research and development are suggested.

6.1 Main advances in vibration control of fluid-conveying pipes

Based on the publication year and research contributions of the literature on the vibration control of fluid-conveying pipes, some main advances are summarized in Table 1.

Table 1 Key advances in vibration control of fluid-conveying pipes

Control technique	Reference	Year	Key contribution
Passive control	Yang et al.[67]	2014	Design of an NES to control the transverse vibration of the fluid-conveying pipe without adding a new resonance frequency
	Song et al.[58]	2016	Design of impact dampers to control vibration in large piping systems
	Duan et al.[76]	2021	Design of an enhanced NES using inerter to control vibration of the fluid-conveying pipe
	Mao et al.[83]	2021	Design of a torsional NES placed at the boundary of the pipe to control the transverse multi-modal resonance of the pipe without changing the resonance frequency of the fluid-conveying pipe
	Ding et al.[15]	2019	Determination of the transmissibility of the transverse vibration of the fluid-conveying pipe, and investigation into the effect of the isolator with quasi-zero stiffness and nonlinear damping on the vibration isolation of pipe
	Bi and Hao[91]	2016	Use of viscoelastic materials laid on the outer surface of the pipe to control the earthquake-induced vibration of the pipe on the ground
	Yano et al.[94]	2019	Use of silicone as a viscoelastic material with a certain width on the outer pipe wall, to act as additional damping and dynamic vibration absorber for vibration control of the pipe
	Sabahi et al.[92]	2022	Use of functionally graded porous micro-pipes to convey fluid embedded in a viscous damping medium
	Yu et al.[54]	2008	Use of the concept of phononic crystals to achieve infinitely repeating alternating periodic binary composite fluid-conveying pipes and local resonance pipes with periodic oscillators
	Nateghi et al.[55]	2019	Fabrication of a local resonance structure using a polymethyl methacrylate panel, and use of a metamaterial aluminum pipe to predict the frequency range of the stopband zone
	Liang et al.[149]	2022	Development of a phononic crystal fluid-conveying pipe model composed of different materials arranged alternately by taking the axially spinning motion into account
	Liu et al.[138]	2020	Establishment of the stiffness model of hoops with uneven distribution, and development of a genetic algorithm to optimize layouts of multiple hoops to reduce the vibration of the pipe
	Bi and Hao[153]	2016	Utilization of the spring and damper in the annular space between the outer and inner pipes to design a pipe-in-pipe system with dynamic vibration absorber performance
Active control	Brennan et al.[121]	1996	Design of hydraulic actuators driven by magnetostrictive elements and sensors integrated into the pipe coupled with fluid motion to control fluid waves in the pipe
	Lin and Chu[108]	1996	Use of piezoelectric ceramics as an actuator to actively control the flutter of a high-speed fluid-conveying cantilever pipe
	Variyart and Brennan[114]	2003	Use of PZT actuator to suppress the propagation of a certain order mode bending propagating wave in an infinitely long pipe
	Guan et al.[117]	2014	Use of piezoelectric direct-drive slide valve as active damper to control fluid pressure pulsation in high-pressure and high-speed hydraulic pipelines
	Lyu et al.[120]	2020	Use of an ultra-thin piezoelectric patch periodically attached to the pipe to form an effective bandgap and control the vibration and sound propagation of the pipe
	Chen et al.[122]	2021	Control of the dynamic behavior of the pipe by applying a magnetic field in a fluid-conveying cantilever pipe made of hard-magnetic soft materials

continued

Control technique	Reference	Year	Key contribution
Semi-active control	Kumar et al.[130]	2013	Use of a semi-active variable stiffness damper to control the seismic response of a 3D fluid-conveying pipe system
	Pisarski et al.[132]	2018	Design of an electromagnetic actuator to control the stability of a vertical cantilever pipe in three flow velocity ranges
	Szmidt et al.[133]	2019	

6.2 Classification of research literature

In the vibration control of fluid-conveying pipes, the research objects mainly include three types of pipe structures, i.e., cantilevered pipes, curved pipes, and straight pipes constrained at both ends. For the fluid-conveying cantilevered pipes, some researchers have explored the active control methods based on piezoelectric[108,112,118] and electromagnetic actuators[12,121–122,132], the transient optimal closed-loop control method[113], the active control methods based on changing the fluid flow velocity[97] and the fluid pressure[129], and the adaptive dynamic vibration absorber control method using the shape memory alloy spring[51,128]. Simultaneously, passive control methods including the use of linear dynamic vibration absorbers[70], NESs[69], and multi-cavity particle damper[101] have been studied and introduced into fluid-conveying cantilever pipes.

For the fluid-conveying pipes of nuclear power plant, the vibration control methods mainly focus on introducing damping to increase the vibration energy dissipation[17,83,85–86,88–89,91,102,156], utilizing linear dynamic vibration absorber[57–58,157–158], using semi-active[130] and active[19] vibration control, and using vibration isolator[107].

For the subsea fluid-conveying pipes, vibration control is achieved by designing the pipe configuration[136–137,153–155] and introducing linear dynamic vibration absorbers[52,58,153–155].

For hydraulic system pipes, vibration control methods mainly focus on adhering piezoelectric sheets[159], controlling fluid pressure[117], regulating flow velocity pulsation[26,121,160], regulating resonance[16], and using the phononic crystal theory to optimize the support design[142].

In studying the vibration control of compressor fluid-conveying pipes, the control methods adopted include semi-active control[134], pipe configuration optimization[3], pressure pulsation reduction[161], active vibration control based on fluid motion[121], and passive control with the introduction of dynamic vibration absorbers[59].

In suppressing the vibration of pipes at pumping stations, parameters such as the pipe shape and size were optimized[2], and the vibration reduction measures for pipes of a natural gas compressor station were studied[3,140].

The vibration control theory of fluid-conveying pipes has been developed mainly based on the linear theory. Nonlinear vibration control methods have also been developed recently, including the introduction of NESs to the control of pipe vibration[67–69,71–80,82] and the use of nonlinear vibration isolator[15].

With the bandgap concept of phononic crystals[53,116,120,141–143,146–147,149,151] and the concept of local resonance[54–56,152], advanced design methods have been introduced into the configuration design to reduce vibration and sound propagation in pipes.

The above literature on active vibration control, semi-active vibration control, and passive control methods to control the vibration of fluid-conveying pipes provides many options to control the pipe vibrations.

6.3 Suggestions for future research

Based on the above review and summary of literature, the following six promising topics are outlined for future research endeavors to be undertaken concerning the vibration control of fluid-conveying pipes.

(i) New and efficient vibration control strategies will always be an important research topic for the vibration control of fluid-conveying pipes. In particular, improving the design of the restraint (support) elements of the pipe can result in a better damping performance[82,138]. It is a research topic of vibration control of pipes that is easier to be implemented to engineering pipes.

(ii) The development of a fluid-structure interaction modeling method and a fast and efficient analysis method for vibration response[38,103–104] is an important research activity to optimize the vibration control design of fluid-conveying pipes, especially for the multi-dimensional vibration modeling and analysis of pipes[22,162–164]. Understanding of vibration features in multi-dimensions will enable the design of effective methods to control the dominant vibrations.

(iii) The design and application of nonlinear vibration absorbers for the vibration control of fluid-conveying pipes are very promising research topics. Nonlinear vibration absorbers, especially NESs, have shown excellent performance in structural vibration control research, and remain an attractive research topic[62,165].

(iv) Pipes in engineering applications are mostly in the form of integrated piping systems[1,39]. The vibration control of complex piping systems composed of multiple pipes is an unavoidable and more challenging issue. It is desirable to use simple control methods (less control effort) to realize the vibration control in practical engineering applications.

(v) There are many excitation sources in large mechanical systems[17,26,44], and it is difficult to identify them. When there are uncertain multiple excitation sources coexisting in the system, how to control the vibration of the pipe is a difficult but necessary research task.

(vi) Fluid-conveying pipes are characterized by long-term use and with difficulty in predicting damage[166–167]. Under high-frequency excitations, even a small amplitude of vibration can lead to a safety hazard during long-term operation[1,82,168]. Controlling high-frequency micro-amplitude vibration is a frontier topic on the vibration control of pipes.

In summary, this review attempts to summarize the research achievements in the field of the configuration design, modeling, and vibration control of fluid-conveying pipes. In addition, future promising topics are suggested. Hopefully, the review will serve as a collection of ideas and a source of inspirations for further investigations and applications of vibration control of pipes.

Conflict of interest Hu DING is an editorial board member for *Applied Mathematics and Mechanics* (*English Edition*) and was not involved in the editorial review or the decision to publish this article. The authors declare no conflict of interest.

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