Optimized Design of Vibration Reduction for Aviation Hydraulic Pipeline System

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Abstract: Aiming at the crack failure problem caused by excessive vibration at the joint of a certain engine pipeline system, two improved schemes of structural optimization and added clamp are proposed, and finite element modeling, inherent characteristics analysis and pipeline vibration response calculation under foundation excitation are also carried out. The vibration response calculation shows that the effect of changing pipe shape on changing inherent characteristics of pipeline and reducing vibration stress of pipeline is not obvious. After adding the clamp, the first and second-order modal vibration stresses are the largest, mainly distributed in the elastic support end and the clamp, and the maximum value appears at the clamp, and the vibration stress at one end of the rigid support is relatively small; the first-order natural frequency of each pipeline deviates, and the first-order resonance stress value of the rigid joint decreases significantly. The first-order resonance stress value of the elastic support increases, but because it is far away from the excitation frequency, and it does not have much influence on the vibration fatigue of the pipeline structure. The simulation results can provide a reference for the optimal design of the engine pipeline system.

Keywords: Pipeline system, Vibration, Pipe shape, Clamp, Optimized design

1 Introduction

All the accessories in the aircraft hydraulic system are connected by hydraulic lines and pipe joints. For example, a certain type of aircraft has a total of 1083 hydraulic lines with the total length of 882 meters, and almost through all parts of the body. Any part of the pipeline's and the connector's damage may cause system's huge failure [1]. The vibration problems of aero-engine pipeline systems mainly include excessive vibration and vibration fatigue. The static pressure bearing capacity of the pipeline is very abundant. The main cause of pipeline damage is that the pipeline is subjected to large repeated loads, that is, vibration which causes fatigue damage [2]. With the development of modern aircraft hydraulic systems in the direction of high pressure and high power, the influence of pipeline vibration has become increasingly

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prominent [3, 4]. Therefore, the theory and method of vibration analysis and dynamic design for aeroengine pipeline systems are becoming more and more important. The analysis, design and optimization of pipeline support structure and pipeline system have important academic value and practical significance [5].

From the existing research work at home and abroad, the research on the dynamic design method based on the vibration of pipeline system is gradually deepening. A H M Kwong and K A Edge [6] used the transfer matrix method to predict the dynamic response of the hydraulic pipeline and optimize the position of the pipe clamp. In view of the vibration problem of the air cylinder pipeline system supported by the flexible clamp, Li Zhanying et al.[7] studied the stiffness characteristics of the flexible clamp and its influence on the response characteristics of the pipeline system. Paidoussis et al.[8] studied the nonlinear dynamic behavior of cantilever hydraulic straight pipe with nonlinear spring support, and gave the influence of spring position and stiffness parameters on the instability mode of the pipeline. Ling Fuyong [9] used the finite element method and CFD method to analyze the influence of the bend angle on the vibration characteristics of the pipeline under the unsteady flow of a high-cycle pressure pipeline of a certain type of aircraft; The force analysis of straight pipe and 90° elbow is carried out by Wang Jianping[10] and it is found that the transverse load per unit length of pipe is proportional to the cross section area of the pipe and the liquid pressure, and inversely proportional to the bending radius. For the aerodynamic bending pipeline, Quan L et al.[11] established a fluid-solid coupling 14-equation model and transformed it into the frequency domain by Laplace transform to study the influence of bending parameters on the frequency response of a single pipeline and the influence of bending parameters on the natural frequency of the double pipe; Li Yanhua et al.[12] established the fourteen equation model of the elbow, and obtained the influence of the elbow angle and the radius of the pipe on the coupled vibration.

However, it is still difficult to directly and effectively guide the fault tracing of the engine pipeline system. Moreover, in terms of dynamics modeling, the traditional dynamic models of single straight pipes and elbows have been “stretched”, and it is difficult to effectively guide the failure mechanism analysis of engine pipelines and guide the design of pipeline systems. The special service environment of the engine and the complicated layout structure of the pipeline system itself bring new scientific challenges to the establishment of the dynamic design method of the pipeline system. Therefore, in-depth study of the dynamic design method of pipeline system is of great significance to the engine design of current and future aircraft.

2 Pipeline structure description and finite element modeling

As shown in Fig. 1, the pipe has a diameter of 6 mm and a wall thickness of 1 mm. One end of the pipe is threaded on the tank, and the other end is connected to the other pipe through the joint. The ball head is welded to the pipe. The pipe material is austenitic stainless steel 1Cr18Ni9Ti, and the pipe joint material is martensitic stainless steel 1Cr17Ni2.

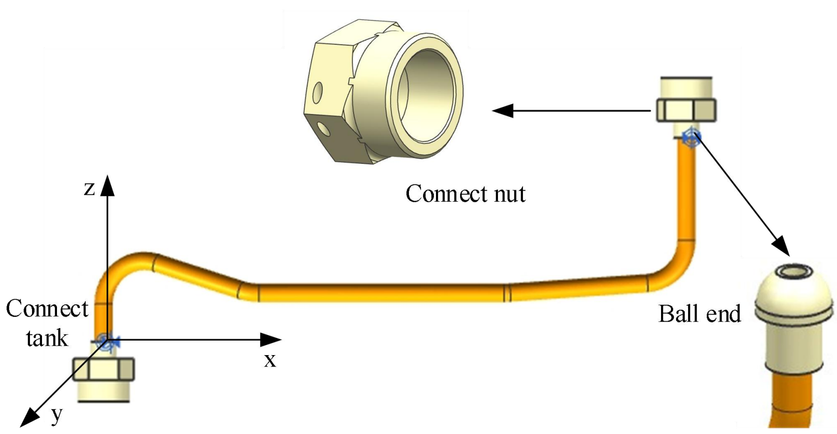


Fig. 1. A certain type of pipeline

For the vibration problem of a certain type of pipe, an improved scheme such as structural optimization and adding clamps is proposed, as shown in Fig. 2. For the structural optimization of the pipe type of a certain type of pipe (1# scheme in Fig. 2), four new pipe types (2#, 3#, 4# and 5# in Fig. 2) can be obtained, and the clamp mounted position is shown in the Fig. 2.

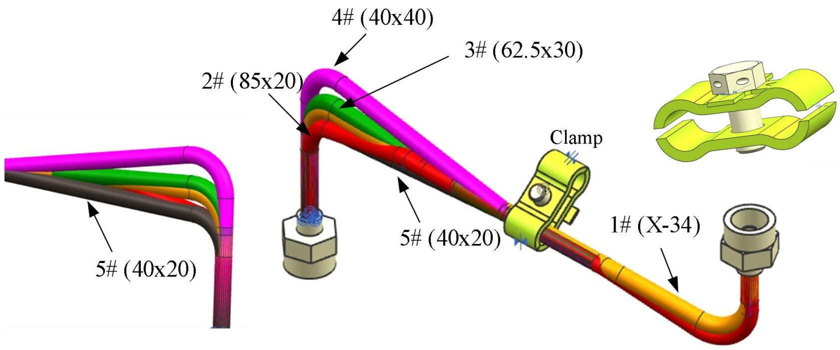


Fig.2. Improved plans for existed pipeline structure

The pipeline entity model established in UG is saved in the PARASOLID format and imported into the finite element analysis software ANSYS. The high-order three-dimensional solid element solid186 is used for meshing. Because there are many complicated structures such as corners in the model, the intelligent mesh is used for free division. The final mesh has 52,053 nodes and 27,744 units. The finite element model of a model pipe is shown in Fig.3. The method of restraining the full degree of freedom of the inner ring node of the connecting nut is used to simulate the actual installation mode of the pipeline, which means the one end of the left joint is fixed.

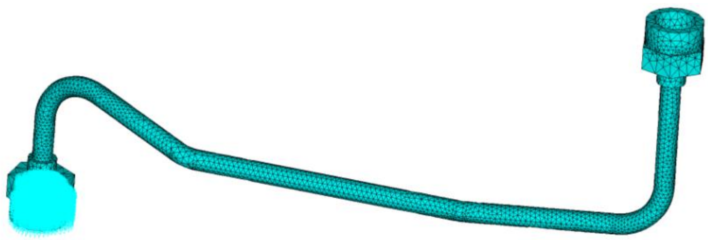


Fig. 3. Finite element model of a certain pipe

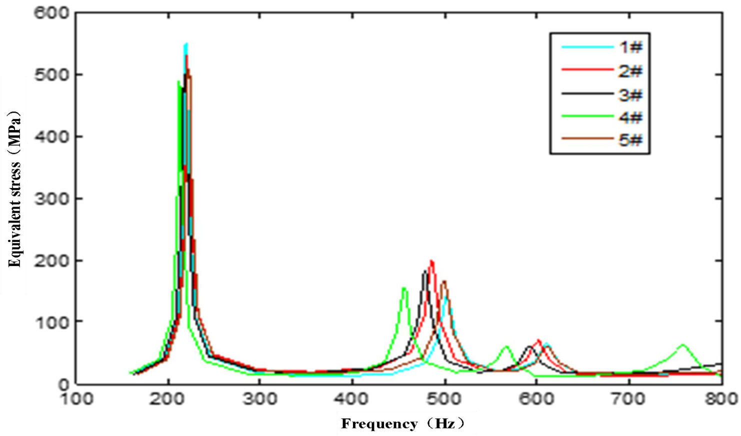
3 Influence of pipe shape on inherent characteristics and vibration stress of pipeline

Using the established finite element model and the confirmed boundary conditions and load conditions, the vibration response of different schemes under the basic vibration is calculated, and the vibration stress distribution of the pipeline is obtained. First, the finite element analysis was carried out on the other four types of pipes, and the natural frequencies of different pipe types were first calculated and listed in Table 1.

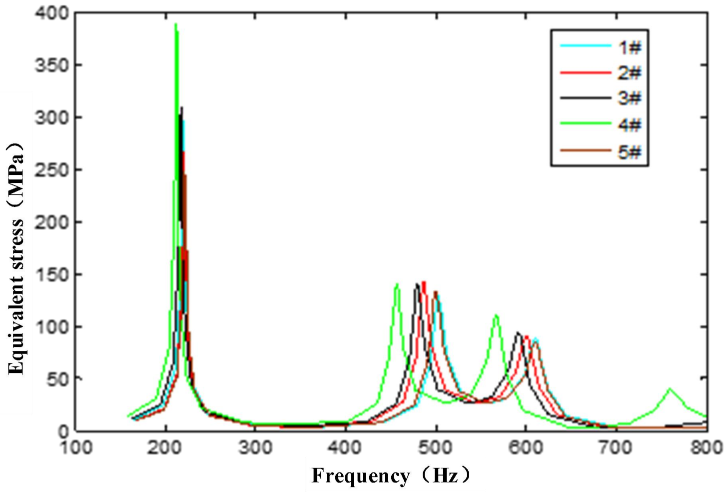
Table 1. The first 5 natural frequencies corresponding to different pipe types

Order	**Natural frequency /Hz**				
	1#	2#	3#	4#	5#
1	219.62	221.12	217.18	212.20	221.83
2	222.01	223.63	219.90	215.52	224.21
3	502.30	486.55	479.22	456.24	499.54
4	610.95	601.20	592.16	567.31	610.85
5	940.20	969.70	872.58	759.13	980.44

It can be seen from Table 1 that the influence of the slight change of the pipe bend angle on the inherent characteristics of the pipe is not obvious compared to the original scheme 1# pipe. The harmonic response analysis of different pipe types is carried out by modal superposition method. The frequency-stress curves of point 1 (rigid connection end) and point 2 (ballistic end) are shown in Fig. 4.



(a) Frequency-stress curve at point 1



(b) Frequency-stress curve at point 2

Fig.4. Vibration stress frequency curves obtained from different pipe types

From the peak frequency of the harmonic response curve and the corresponding stress value, the peak value of the resonance stress of each pipe type is not much different, and it is very close. Further, the stress values of points 1 and 2 at 650 Hz and 194 Hz excitation were extracted as shown in Table 2.

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Table 2. Vibration stress values corresponding to different pipe types

Pipe shape	Vibration stress /Mpa		
	Point1(650Hz)	Point2(650Hz)	Point2(194Hz)
1#(Experimental pipe)	17.35	12.44	22
2#(85x20)	16.28	11.15	19.57
3#(62.5x30)	17.88	8.71	23.95
4#(40x40)	12.16	3.21	41.81
5#(40x20)	19.04	11.51	17.742

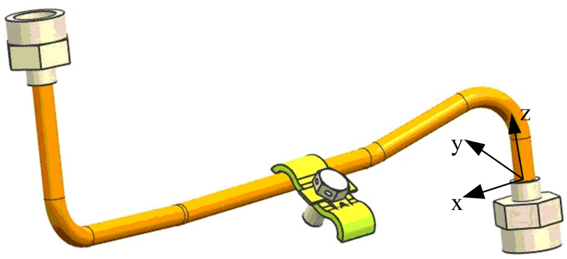
From the analysis results, the vibration stress value of point 1 and point 2 of the 4# pipeline reduced at the excitation frequency of 650 Hz, but because the first-order natural frequency is closer to the excitation frequency of 194 Hz, the vibration stress is relatively large.

Based on the results of the calculation of the natural frequency and stress values, it is possible to draw an important conclusion that the effect of changing the pipe shape on reducing the vibration stress of the pipeline is not obvious.

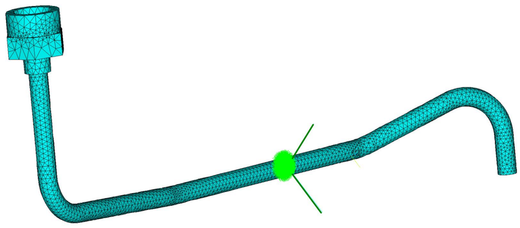
4 Influence of the position of the clamp on the inherent characteristics and vibration response of the pipeline

In the dynamic design of the pipeline, the resonance frequency of the excitation source can be avoided by changing the layout of the pipeline. However, due to the limited installation space, it is difficult to make major changes in the pipeline direction. Since the clamp is connected to the structural bracket or other pipelines, it is easy to change its position and is one of the effective means to improve the dynamic characteristics of the pipeline system.

In the finite element analysis, the clamp can be simplified to a fixed-end spring unit with a certain rigidity in both directions, and it is assumed that the portion where the clamp is clamped does not undergo local deformation, as shown in Fig. 5.



(a) Clamp position



(b) Clamp unit

Fig. 5. The method of adding the clamp

The clamp stiffness parameter has a great influence on the vibration characteristics of the pipeline system. According to the test and finite element analysis, the transverse stiffness of the double clamp is taken as $K_y = 10^5\,\mathrm{N/m}$, $K_z = 10^6\,\mathrm{N/m}$ and the COMBIN14 unit is added in the y and z directions of the pipe clamp, and the stiffness value is defined in the form of a real constant to simulate clamp constraint on the pipeline. Calculate the 5th natural frequency of the pipeline after adding the clamp, see Table 3.

It can be seen from Table 3 that the first-order natural frequency of the pipeline after adding the clamp is much larger than the first-order natural frequency of the original pipeline system (see Table 1).

Table 3. Natural frequencies after adding clamps to different pipe types

Order	1#(Experim ental pipe)	2#(85)	3#(62.5)	4#(40)	5#(40)
1	287.39	296.69	288.75	279.96	297.26
2	300.82	312.48	303.29	293.11	313.68
3	592.69	572.06	557.72	514.41	590.89
4	1038.2	1080.8	979.35	874.09	1079.3
5	1343.1	1340.9	1279.8	1157.5	1376.6

Calculating the vibration stress distribution of the pipeline after adding the clamp constraint, and analyzing the influence of the clamp on the vibration stress distribution of the pipeline. Fig. 6 shows the distribution of the vibration stress in the first 4th mode.

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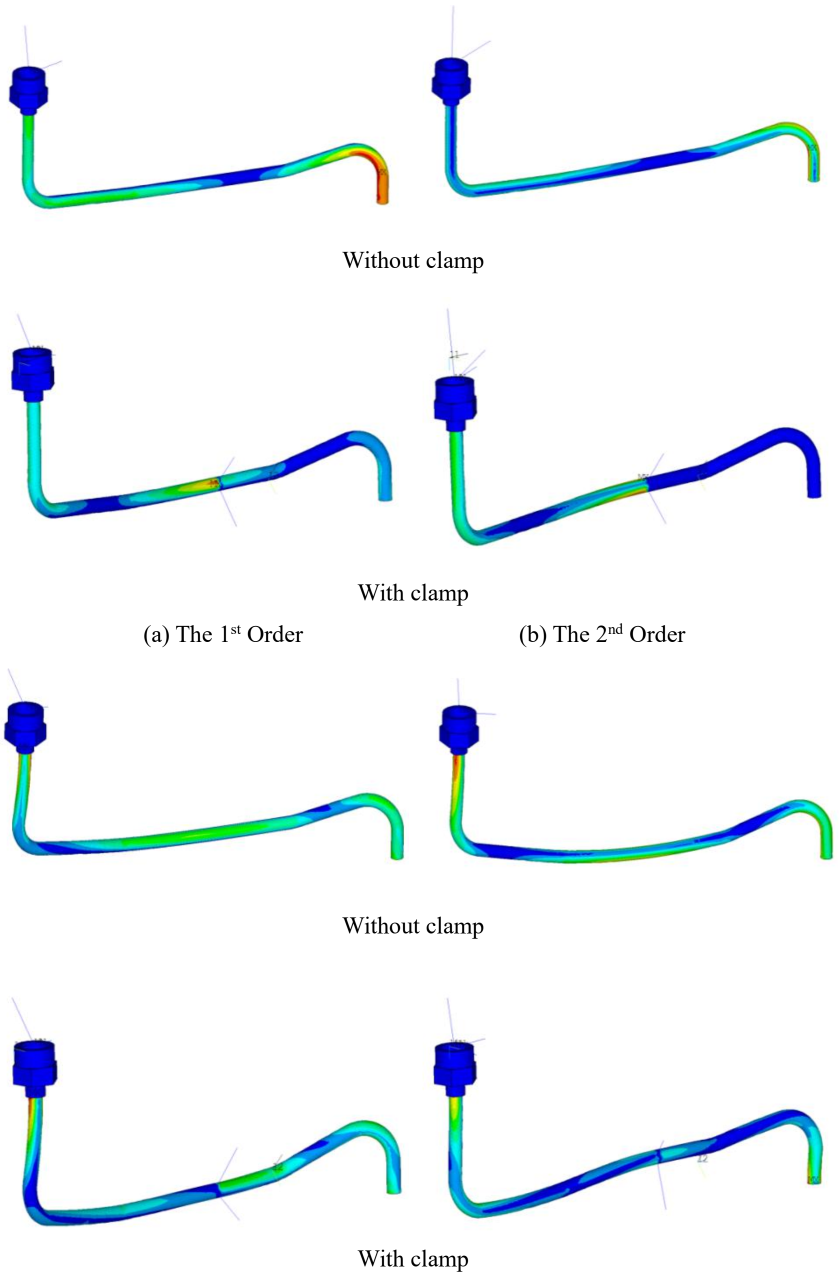
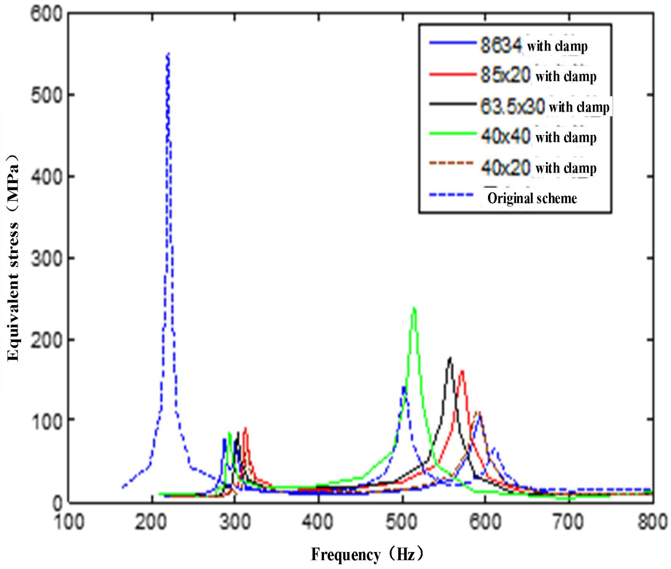


Fig. 6. Vibration stress distribution of the first 4 modes after adding the clamp

It can be seen from Fig. 6 that after adding the clamp, the first and second order modes have the largest vibration stress, mainly distributed at the elastic end and the clamp, and the maximum value appears at the clamp. The vibration stress at one end of the rigid support is relatively small. The vibration stresses of the third and fourth orders are still mainly distributed at both ends of the pipeline.

Fig. 7 is the Von Mises stress-frequency curve of point 1 and point 2 after adding the clamp obtained by harmonic response analysis. It can be seen from Fig. 7(a) that the first order natural frequency after each pipe is added with the clamp constraint The 219 Hz is very far, and the first-order resonance stress value of the rigid connection end (point 1) significantly decreased, and the second-order resonance stress value is not particularly changed.



(a) Stress value obtained by harmonic response analysis at point 1

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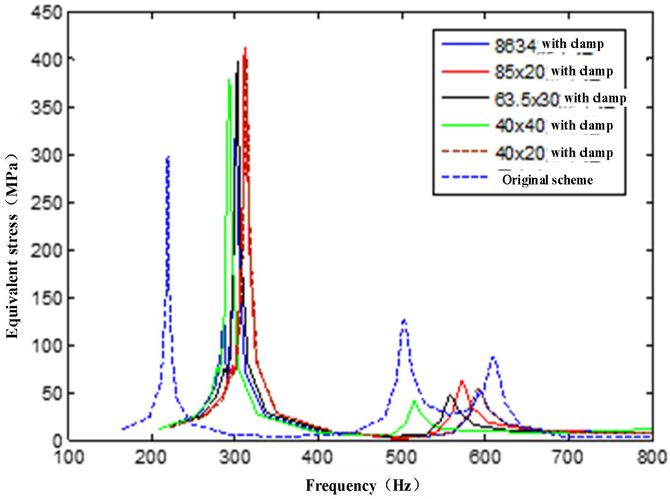

(b) Stress value obtained by harmonic response analysis at point 2

Fig. 7. Stress-frequency curves of points 1 and 2 obtained from harmonic response analysis

It can be seen from Fig. 7(b) that for the elastic end (point 2), the first-order resonance stress value increases, but because it is far away from the excitation frequency of 219 Hz, there will not be huge influence on the vibration fatigue of the pipeline structure. The second-order resonance stress value is significantly lowered.

Because the natural frequency of the first order of the pipeline after adding the clamp is much larger than 194 Hz, the vibration stress values of the rigid connecting end and the elastic end of the pipeline under 650 Hz excitation are mainly investigated below. From the stress-frequency curve obtained from the harmonic response analysis, the vibration stress values of points 1 and 2 at 650 Hz excitation are extracted and listed in Table 4 to compare with the original program.

Table 4. Vibration stress values of 650Hz excited by different pipe type clamps

Plan	Point 1	Point 2
1#Without Clamp (Experimental pipe)	17.35	12.44
1#With clamp (Experimental pipe)	16.74	13.22
2# With clamp (85x20)	16.74	9.88
3# With clamp (62.5x30)	8.80	8.55
4# With clamp (40x40)	9.05	7.56
5# With clamp (40x20)	15.31	12.75

It can be seen from Table 4 that the 3# and 4# pipe types obtained the smallest the vibration stress values at 650 Hz excitation after adding the clamps. Compared with the original scheme, the vibration stress values of the rigid joint and the non-rigid joint are decreased by 49.28%, 31.27%, 47.84% and 39.23%, respectively.

5 Conclusion

Aiming at excessive vibration problem of a certain type of engine, two improved schemes of structural optimization and increased clamp were proposed. The finite element modeling of hydraulic pipeline system was established and the modal analysis and harmonic response calculation were carried out. Five different pipelines were analyzed. The influence of the bending angle and the presence or absence of the clamp on the inherent characteristics and vibration response of the pipeline were analyzed.

Several main conclusions are concluded as follows:

(1) The influence of the slight change of the pipe bending angle on the inherent characteristics of the pipeline is not obvious, and the effect of changing the pipe shape on reducing the vibration stress of the pipeline is not obvious.

(2) After adding the clamp, the first and second modes of the pipeline have the largest vibration stress, mainly distributed at the elastic end and the clamp, and the maximum value appears at the clamp. The vibration stress at one end of the rigid support is relatively small. The third and fourth order vibration stresses are still mainly distributed at the two ends of the pipeline; the first order natural frequency of each pipe with clamp deviates, far from 219 Hz, and for the rigid connection end, the first-order resonance stress value decreased significantly, and the second-order resonance stress value did not change significantly. For the elastic support end, the first-order resonance stress value increased, but because it was far away from the excitation frequency of 219 Hz, and did not have much influence on the pipeline structure vibration fatigue, and the second-order resonance stress value decreases significantly; after adding the clamps to the 3# and 4# pipes, the vibration stress values at 650 Hz excitation are the smallest.

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