CHIN. PHYS. LETT. Vol. 33, No. 4 (2016) 044303

Theoretical and Experimental Investigation of Flexural Vibration Transfer Properties of High-Pressure Periodic Pipe

Zhen-Dong Wei(魏振东), Bao-Ren Li(李宝仁), Jing-Min Du(杜经民)**, Gang Yang(杨钢)

School of Mechanical Science and Engineering, and the FESTO Pneumatic Center, Huazhong University of Science and Technology, Wuhan 430074

(Received 1 December 2015)

According to the theory of phononic crystals, the hydraulic pipeline is designed to be a periodic structure composed of steel pipes and hoses to suppress the vibration of the hydraulic system with band gaps. We present theoretical and experimental investigations into the flexural vibration transfer properties of a high-pressure periodic pipe with the force on the inner pipe wall by oil pressure taken into consideration. The results show that the vibration attenuation of periodic pipe decreases along with the elevation of working pressure for the hydraulic system, and the band gaps in low frequency ranges move towards high frequency ranges. The periodic pipe has good vibration attenuation performance in the frequency range below 1000 Hz and the vibration of the hydraulic system is effectively suppressed. All the results are validated by experiment. The experimental results show a good agreement with the numerical calculations, thus the flexural vibration transfer properties of the high-pressure periodic pipe can be precisely calculated by taking the fluid structure interaction between the pipe and oil into consideration. This study provides an effective way for the vibration control of the hydraulic system.

PACS: 43.40.+s, 62.30.+d, 62.50.−p, 07.10.Fq DOI: 10.1088/0256-307X/33/4/044303

In hydraulic systems, vibration and noises can not only reduce the service life of mechanical equipment but also be harmful to human health. Since the hydraulic pipeline serves as the main transfer path of vibration and noises, it is necessary to take some vibration control measures for the hydraulic pipeline to restrain the transfer of vibration and noises. In recent years, the theory of phononic crystals (PCs)[1−5] has received considerable attention and the relevant studies are focused on the existence of band gaps (BGs) within which both noises and vibration are effectively suppressed.[6−11] Accordingly, the hydraulic pipeline is designed to be a periodic structure to reduce vibration and noises of the hydraulic system with BGs.

Various periodic pipe structures have been studied theoretically in the past. Sorokin *et al.*[12] modeled the liquid-conveying pipe wall as a periodic cylindrical shell structure, derived the band structure of the infinite periodic pipe with Floquet theory, and analyzed the energy transmission in the periodic pipes which conveyed or did not convey fluid using boundary integral equations. Yu *et al.*[13] designed two straight liquid-conveying periodic pipe structures in line with Bragg scattering mechanism and the locally resonant (LR) mechanism respectively, studied the flexural vibration transfer properties and the effects of fluid velocity on BGs, and found that the frequency ranges of BGs went up along with the rise of fluid velocity. Shen *et al.*[14] designed a 3D periodic pipe by the Bragg scattering mechanism, calculated the band structure and the vibration transfer properties by the transfer matrix (TM) method and the finite element (FE) method, and demonstrated that there existed BGs in periodic pipes which could suppress vibration effectively. However, the aforesaid studies still lacked experimental validation. Later, Wen *et al.*[15] validated the vibration attenuation ability of the straight low-pressure liquid-conveying periodic pipe by experiment, while there were great differences between experimental results and numerical calculations due to the fact that the researchers did not give consideration to fluid structure interaction (FSI) in theoretical analysis.

To our knowledge, no study on the periodic pipe with FSI considered has been reported so far. In this Letter, the FSI between pipe and oil is taken into account in calculating the flexural vibration transfer properties of 3D high-pressure periodic pipe. The 3D high-pressure periodic pipe is made up of steel pipes and hoses. The band structure and the frequency response function (FRF) of the periodic pipe are calculated by the TM method and the FE method, and the effects of the working pressure on BGs and the vibration transfer properties of the periodic pipe are studied. The 3D pipe vibration experiments are performed.

An infinitely extending periodic pipe is shown in Fig. 1. The lengths of steel pipe and hose are denoted as a_1 and a_2 respectively, the lattice constant is $a = a_1 + a_2$, the internal and external diameters of the periodic pipe are d_1 and d_2, respectively, and the number of cells is n. A coordinate system is established, of which the x-axis is parallel to the pipe axis.

Based on the Euler beam model, the differential equation for the flexural vibration of the pipe can be written as[16,17]

$$EI\frac{\partial^4 f_y}{\partial x^4} + \mathrm{m_f}v^2\frac{\partial^2 f_y}{\partial x^2} + 2m_f v\frac{\partial^2 f_y}{\partial x \partial t} + (m_\mathrm{f} + m_\mathrm{p})\frac{\partial^2 f_y}{\partial t^2} = 0, \quad (1)$$

where f_y is the flexural displacement, v and$t$ are the fluid velocity and time, respectively, I is the area moment of inertia with respect to the axis perpendicular to the pipe axis, m_f and m_p are fluid and pipe masses per unit length respectively, and E is Yong's modulus of the pipe.

**Corresponding author. Email: 87740661@163.com

© 2016 Chinese Physical Society and IOP Publishing Ltd

In accordance with the Bloch theorem,[18] a periodic boundary is applied in the x-direction, the eigenvalues of the infinite periodic pipe without considering the flow velocity are the roots of the equation

$$|\boldsymbol{T} - e^{ika}\boldsymbol{I}| = 0, \tag{2}$$

where $\boldsymbol{I}$ is the 4×4 unit matrix.

For a given frequency ω, the corresponding wave vector k can be calculated by Eq. (2) and further BGs of flexural vibration of the infinite periodic pipe can be derived by judging whether k is real or has a nonzero imaginary part.

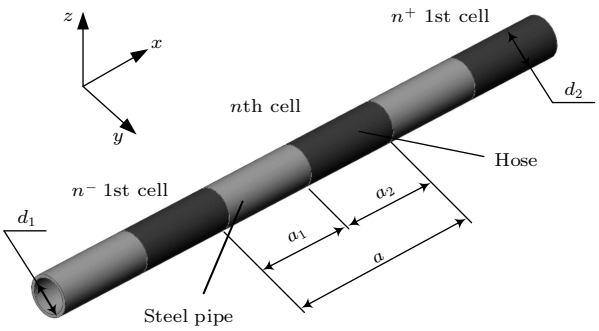

Fig. 1. The infinite periodic pipe structure.

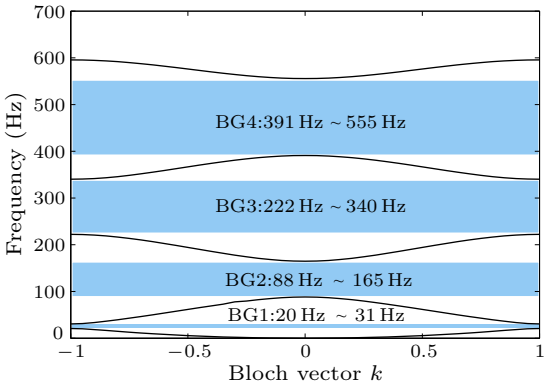

Fig. 2. The flexural band structure of the periodic pipe.

In this study, the equivalent physical parameters of the steel pipe and the hose of the periodic pipe are listed in Table 1. The lattice constant $a = 1000$ mm, the lengths of the steel pipe and the hose are $a_1 = a_2 = 500$ mm, and the internal and external diameters of the periodic pipe are $d_1 = 32$ mm and $d_2 = 42$ mm, respectively. The fluid conveyed by the periodic pipe is hydraulic oil.

Table 1. The equivalent physical parameters of the pipes.

	Density ρ (kg·m^{-3})	Young modulus E (Pa)	Poisson's ratio μ
Steel pipe	7780	2.106×10^{11}	0.31
Hose[19]	2800	7.5×10^{8}	0.33

Figure 2 illustrates the flexural band structure calculated by Eq. (2). In the whole frequency range, there are four complete BGs which lie in the frequency ranges of 20–31 Hz, 88–165 Hz, 222–340 Hz and 391–555 Hz, respectively. The periodic pipe is a phononic crystal structure based on the Bragg scattering mechanism, and the larger its dimension, the lower the frequency ranges band gaps are located in. In this study, the lattice constant $a = 1000$ mm for the periodic pipe, thus band gaps appear in low frequency ranges. The periodic pipe causes vibration to scatter on the one hand and be obstructed in the transfer path on the other hand, and the vibration within the frequency range of BGs is suppressed.

The FRF is used to describe the vibration transfer properties.[20,21] Due to the high working pressure of the hydraulic system, the force on the inner pipe wall applied by the oil pressure is considered in the FSI model of the periodic pipe in this work.

In calculation, one end of the periodic pipe is set as the velocity inlet, and the other end is set as the pressure outlet. The other faces are set as the wall type. The pressure of flow fields is figured out first and then applied on the inner pipe wall, as shown in Fig. 3(a). There are no boundary limitations at both ends of the periodic pipe, and the white noise signal is excited at the end of velocity inlet of the pipe in the y-direction. The FRF of the periodic pipe can be obtained by[22,23]

$$T = 20 \log \frac{a_2}{a_1}, \tag{3}$$

where a_1 and a_2 are the vibration accelerations at the ends of the inlet and the outlet, respectively.

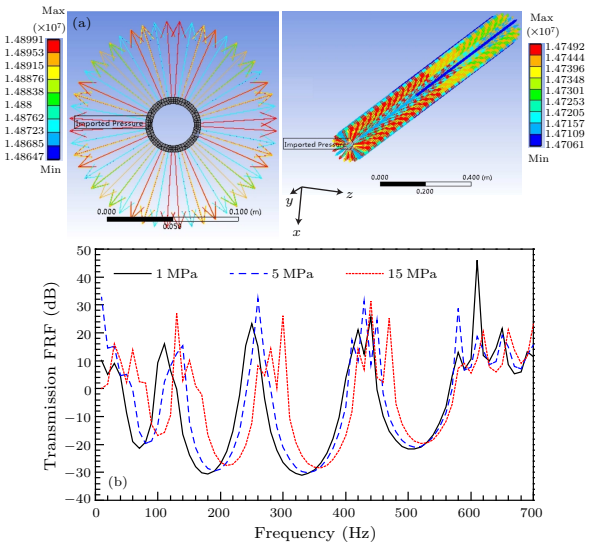

Fig. 3. The FRF of the straight periodic pipe.

In Fig. 3(b), the solid, the dashed and the dotted lines represent the FRFs of the periodic pipe under the oil pressures of 1 MPa, 5 MPa and 15 MPa, respectively. When the oil pressure is 1 MPa, there are four sharp drops of FRFs in Fig. 3, which indicate the existence of four complete BGs in the frequency ranges of 42–95 Hz, 129–232 Hz, 267–398 Hz and 449–567 Hz, respectively. Moreover, it can be found in Fig. 3(b) that BGs in low frequency ranges move towards high frequency ranges along with the elevation of oil pressure, due to the fact that the elevation of oil pressure leads to the increase of the natural frequency of the periodic pipe. Figure 3(b) also shows that the vibra-

tion attenuation of the periodic pipe decreases with increasing the oil pressure.

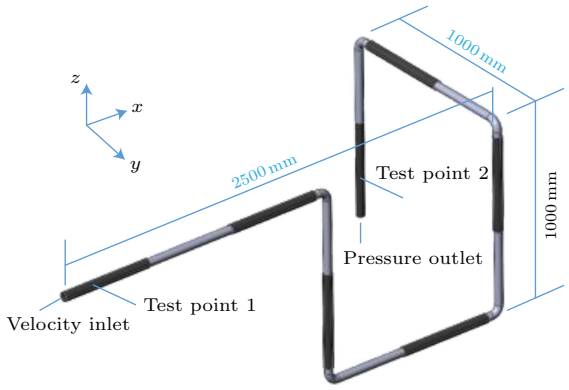

Fig. 4. The FSI model of the 3D periodic pipe.

Fig. 5. The experimental measurement setup.

The hydraulic pipeline is usually set along the wall, which features a complicated structure. In this study, the spatial dimensions of the 3D periodic pipe are 2.5 m×1 m×1 m and the total length is 6.5 m. According to the vibration transfer properties of the straight periodic pipe, the FSI model of the 3D periodic pipe is built as shown in Fig. 4. In the calculation, the oil velocity is fixed at 2 m/s, and the oil pressure is fixed at 1 MPa. The numerical calculation results are shown in Figs. 8 and 9 in comparison with the experimental results.

To further verify the numerical calculation results, the vibration transfer experiments are performed. As shown in Fig. 5, the experimental platform consists of a hydraulic oil tank, the test sample and the data acquisition system, etc. The test sample is made of steel pipes and hoses which are connected with pipe joints. The test sample and the axial directions are shown in Fig. 6. There are some foam between the test sample and the support to reduce the constraint by the support.

Figure 7 shows the schematic diagram of the experimental measurement. After the magnetic valve is opened, the oil pressure can be adjusted by overflow valve 1, and the oil velocity can be fixed at 2 m/s by overflow valve 2. Under such conditions, the compound vibration of the hydraulic oil tank is caused by many factors, e.g., the motor rotor imbalance, cavitations and pump pressure pulsations, and propagated along the pipeline. Two PCB vibration acceleration sensors are respectively stalled on both ends of the test sample, which can collect the vibration acceleration signals by the data acquisition front-end (LMS SCADAS mobile). The positions of sensors are shown as points 1 and 2 in Fig. 4. The FRFs of the test sample can be obtained by Eq. (3).

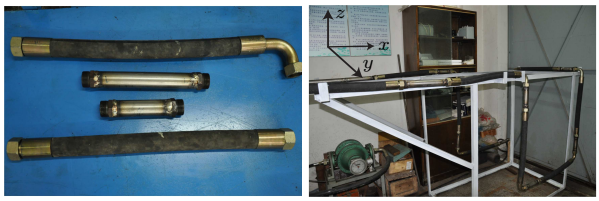

Fig. 6. The test sample.

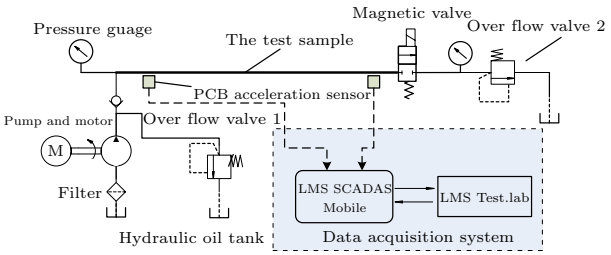

Fig. 7. Schematics of the experimental measurement.

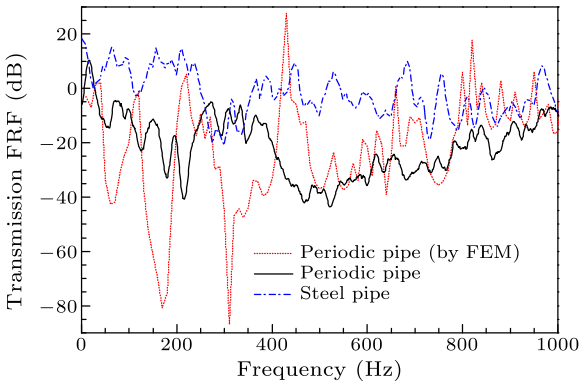

Fig. 8. The flexural transmission FRF in the y-direction.

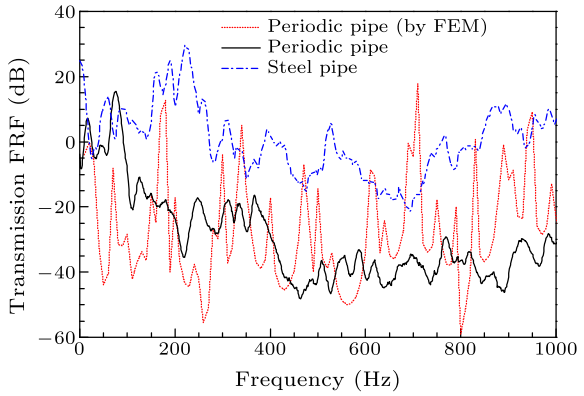

Fig. 9. The flexural transmission FRF in the z-direction.

Figures 8 and 9 show the flexural vibration transfer curves along y and z directions when the oil pressure is 1 MPa. The dotted and the solid lines stand for the experimental results of FRFs of the steel pipe and the periodic pipe respectively, and the dashed line (by FEM) represents the numerical results of FRFs of the

periodic pipe.

Figure 8 shows that there are two obvious BGs in the frequency ranges of 32–271 Hz and 370–963 Hz in the y-direction, and the average attenuation is up to 25 dB for the periodic pipe, while the vibration attenuation is not observed for the steel pipe. Figure 9 shows that there exist obvious BGs in the frequency range of 90–1000 Hz in the z-direction and the average attenuation is up to 30 dB for the periodic pipe, while the vibration attenuation only occurs in a narrow frequency ranges of 560–720 Hz for the steel pipe. This is due to the fact that the straight pipe can improve the flowing properties of the oil and the fluid noise is suppressed. In a word, the periodic pipe can suppress vibration effectively and has better vibration suppression performance than the steel pipe.

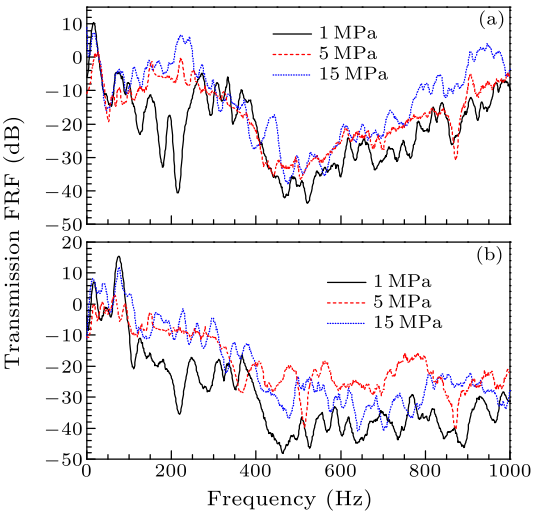

Fig. 10. The FRFs under various pressures: (a) the y-direction, and (b) the z-direction.

For the periodic pipe, the experimental results change more gently than numerical ones, and the vibration attenuation is smaller in experiments than numerical calculations in some frequency ranges. There are three reasons behind the differences between experimental and numerical results. Firstly, the geometric parameters of experiments and numeric calculations are different, e.g., the pipe joints are ignored in the numerical model. Secondly, the boundary conditions of experiments differ from those in numerical calculations, e.g., the inlet and outlet ends of the test sample are connected to a hydraulic oil tank, which leads to the propagation of a part of vibration through the outlet end and further interferes with the experimental results. Thirdly, the physical parameters of the periodic pipe are different, e.g., the hose is assumed to be linear in numeric calculation while it is not in reality. Despite all this, the experimental results show a good agreement with the numerical calculations. Consequently, the validity of the improved method proposed in this work and the vibration suppression performance of the periodic pipe are verified.

As shown in Fig. 10, the solid, the dashed and the dotted lines are the FRFs of the periodic pipe under the oil pressures of 1 MPa, 5 MPa and 15 MPa, respectively. Compared the FRFs of the periodic pipe under the oil pressures of 1 MPa with 15 MPa, it can be found that the amplitude of vibration attenuation decreases by about 10 dB in the whole frequency range along with the elevation of oil pressure, especially the vibration attenuation drops sharply around 200 Hz. This is due to the fact that the vibration of the hydraulic oil tank is intensified along with the elevation of oil pressure and the vibration resonates with the support around 200 Hz. As the oil pressures goes up, the vibration attenuation of the periodic pipe drops, while the oil pressure has no significant effects on the frequency ranges of BGs, which is consistent with the numerical analysis.

In summary, a periodic pipe structure has been designed to suppress the vibration transfer of a hydraulic system. The force on the inner pipe wall applied by the oil pressure is taken into consideration to study the vibration transfer properties of the 3D high-pressure periodic pipe. It is found that the hydraulic fluid pressure exerts certain influences on the vibration transfer properties of the periodic pipe. As the hydraulic fluid pressure rises, BGs in low frequency ranges move towards high frequency ranges, while the amplitude of vibration attenuation is apparently decreased, and in particular the vibration attenuation declines sharply in low frequency ranges. A comparison is made between the experimental results and numerical calculations. The experimental results show a good agreement with the numerical calculations and thus the validity of the improved method proposed in this study is verified, which can exactly calculate the vibration transfer properties of a high-pressure periodic pipe.

References

[1] Kushwaha M S et al 1993 *Phys. Rev. Lett.* **71** 2022
[2] Kushwaha M S et al 1994 *Phys. Rev.* B **49** 2313
[3] Koo G H and Park Y S 1996 *Int. J. Press. Ves. Pip.* **67** 249
[4] Koo G H and Park Y S 1998 *J. Sound Vib.* **210** 53
[5] Xu Y L et al 2013 *Chin. Phys. Lett.* **30** 044301
[6] Liu Z et al 2000 *Science* **289** 1734
[7] Liu Z et al 2002 *Phys. Rev.* B **65** 165116
[8] Wu F et al 2003 *Eur. Phys. J.* B **34** 265
[9] Wen J H et al 2005 *J. Appl. Phys.* **97** 114907
[10] Zhao H G et al 2007 *Chin. Phys. Lett.* **24** 1305
[11] Ma F Y et al 2015 *Eur. Phys. J. Appl. Phys.* **71** 30504
[12] Sorokin S V and Ershova O A 2006 *J. Sound Vib.* **291** 81
[13] Yu D L et al 2008 *J. Sound Vib.* **318** 193
[14] Shen H J et al 2009 *J. Sound Vib.* **328** 57
[15] Wen J H et al 2010 *Chin. Phys. Lett.* **27** 114301
[16] Lee S Y and Mote Jr C D 1997 *J. Sound Vib.* **204** 735
[17] Kang M G 2000 *J. Sound Vib.* **238** 179
[18] Kittle C 1986 *Introduction to Solid State Physics* (New York: Wiley)
[19] Cong H B et al 2010 *Machine Tool Hydraulics* **38** 81 (in Chinese)
[20] Zhang S W et al 2013 *J. Appl. Phys.* **113** 163511
[21] Wen J H et al 2011 *Chin. Phys. Lett.* **28** 094301
[22] Beranek L L and Ver I L 1992 *Noise and Vibration Control Engineering* (New York: Wiley)
[23] Fahy F J 1985 *Sound and Structural Vibration: Radiation, Transmission and Response* (London: Academic Press)