





Analysis and Treatment of Gas Pulsation in the Pipeline of a Hydrogen Diaphragm Compressor

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Abstract. Based on the plane wave theory and the transfer matrix method, the modeling, calculation and analysis of gas pulsation in a complex pipeline system of a hydrogen diaphragm compressor were carried out. The calculations show that the gas pulsations of the intake and exhaust pipelines of the cylinder gradually attenuate along the pipeline upstream and downstream, and the gas pulsation of the exhaust pipeline of the second stage cylinder is larger. The gas pulsations of the exhaust pipelines of the first and second stage cylinders are significantly higher than the limit values of API 618. From the perspective of pulsation control, an attempt was made to add orifice plates at appropriate positions of the pipeline to reduce the pulsation amplitude. The trial calculations show that the added orifice plate can effectively reduce the maximum gas pulsation of the pipeline, and can provide a useful reference for the treatment of gas pulsation.

Keywords: Gas pulsation · Hydrogen diaphragm compressor · Pipeline system · Plane wave theory

1 Introduction

Hydrogen energy, as an important technology path for sustainable energy development and strategic transformation, is becoming a hot spot in the field of global energy and transportation. With the rapid development of the hydrogen energy industry, the demand for terminal hydrogen will increase significantly. Hydrogen compressor is an important power source for compressing and transporting hydrogen, which is widely used in hydrogen storage and transportation, hydrogen station and gas hydrogen canning. As the core equipment of the pressurization system in the hydrogen storage and transportation process, the reliability of the diaphragm hydrogen compressor has received increasing attention. The diaphragm hydrogen compressor is a positive displacement compressor. The intermittent suction and discharge will cause strong gas pulsation in the pipeline. Gas pulsation not only affects the working performance of the compressor, but also induces severe vibration of the pipeline, which has a significant impact on the safe operation of the hydrogenation station. Therefore, studying the gas pulsation characteristics

of the hydrogen compressor pipeline and finding the methods to suppress the gas pulsation have important theoretical significance and application value for improving the long-term safety and reliability of the hydrogen compressor and the hydrogen storage and transportation system.

At present, the existing studies on pipeline gas pulsation, mainly adopt the electro-acoustic analogy method [1, 2], transfer matrix method [3] based on the plane wave theory [4], full three-dimensional CFD method [5] and one-dimensional CFD method [6–8]. Pipeline gas pulsation is essentially a fluid dynamics problem, which can be solved theoretically by CFD. However, there are many branches and pipeline elements in the actual pipeline system, and the internal structures of mixers, separators, heat exchangers, valves and other elements as well as the suction and exhaust passages of cylinders are very complex, which brings huge modeling difficulties and time costs to the three-dimensional flow field analysis. The frequency domain method based on the plane wave theory is suitable for the acoustic analysis of complex pipelines, which can directly obtain the accurate analytical solution of gas pulsation. It has low computational complexity and can accurately simulate the pressure pulsation of the pipeline gas flow. In this paper, the gas pulsation of a two-stage diaphragm hydrogen compressor is analyzed based on the plane wave theory, and the gas pulsation is effectively suppressed by adding orifice plates at appropriate positions of the pipeline.

2 Calculation Method of Gas Pulsation

2.1 Plane Wave Theory

The pipeline gas flow is an oscillating system. According to the combination of different pipeline elements (such as constant cross-section pipe, snubber, tee branch) and different boundaries (such as close end, open end, anechoic end, etc.), the gas pulsation presents different characteristics. In the early 1960s, Kinsler and Frey [4] proposed the plane wave theory. Based on the continuity equation and motion equation, the theory describes the transfer of mass and momentum in the pipeline fluid, and can accurately simulate the pressure pulsation of gas flow.

Under normal circumstances, the pulsation value of the gas pressure in the pipeline is a small value relative to the average value of the pressure (generally within 8% in terms of amplitude), which conforms to the assumption of the plane wave theory. Therefore, the gas pulsation in the pipeline of the reciprocating compressor can be analyzed using the plane wave theory. The wave equation with linear damping can be expressed as

$$\frac{\partial^2 p_t}{\partial x^2} - \frac{R}{a^2} \frac{\partial p_t}{\partial t} - \frac{1}{a^2} \frac{\partial^2 p_t}{\partial t^2} = 0 \quad (1)$$

where p is the pressure, and the subscript t denotes the pulsation value. a is the sound of speed, $R = 4fu/D$ is the damping coefficient along the pipeline, f is the friction coefficient between gas and pipe wall, D is the pipe diameter, and u is the average flow velocity of pipe gas. The solution of (1) is

$$p_t = [A * e^{(\alpha+j\omega/a)x} + B * e^{-(\alpha+j\omega/a)x}]e^{j\omega t} \quad (2)$$

where $\alpha = R/(2a)$.

The equation of motion with damping can be expressed as

$$\frac{\partial \xi_t}{\partial t} + R\xi_t + S \frac{\partial p_t}{\partial x} = 0 \quad (3)$$

where $\xi_t = \rho S u_t$ is the pulsation massflow, and S is the cross-sectional area of pipe. The solution of (3) is

$$\xi_t = -[A * e^{(\alpha+j\omega/a)x} - B * e^{-(\alpha+j\omega/a)x}] e^{j\omega t} S/a \quad (4)$$

A complex pipeline system is composed of basic elements such as constant cross-section pipes, vessels, confluence points, reducers, etc. During the calculation, the pipeline system is first divided into many elements, each element representing different pipeline elements. First the transfer matrix of each element is obtained, and then the relationship of the pulsating pressure and the pulsating velocity between the nodes of each element can be established according to the transfer matrix. Assuming that the pulsating pressure and pulsating velocity at the beginning and terminal end of the compressor pipeline system are $p_{t,1}$ 、 $u_{t,1}$ 、 $p_{t,n}$ 、 $u_{t,n}$ respectively, then the following transfer relationship exists

$$\begin{bmatrix} p_{t,n} \\ u_{t,n} \end{bmatrix} = M_{n,n-1} M_{n-1,n-2} \cdots M_{21} \begin{bmatrix} p_{t,1} \\ u_{t,1} \end{bmatrix} \quad (5)$$

where $M_{n,n-1}$ is the transfer matrix between nodes. According to the boundary conditions at both ends of the pipeline system, the natural frequency of the gas column in the pipeline system can be obtained.

2.2 Limitation of Pipeline Pressure Pulsation

The fifth edition of API 618 defines the pressure pulsation p-p value of general pipes with average pressure below 350 bar as follows:

$$P_1 = \sqrt{\frac{a}{350}} \frac{400}{(P_L \times D_I \times f)^{0.5}} \quad (6)$$

where P_1 is the p-p value of pulsation pressure corresponding to fundamental frequency and harmonic frequency, expressed as a percentage of the average pipeline pressure P_L . D_I is the inner diameter of the pipe, f is the fundamental frequency and harmonic frequency, and a is the speed of sound. The pressure pulsation p-p value of the suction and exhaust pipes of the compressor cylinder is limited as follows:

$$P_1 = 0.03R \quad (7)$$

where P_1 is the same as the above, and R is the exhaust/suction pressure ratio of the cylinder.

3 Acoustic Analysis

The research object is a two-stage double cylinder water-cooled hydrogen diaphragm compressor. The secondary exhaust pressure is more than 20 MPa. The gas pulsation in the inlet and outlet pipeline may cause strong vibration of pipeline components. The compression process of the two-stage double cylinder water-cooled hydrogen diaphragm compressor is to compress the hydrogen stored in the hydrogen storage tank to a pressure of more than 20 MPa through two stages and send it into the high-pressure hydrogen bottle. The inlet pressure range is 1.6 ~ 2.0 MPa, the rotation speed is 420 rpm, the piston stroke is 180 mm, the exhaust pressure is 20 ~ 22 MPa, the exhaust temperature is ≤ 150 °C, and the capacity of the compressor is 1000 Nm³/H. Figure 1 shows the CAD modeling of the compressor cylinder and intake and exhaust pipeline. In the drawing, the primary intake pipeline is marked with white, the primary exhaust to secondary intake pipeline is marked with blue, and the secondary exhaust pipeline is marked with purple and red.

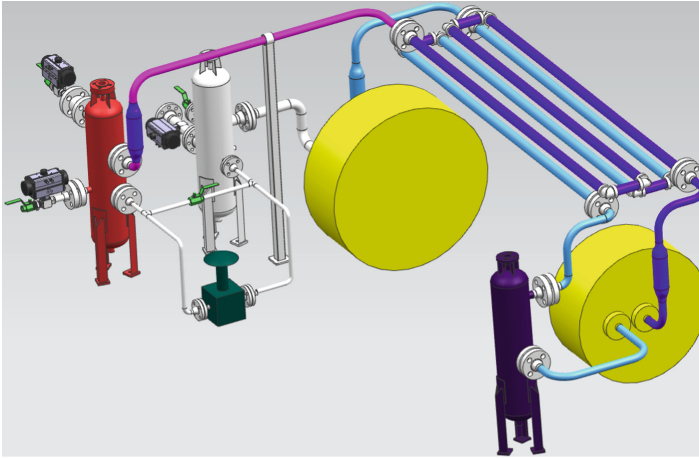


Fig. 1. Pipeline system model of a two-stage hydrogen diaphragm compressor

The compressed working medium is hydrogen. For the interstage pipeline, the first stage exhaust temperature is 6.2 MPa, and the second stage inlet pressure is 6 MPa.

Any changes to the drawings and process parameters can lead to changes in the calculation results. Considering that the primary intake snubber and the secondary exhaust snubber are connected to the upstream and downstream pipelines respectively, and the pulsation boundary conditions of upstream and downstream pipelines cannot be accurately given, and the volume of the snubber is much larger than the stroke volume, the primary intake snubber and its upstream pipeline, the secondary exhaust snubber and its downstream pipeline are not modeled in this study, and the open end boundary condition is given at the position of the pipeline connecting the snubber for pulsation calculation. The first stage cylinder is close to the intake snubber, and the pressure pulsation of this segment of pipeline is low. The calculation shows that the pressure pulsation of the intake

pipeline is far lower than the limit value in API 618, so the following discussion is only for the interstage pipeline and the second stage exhaust pipeline.

3.1 Pipeline from Primary Exhaust to Secondary Intake

The acoustic model of the pipeline from the 1st-stage exhaust to the 2nd-stage intake is shown in Fig. 2. The calculation results show that the pressure pulsation of the pipeline segment from the 1st-stage cylinder to the outlet of the 1st-stage cooler significantly exceeds the limit value of API 618. The pulsation amplitudes excited by the 6th and 14th harmonics are the largest. Within the $\pm 5\%$ change range near the rated speed, the maximum pressure pulsation is excited by the 14th harmonic of 420 rpm. The maximum pressure pulsation location is the cylinder outlet node. The pressure pulsation of the pipeline after the outlet of the primary cooler is significantly reduced, and the pressure pulsation of the pipeline segments before and after the snubber downstream of the cooler is not weakened.

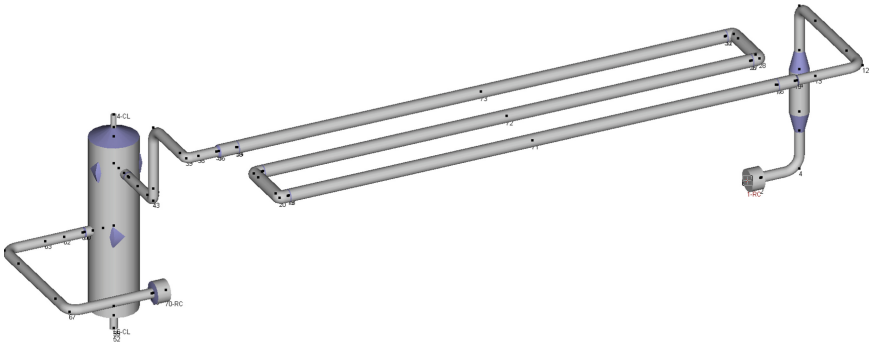


Fig. 2. Acoustic model of interstage pipeline

Figures 3, 4 and 5 respectively show the time-domain and frequency-domain data of pressure pulsation at the outlet node of the 1st-stage cylinder, the inlet node of the 1st-stage cooler and the outlet node of the 1st-stage cooler. The time-domain curve shows the pressure pulsation in several cycles under the rated speed, and the spectrum distribution shows the pulsation amplitude of each harmonic within the $\pm 5\%$ speed change range and the curve representing the limit value of API 618. It can be seen from the figure that the sixth and fourteenth harmonic amplitudes of the pressure pulsation at the inlet node of the primary cooler have exceeded the limit value of API 618, and the sixth harmonic amplitudes at the outlet node of the cooler have been fully attenuated, but the fourteenth harmonic amplitudes are still significantly higher than the limit value.

3.2 Secondary exhaust pipeline

The acoustic model of the secondary exhaust pipeline is shown in Fig. 6. The calculation results show that the pressure pulsation of the secondary exhaust pipeline has greatly

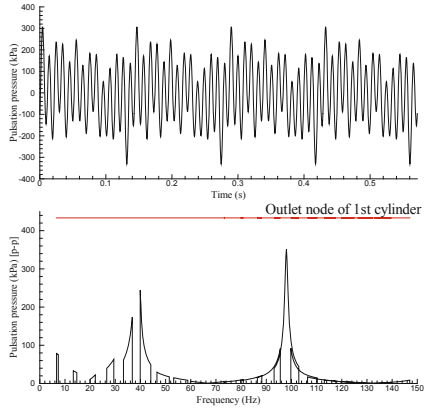


Fig. 3. Temporal and spectrum diagrams of pressure pulsation at the outlet node of 1st cylinder

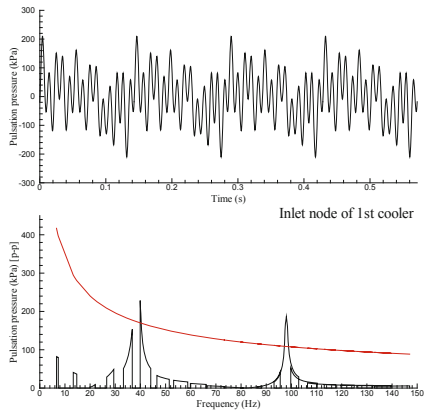


Fig. 4. Temporal and spectrum diagrams of pressure pulsation at the inlet node of 1st cooler

exceeded the limit value of API 618. The pulsation energy is basically contributed by the 4th and 14th harmonic components. Within the $\pm 5\%$ change range near the rated speed, the maximum pressure pulsation is excited by the 4th harmonic of 421 rpm. The maximum pressure pulsation is located at the outlet node of the secondary cylinder. After the cylinder outlet, the pressure pulsation gradually decreases with the decrease of the distance from the exhaust snubber, and the pressure pulsation in the cooler pipeline segment decreases greatly.

Figures 7, 8 and 9 respectively show the time-domain and frequency-domain data of pressure pulsation at the outlet node of the secondary cylinder, the inlet node of the secondary cooler and the outlet node of the secondary cooler. It can be seen from the figure that the amplitudes of the 4th harmonic and the 14th harmonic of pressure pulsation at the inlet and outlet nodes of the cooler are much higher than the limit value of API 618, and the amplitudes of the 14th harmonic at each node are similar. However, the limit value of the cylinder outlet pipe calculated according to formula (7) is higher than

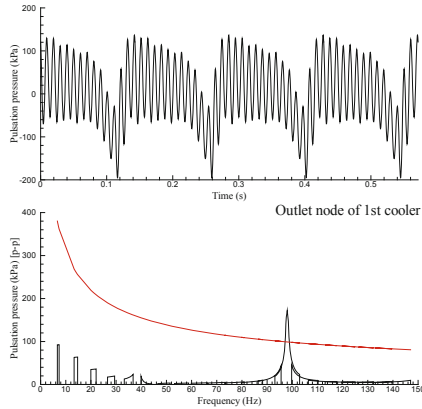


Fig. 5. Temporal and diagrams of pressure pulsation at the outlet node of 1st cooler

the limit value of the general pipe calculated according to formula (6), so the pulsation excited by the 14th harmonic here does not exceed the limit value.

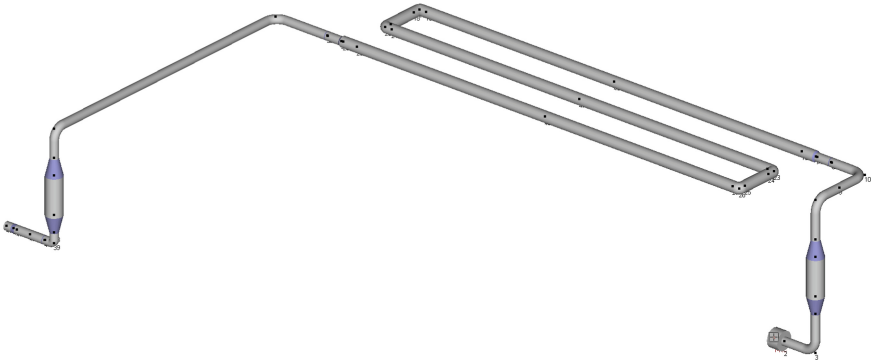


Fig. 6. Acoustic model of 2nd discharge pipeline

4 Study on pulsation suppression

In the relevant standards of petrochemical pipelines, it is clearly required that for the pulsation suppression of super-high pressure pipelines, the rectification scheme must be determined through acoustic analysis. On the basis of acoustic calculation of pipeline and from the perspective of pulsation control, this paper attempts to add orifice plates at the proper positions of pipeline to weaken the pulsation amplitude.

4.1 Pulsation suppression scheme of interstage pipeline

An orifice plate was added to the interstage pipeline from the primary exhaust to the secondary intake to control the gas pulsation. It was found that the orifice plate with

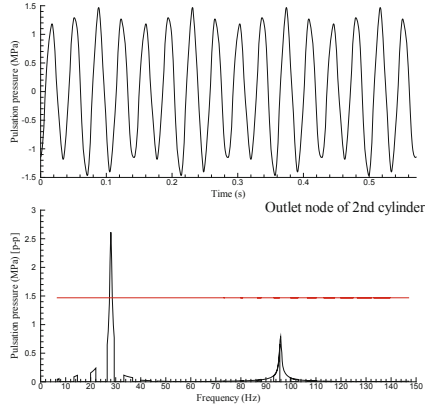


Fig. 7. Temporal and spectrum diagrams of pressure pulsation at the outlet node of 2nd cylinder

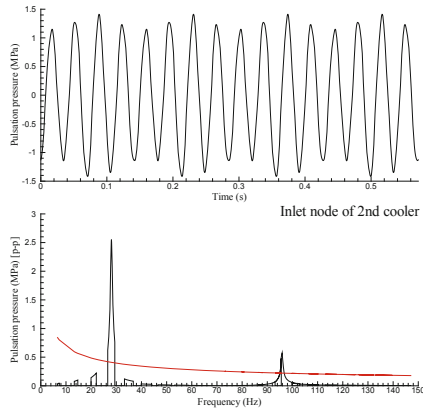


Fig. 8. Temporal and spectrum diagrams of pressure pulsation at the inlet node of 2nd cooler

a thickness of 35mm and a hole-diameter ratio of 0.25 was added after the primary cylinder outlet and before the reducer pipe in front of the cooler, which had a better effect on suppressing the high-order pressure pulsation. After the orifice plate is added at this position, the pulsation amplitudes excited by the 6th and 14th harmonics at the cooler and its front and rear pipeline segments are significantly reduced. The pipeline model near the location where the orifice plate is added is shown in Fig. 10, and the yellow component in the drawing is the added orifice plate. Figures 11, 12 and 13 are time-domain and frequency-domain diagrams of pressure pulsation at the outlet of the primary cylinder, the inlet and outlet nodes of the primary cooler after adding orifice plates. It can be seen from the figures that the low-frequency pulsation amplitude at the cylinder outlet node increases, but it does not exceed the limit value of API 618, and the highest amplitude pulsation component caused by the 14th harmonic has been seriously weakened.

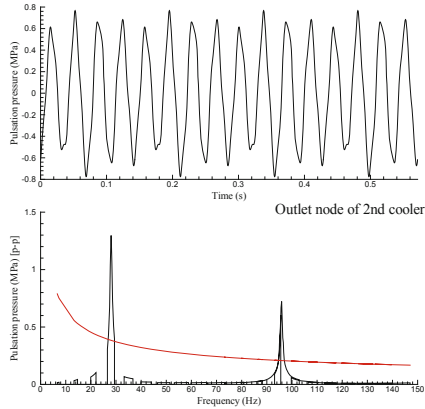


Fig. 9. Temporal and spectrum diagrams of pressure pulsation at the outlet node of 2nd cooler

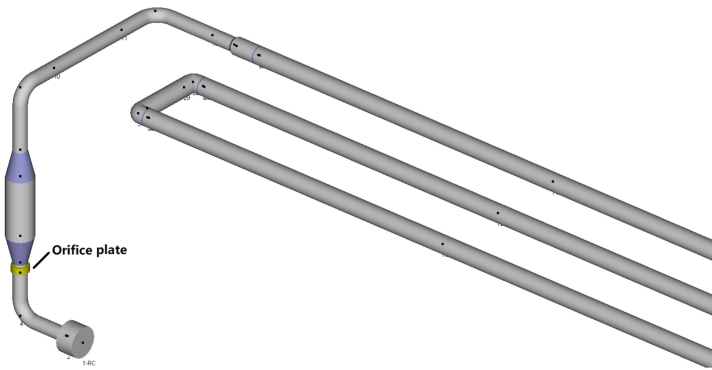


Fig. 10. Pipeline near the orifice plate (interstage pipeline)

4.2 Pulsation Suppression Scheme of 2nd-Stage Exhaust Pipeline

It is found that adding orifice plates with a thickness of 35mm and a hole-diameter ratio of 0.2 in front of the elbow after the outlet of the secondary cylinder and before the cooler has a better effect on suppressing gas pulsation. After the orifice plate is added at this position, the pulsation amplitudes excited by the 4th and 14th harmonics in the whole pipeline is greatly reduced. The pipeline model near the location of the added orifice plate is shown in Fig. 14. Figures 15, 16 and 17 are time-domain and frequency-domain diagrams of pressure pulsation at the outlet of the secondary cylinder, the inlet and outlet nodes of the secondary cooler. It can be seen from the figure that after the orifice plate is added, the low-frequency pressure pulsation of the 1st to 3rd harmonics at the cylinder outlet is slightly enlarged, but it is still far below the limit value of API 618. The large-scale pressure pulsation component of the 4th harmonic is greatly suppressed, and the high-frequency pressure pulsation including the 14th harmonic basically disappears.

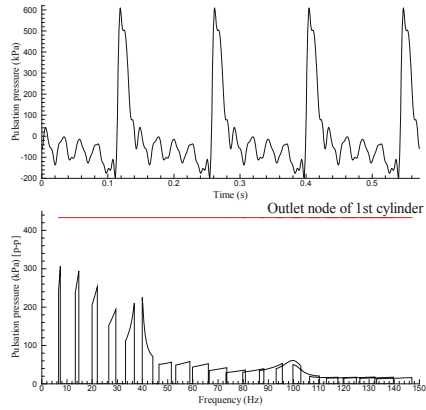


Fig. 11. Temporal and spectrum diagrams of pressure pulsation at the outlet of 1st cylinder after adding orifice plate

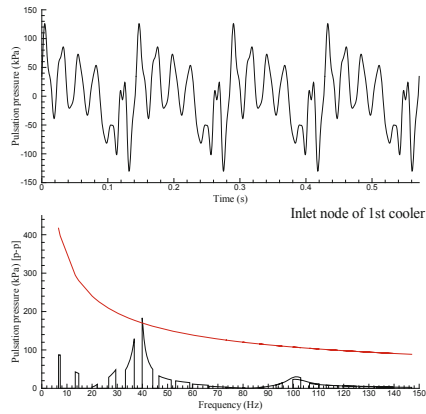


Fig. 12. Temporal and spectrum diagrams of pressure pulsation at the inlet of 1st cooler after adding orifice plate

5 Conclusions

Aiming at the analysis of gas pulsation in complex pipeline system of hydrogen compressor, on the basis of mastering the compressor parameters and pipeline data, the gas pulsation modeling and calculation analysis of two-stage pipeline of compressor are carried out based on the plane wave theory and transfer matrix method. The results show that the pressure pulsations of the primary intake and secondary exhaust pipelines gradually attenuate along the pipeline upstream and downstream. The pressure pulsation of the primary intake pipeline is relatively small, and the pulsation value is lower than the limit value of API 618. The pressure pulsation of the exhaust pipeline of the secondary cylinder is larger, and the pressure pulsations of exhaust pipelines of the primary and secondary cylinders are significantly higher than the limit values of API 618. On the basis of pulsation analysis, the attempt to add orifice plates to suppress gas pulsation is

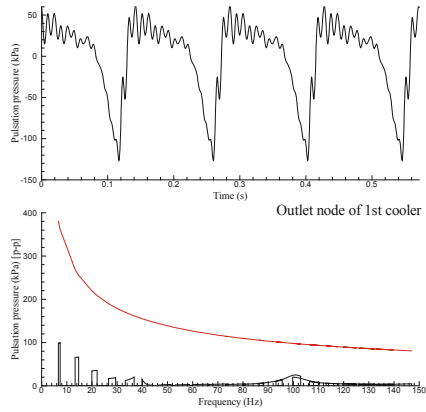


Fig. 13. Temporal and spectrum diagrams of pressure pulsation at the outlet of 1st cooler after adding orifice plate

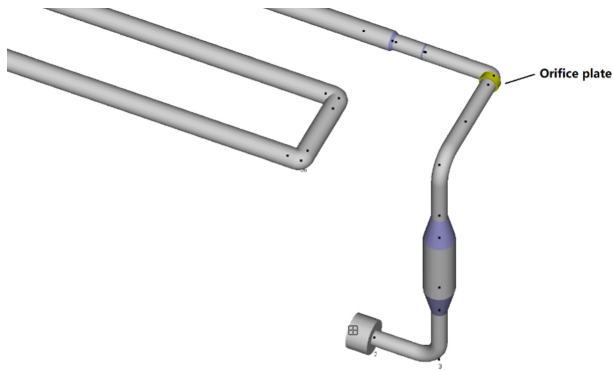


Fig. 14. Pipeline near the orifice plate (2nd discharge pipeline)

further carried out. The results show that the added orifice plates effectively reduce the maximum pressure pulsation of the pipeline, which can provide a useful reference for the follow-up pulsation treatment.

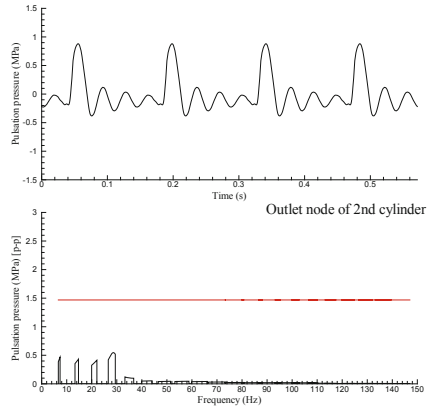


Fig. 15. Temporal and spectrum diagrams of pressure pulsation at the outlet of 2nd cylinder after adding orifice plate

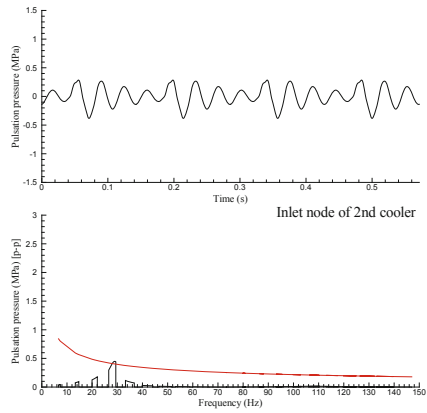


Fig. 16. Temporal and spectrum diagrams of pressure pulsation at the inlet of 2nd cooler after adding orifice plate

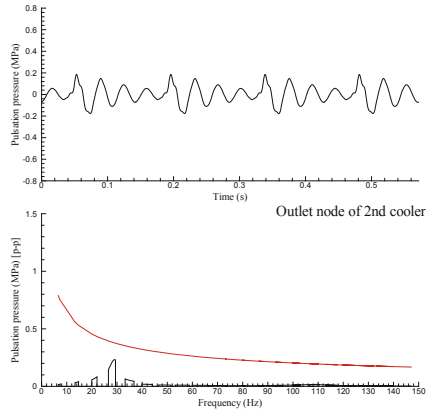


Fig. 17. Temporal and spectrum diagrams of pressure pulsation at the outlet of 2nd cooler after adding orifice plate

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