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Experimental study on barrel viscous dampers and pipe hoops in pipeline vibration reduction①

Zhang Bingkang (张炳康)②, He Lidong, Chen Guo, Zhang Zhenkun, Lv Chenglong
(Ministry of Education Engineering Research Center of Chemical Safety, Beijing University of Chemical Technology, Beijing 100029, P. R. China)

Abstract

The centrifugal air compressor outlet pipeline vibration was not decreased after barrel viscous dampers were installed in a petrochemical plant in Tianjin. A pipeline-damper experiment apparatus was built for studying the influence factors of the barrel viscous damper and pipe hoop in pipeline vibration reduction. The performance of the damper under different frequency and amplitude was researched respectively, the results showed that damping effect dependsed mainly on frequency and was not related to amplitude. Damper will fail when its vibration frequency exceeds its limit working frequency which was 40Hz in test. The mechanical properties and energy dissipation were analyzed by using the Maxwell model, which explains experimental results well. According to damping effect and calculation of stiffness with ANSYS in different hoop width, hoop stiffness should match pipe stiffness and keep uniform along transfer path. Damping effect will get worse when local stiffness is too small or too large. Finally, the outlet pipeline vibration was decreased by 70% after using appropriate pipe hoop width and replacing the original damping liquid.

Key words: pipeline vibration reduction, barrel viscous damper, frequency, amplitude, pipe hoop stiffness

0 Introduction

Pipeline is an indispensable element used to deliver and transport liquid and gas[1]. Yet it would vibrate inevitably in actual operation, which does great harm to personnel and production[2]. Viscous damper is one of the most effective technical means for inhibiting vibration, which is from the areas of aerospace and military industry to the field of engineering structure. Barrel viscous dampers have been used in pipeline vibration reduction[3]. In Ref. [4], feed water pipeline vibration was decreased by 2/3 after installation of dampers in a nuclear power station.

This paper presents severe vibration of a large centrifugal air compressor outlet pipeline in Tianjin. Vibration was not decreased after a barrel viscous damper had been installed firstly. So it is necessary to study the performance of damper and pipe hoop in pipeline vibration reduction.

The mechanical properties and energy dissipation of a majority of dampers have been analyzed deeply. Refs [5] and [6] researched damping materials and the performance of viscous damper that was applied in engineering structure. And a calculation model of viscous damper was put forward for the first time. Ref. [7] depicted that damping coefficient was nonlinear and inversely proportional to the amplitude and frequency of motion in a squeeze film damper. Ref. [8] presented three calculation formulas of damping force that were applied in heat exchanger. Ref. [9] discussed the characteristics of different kinds of viscous fluid, and damping force is established based on power law fluid. Ref. [10] manufactured a fluid viscous damper with double guide bars. And the energy dissipation mechanism and damping force were analyzed in detail. However, further research is needed for the characteristics and energy dissipation of barrel viscous damper.

Pipe hoop, as the only connection component between pipeline and damper, its stiffness has an effect on pipeline vibration reduction. Ref. [11] determined pipe hoop stiffness precisely for the analysis of aero-engine pipeline system by finite element calculation and

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② To whom correspondence should be addressed. E-mail: zhanbgingkang@126.com
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experiment method. This paper researched the effects of hoop stiffness in pipeline vibration reduction concretely.

A pipeline-damper experiment apparatus was established for studying the influence factors of barrel viscous damper and pipe hoop in pipeline vibration reduction. The mechanical properties of damper were analyzed by using Maxwell model. Then experiment of different hoop width was carried out for researching the relationship between damping effect and hoop stiffness. Finally, the experimental conclusions were applied to the secondary damping transformation project of outlet pipeline in Tianjin.

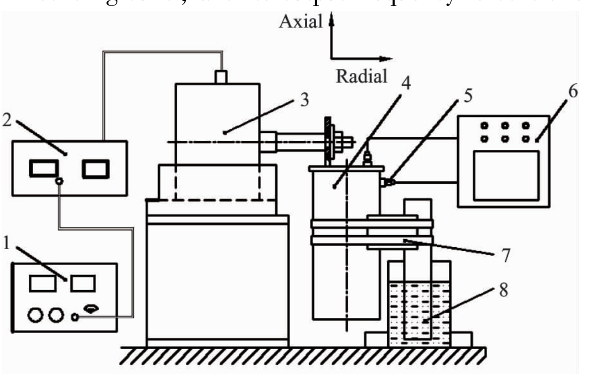
1 Experimental design

1.1 Experimental principles

A barrel viscous damper is connected with a pipeline through a pipe hoop at one end, and the other end is connected with fixed foundation. There is a mutual movement between the damper piston rod and the large number of internal viscous damping liquid when the pipeline vibrates. The generated damping force is opposite to the piston rod movement all the time. The mechanical energy of vibration passes quickly to high viscous damping liquid, which is released by heat[12]. The above discussion is the principle of damper in pipeline vibration reduction.

1.2 Experimental apparatus

According to the principles of damper in pipeline vibration reduction, the entire experiment apparatus is shown in Fig. 1, which consists of five parts: excitation device, pipeline, pipe hoop, viscous damper and vibration measurement instrument. The excitation device includes permanent magnet vibrator of JZ-5 type which is fixed on ground, and its output frequency is controlled



1-signal generator; 2-power amplifier; 3-vibrator; 4-pipeline; 5-acceleration sensor; 6 -vibration measurement instrument; 7-pipe hoop; 8-barrel viscous damper

Fig. 1 Pipeline-damper experiment apparatus

by an XD1022 signal generator which can produce 1 ~ 10000Hz continuous adjustable sine wave signal. The signal is enlarged through GF-500 power amplifier which provides enough power drive for vibrator. The outer diameter of pipeline is 80mm. The damper is composed of an inner piston rod, viscous damping liquid and an outer barrel. The vibration measurement instrument adopts Smart Balancer vibration tester which can measure dual channels at the same time. Two acceleration sensors are installed for exploring the difference of radial and axial vibration, and the maximum range of sensor is 6mm.

2 Experiment of the barrel viscous damper

2.1 Experimental results of frequency

The signal generator is adjusted for sampling every 10Hz interval between 10Hz and 100Hz while the power amplifier stays the same. The radial and axial displacements of pipeline vibration in different frequency are measured respectively. The vibration time-domain waveforms are shown in Fig. 2 and Fig. 3. (only 10Hz in radial direction)

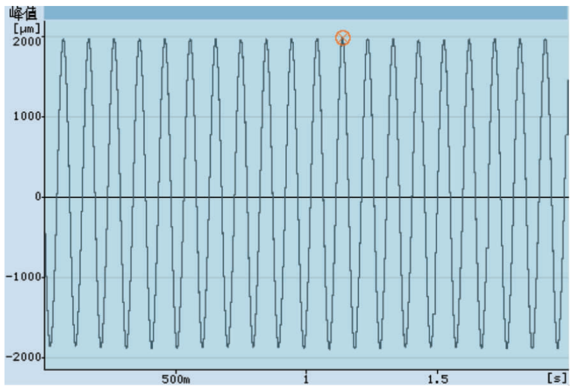


Fig. 2 Radial vibration time-domain waveform without damper

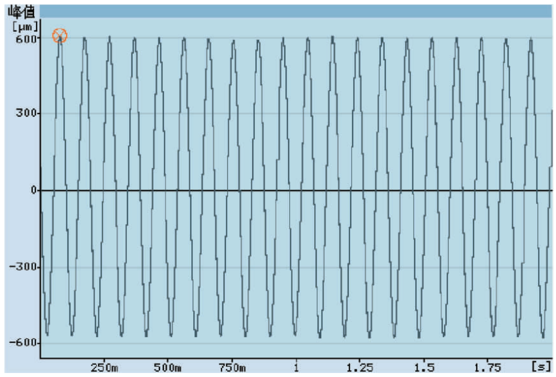


Fig. 3 Radial vibration time-domain waveform with damper

All measures are inputted into computer (Intel (R) Core (TM) i7, 8GB RAM), and the operating system is Window 7 Professional (x64). The changing curves of peak-peak (P-P) values of radial and axial vibration displacement of are shown in Fig. 4 and Fig. 5 by Origin Pro 8.0 Data Analysis & Scientific Graphing software.

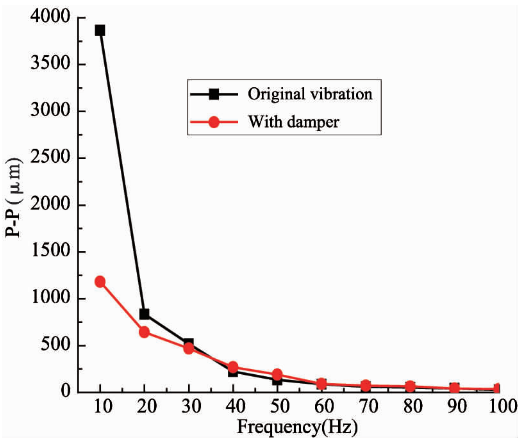


Fig. 4 P-P values of radial vibration

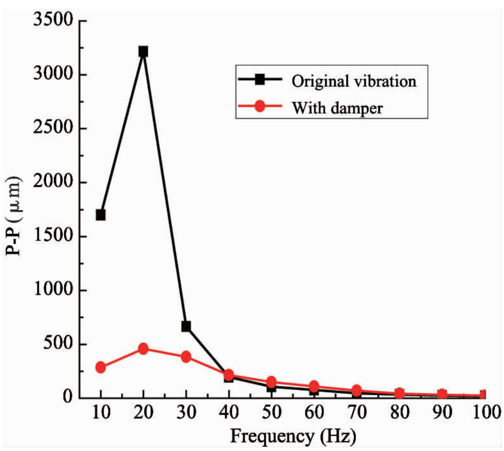


Fig. 5 P-P values of axial vibration

Fig. 4 shows that radial vibration reduction is significant when the frequency is less than 40Hz. And vibration is reduced by 69.5% at 10Hz, 23.0% at 20Hz and 9.1% at 30Hz. However, vibrations are increased by 21.1% at 40Hz and 19.2% at 100Hz.

Note from the above calculation, damping effect becomes worse gradually as frequency increases, and there is not damping effect when the frequency is high enough. Barrel viscous damper has a limit working frequency on radial vibration. Damper is no longer suitable when the frequency exceeds 40Hz in test.

Fig. 5 depicts the results of axial vibration that is similar to radial direction. The vibration decrement is calculated respectively as follows: 83.1% at 10Hz, 85.7% at 20Hz and 42.2% at 30Hz. The turning point is occurred at 40Hz, and vibration increment is about 8.6% at 40Hz and 17.6% at 100Hz.

Axial vibration reduction is also very obvious at low frequency. The damper cannot work anymore when frequency increases to 40Hz, which shows that barrel viscous damper also has a limit working frequency on axial direction.

It can be also found that damping effect of axial is much better than radial by comparing experimental results under the condition of less than 30Hz, which shows that damping effect has a difference in each direction for barrel viscous damper. It is significant to installing direction of damper for engineering project.

2.2 Experimental results of amplitude

The above results may be influenced by two factors for the output amplitude of vibrator decreased as frequency increased. So the output amplitude is increased by turning up voltage. Especially when frequency is above 40Hz, amplitude is adjusted to the values of lower frequency, such as 30Hz. The P-P values of vibration displacement of radial and axial in high amplitude are shown in Fig. 6 and Fig. 7.

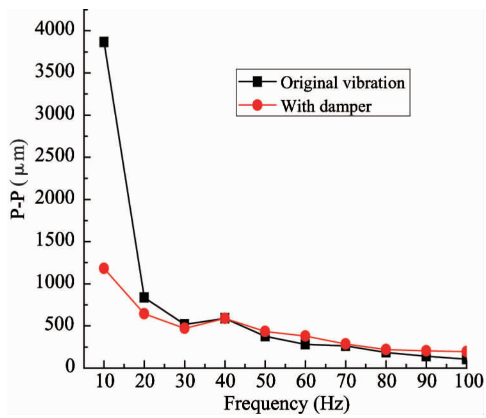


Fig. 6 P-P values of radial vibration in high amplitude

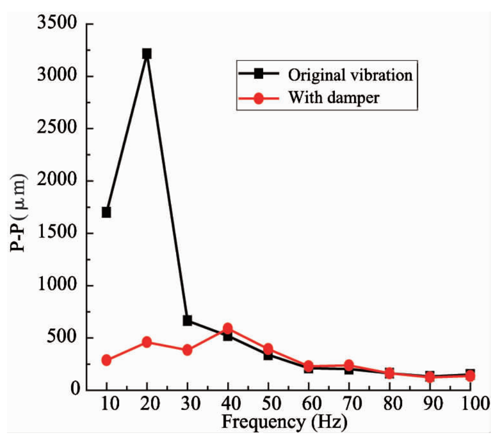


Fig. 7 P-P values of axial vibration in high amplitude

It can be seen from the experimental data that damping effect remains unchanged in high amplitude at the same frequency. Beginning from 40Hz, the radial and axial vibration still has no damping effect although turning up the amplitude. For example, the radial vibration stays the same and the axial vibration increases by 13.2% at 40Hz.

Fig. 6 and Fig. 7 show that barrel viscous damper only works effectively below the limit working frequency. Damping effect of barrel viscous damper mainly depends on frequency and is not related to amplitude.

2.3 Analysis of experimental results

It is very necessary to analyze the mechanical properties and energy dissipation of barrel viscous damper to explain the experimental results.

The Maxwell model can be used to analyze damper for its mechanical properties are closely related to frequency. It depicts a continuous theory of "damp- stiffness", which reflects the performance of damper accurately[13]. The Maxwell model is constituted by a damping unit and a spring unit in series connection. Assuming displacements of the damping unit and the spring unit are $u_1(t)$ and $u_2(t)$ respectively, then the following relationships can be established.

$$u_1(t) + u_2(t) = u(t) \tag{1}$$

$$C_0\dot{u}_1(t) = Ku_2(t) = F(t) \tag{2}$$

From Eqs(1) and (2),

$$F(t) + \lambda\dot{F}(t) = C_0\dot{u}(t) \tag{3}$$

(where $F(t)$ represents damping force, C_0 represents linear damping constant of zero frequency , K represents stiffness coefficients of infinite frequency, and λ represents time coefficient, $\lambda = C_0/K$.)

By using the Euler formula and Fourier transform, another expression under frequency domain can be obtained.

$$u(\omega) = u_1(\omega) + u_2(\omega) \tag{4}$$

$$K^*(\omega)u(\omega) = iC_0\omega u_1(\omega) = Ku_2(\omega) \tag{5}$$

From Eqs(4) and (5), the following is obtained,

$$K^*(\omega) = K\frac{iC_0\omega}{K + iC_0\omega} = \frac{C_0\lambda\omega^2}{1+\lambda^2\omega^2} + i\frac{C_0\omega}{1+\lambda^2\omega^2} \tag{6}$$

Substitution of $\lambda = C_0/K$ into Eq.(6) yields,

Storage stiffness:

$$K_1(\omega) = \frac{C_0\lambda\omega^2}{1+\lambda^2\omega^2} = \frac{K\lambda^2\omega^2}{1+\lambda^2\omega^2} \tag{7}$$

Dissipation stiffness:

$$K_2(\omega) = \frac{C_0\omega}{1+\lambda^2\omega^2} \tag{8}$$

Hence, the damping coefficient is

$$C(\omega) = \frac{K_2(\omega)}{\omega} = \frac{C_0}{1+\lambda^2\omega^2} \tag{9}$$

From the above derivation, it can be seen that damping coefficient decreases as frequency increases. Then damping force can be expressed as

$$F = C(\omega)\cdot V^\alpha = \frac{C_0}{1+\lambda^2\omega^2}\cdot V^\alpha \tag{10}$$

where α represents speed index which is related with internal structure of damper and damping fluid. It is called nonlinear viscous damper when $\alpha < 1$, linear viscous damper when $\alpha = 1$ and ultra linear viscous damper when $\alpha > 1$. The speed index of barrel viscous damper is usually $0.3 \sim 1.0$[10].

Damping characteristics can be also analyzed by calculating the internal energy dissipation of damper. The energy dissipation per cycle may be expressed as

$$W = \oint F\mathrm{d}x = \oint C(\omega)\cdot V^\alpha \mathrm{d}x \tag{11}$$

Assuming displacement

$$x = X\sin(\omega t - \varphi) \tag{12}$$

Velocity can be obtained from the derivative of Eq.(12), and substituted into Eq.(11):

$$W = C\omega^{\alpha+1}X^{\alpha+1}\int_0^{\frac{2\pi}{\omega}}\cos^{\alpha+1}(\omega t - \phi)\mathrm{d}t \tag{13}$$

The integration of Eq.(13) can be expressed as the function of α, which is defined as $f(\alpha)$, so

$$W = \frac{f(\alpha)C_0\omega^\alpha X^{\alpha+1}}{1+\lambda^2\omega^2}\ (\omega = 2\pi f,\ \alpha = 0.3 \sim 1.0) \tag{14}$$

The Eqs(10) and (14) show that F and W are only functions of f when α is a certain value. Damping force and energy dissipation is not related to amplitude. Both of them decrease as frequency increases. Damping effect is good enough at low frequency while getting worse at high frequency. That explains the experimental results well. Barrel viscous damper becomes invalid when frequency increases to a certain extent. That is the limit working frequency, which mainly depends on the internal properties of viscous damping liquid and damper structure.

3 Experiment of the pipe hoop

3.1 Experimental results of different hoop width

Vibration transmissibility of pipe hoop decides the effects of pipeline vibration reduction. Hoop stiffness has an effect on vibration transmissibility, and pipe hoop width is one of the factors that affect stiffness. Pipe hoop width is chosen for 10mm, 20mm and 40mm respectively, which meant 12.5%, 25% and 50% of the outer diameter of pipeline. By measuring the radial and axial vibration every 10Hz interval in different hoop width, the P-P values changing curves of vibration displacement are shown in Fig. 8 and Fig. 9.

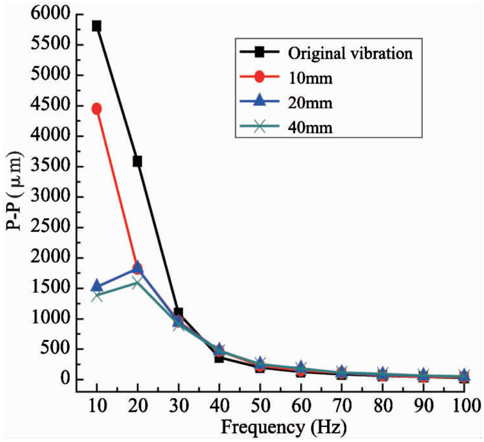


Fig. 8 P-P values of radial vibration in different hoop width

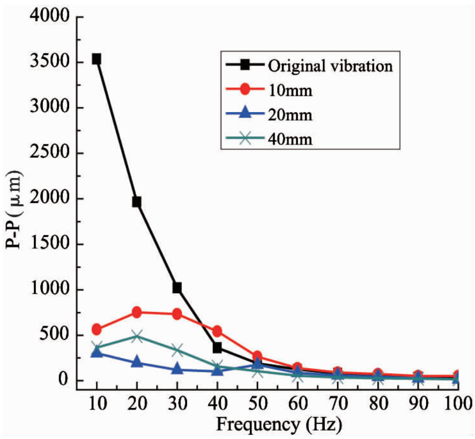


Fig. 9 P-P values of axial vibration in different hoop width

The vibration decrement of radial vibration in different hoop width can be calculated from Fig. 8. It is 23.4% at 10mm, 73.8% at 20mm and 76.1% at 40mm when frequency is 10Hz. And when frequency rises to 30Hz, it is 12.2% at 10mm, 13.8% at 20mm and 16.5% at 40mm.

Note from the results, damping effect is greatly improved after the pipe hoop width is inereased from 10mm to 20mm, while it is almost unchanged from 20mm to 40mm. There is little difference of damping effect when frequency is over 40Hz, which indicates that widening the pipe hoop can't solve the problem of high frequency vibration.

Fig. 9 depicts the results of axial vibration that is unlike radial direction. Vibrations are dropped by 84.1% at 10mm, 91.4% at 20mm and 89.8% at 40mm than original vibration when frequency is 10Hz, and 28.1% at 10mm, 88.3% at 20mm and 67.1% at 40mm when frequency is 30Hz.

The calculation data shows that axial vibration reduction is not proportional to pipe hoop width when frequency is blow 40Hz. It would be best when hoop width is 25% of pipe diameter, and it may be worse for continuing widen. Frequency range of vibration reduction is not related to pipe hoop width.

3.2 Analysis of experimental results

The above results show that pipe hoop width affects pipeline vibration reduction significantly. Calculating the corresponding stiffness in different hoop width is demanded for selecting an optimum hoop for specific pipeline. The finite element model of 10mm, 20mm and 40mm pipe hoops were established respectively with ANSYS 14.0 finite element software on the computer. The radial and axial deformation in different width can be obtained by structural analysis. According to stiffness definition: $K = F/\delta$ (where F represents concentrated force, δ represents linear displacement), the corresponding stiffness of pipe hoop can be calculated.

External loads can't be applied on pipe hoop directly. Otherwise the displacement of loading point is far more greater than displacement of the adjacent nodes. It doesn't conform to actual situation[11]. Therefore, pipeline was simulated as a large rigid rod, and 1000N loads were applied on the radial and axial pipeline respectively. Pipe hoop which is connected with damper was set as fixed constraint. The radial and axial deformation of pipe hoop was shown in Fig. 10 and Fig. 11. (Only 20mm pipe hoop)

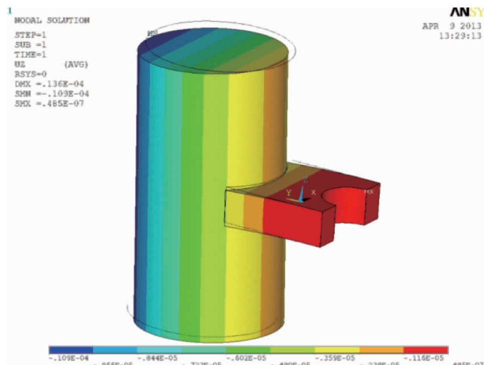


Fig. 10 The radial deformation of 20mm pipe hoop

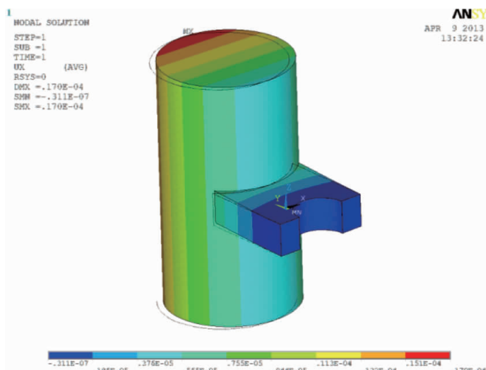


Fig. 11 The axial deformation of 20mm pipe hoop

According to the radial and axial deformation, radial stiffness (Kr) and axial stiffness (Ka) of the pipe hoop in different width was listed in Table 1.

Pipeline ring stiffness is adopted empirical formula: $K = E_P \times I_P / 8R_0^3$ (where E_P represents elasticity modulus, I_P represents cross sectional moment of inertia and R_0 represents outer radius of pipeline). According to the structure parameter of pipeline in test, pipeline stiffness (K) is also listed in Table 1.

Table 1 Stiffness of pipe hoop and pipeline

Width(mm)	Kr(N/M)	Ka(N/M)	K(N/M)
10	2.21×10^8	3.35×10^7	3.41×10^8
20	5.29×10^8	2.79×10^8	3.41×10^8
40	1.25×10^9	1.12×10^9	3.41×10^8

Note from Table 1, it can be seen that:

Both the radial and axial stiffness of 10mm pipe hoop is less than the pipeline stiffness, vibration can't be completely transmitted to damper through pipe hoop, so damping effect is not good enough.

The radial and axial stiffness of 20mm pipe hoop is 1.55 times and 0.82 times of pipeline stiffness respectively, which means a uniform distribution of stiffness along transfer path. Vibration transmissibility is good enough. That explains 20mm pipe hoop achieves the best damping effect.

The radial and axial stiffness of 40mm pipe hoop is 3.67 times and 3.25 times of pipeline stiffness respectively. Both radial and axial stiffness are far more bigger than pipeline stiffness. Local stiffness is too big, which reduces vibration transmissibility remarkably. Damping effect can't be better for continuing widing.

Pipeline vibration should transmit as good as possible along the transfer path to barrel viscous damper. Pipe hoop must have enough stiffness and keep uniform, never too large or too small of local stiffness. That requires pipe hoop stiffness must match with pipeline stiffness. In other words, there is an optimum pipe hoop width, so that pipeline vibration can transfer to viscous damping fluid, which can't be weakened or shared by pipe hoop.

4 Engineering application

4.1 Before damping transformation

The Siemens centrifugal air compressor was used in a petrochemical plant in Tianjin. Its third class outlet pipeline is of cone-shaped and 760mm of the maximum nominal diameter. The piping system slopes 27° and has no any support along piping direction. The M20 bolt of flange was cracked by severe vibration, as shown in Fig. 12.

Fig. 12 Cracked M20 bolt

The displacement of test point A was up to 5mm, as shown in Fig. 13. The vibration exceeds the pipeline vibration standard of America greatly, which regulates that vibration amplitude must be less than 0.2mm[14]. The reinforcing hoop is a protection for flange. But it also failed to resolve the fundamental problems of vibration. Barrel viscous damper can dissipate energy of vibration. But it is difficult to install for its narrow space and oblique direction.

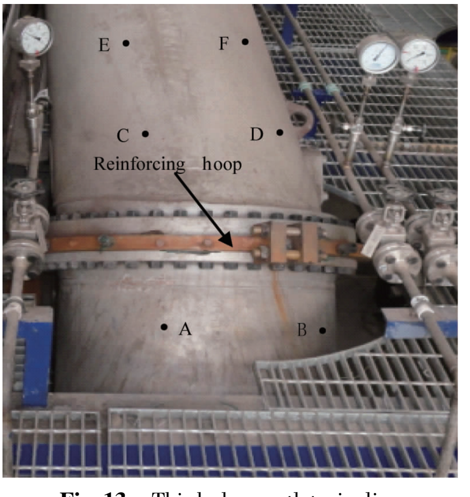


Fig. 13 Third class outlet pipeline

4.2 After damping transformation

The vibration reduction was not obvious after a barrel viscous damper was installed for the first transformation. So it is necessary to analyze the characteristic of pipeline vibration and the pipe hoop. The results show that: (1) The frequency range of pipeline vibration is 2 ~ 200Hz, where the peak velocity of dominant frequency can reach 276mm/s at 2Hz, while the second frequency is 98.8mm/s at 200Hz. The limiting working frequency of barrel viscous damper is only 40Hz in experiment, so it is unable to work effectively for this situation. (2) The pipe hoop width is only 80mm for the first time, and its stiffness is much smaller than that of pipeline. So it is unable to meet the requirements of vibration transmissibility.

Experimental conclusion was applied to the secondary damping transformation. Low viscosity damping liquid was adopted and the pipe hoop width was increased to 160mm. The mounting position of damper was changed to the area of maximum pipeline vibration. The peak velocity of 6 test points (Fig. 13) was shown in Table 2 after transformation.

Table 2 Peak velocity of 6 test points

Test point	Original (mm/s)	Transformation (mm/s)	Decrement (%)
A	249	78.2	68.6
B	747	169	77.3
C	652	198	69.6
D	756	123	83.7
E	38.8	12.1	68.8
F	95.9	31.9	66.7

It can be seen that evident damping effect has been achieved for the secondary damping transformation. The average vibration decrement can reach about 70%, which fully proves the correctness and effectiveness of experiment.

5 Conclusions

This paper presents the experimental study of barrel viscous damper and pipe hoop. And the outlet pipeline vibration has been reduced successfully in Tianjin. It turned out that:

(1) Vibration reduction effect of barrel viscous damper depends on frequency and not related to amplitude. The limit working frequency of damper is about 40Hz in test. Vibration reduction effect is remarkable when frequency is less than 40Hz, while it is invalid when frequency is over 40Hz.

(2) Damping coefficient, damping force and energy dissipation was analyzed by using Maxwell model, which shows that both of them decrease as frequency increases.

(3) Axial damping effect is better than radial at the same frequency for barrel viscous damper.

(4) Pipeline vibration should transmit as well as possible along transfer path to barrel viscous damper, which requires pipe hoop stiffness should be equal or close to pipeline stiffness. Experimental results show that pipe hoop of 25% pipe diameter is optimal to achieve the best damping effect in general.

(5) The outlet pipeline vibration has been decreased by 70% in Tianjin after secondary damping transformation.

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Zhang Bingkang, born in 1989. He received his B.S. degrees in Mechanical Engineering and Automation Department of Beijing University of Chemical Technology in 2011. He received his M.S degree in Chemical Process Equipment Department of the same school in 2014. His research interests are vibration control of pipeline and rotating machinery.