2016

Analysis And Control Of Severe Vibration Of A Screw Compressor Outlet Piping System

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Analysis and Control of Severe Vibration of a Screw Compressor Outlet Piping System

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ABSTRACT

The severe vibration of a screw compressor outlet piping system caused the fatigue failure of some thermowells and the unscheduled shut down of the system. The main reasons of the abnormal vibration in the outlet piping system were investigated by developing an acoustic model to simulate the gas pulsation and establishing two finite element models to conduct the mechanical vibration analyses. The acoustic analysis results showed that the pulsation amplitudes of most nodes in the outlet piping system exceeded the allowable values. The results of mechanical vibration analyses indicated that the insufficient stiffness of the outlet piping system and the first-order structure resonance occurred on thermowells were also the key factors inducing vibration. Several methods were put forward to attenuate vibration amplitude of the outlet piping system as well as the thermowells. A new pulsation attenuator was installed and the piping layout was rearranged to reduce pulsation amplitudes and shaking forces of all nodes in the outlet piping system. Several reasonable supports were added to improve the stiffness of the outlet piping system. After reinforcing the thermowells, the first-order natural frequency of the thermowells increased from 207.4Hz to 280.7Hz, away from the excitation frequency of 196.67Hz. The field measurement results showed that vibration amplitude and the vibration velocity decreased significantly after modifications.

1. INTRODUCTION

Compressors are widely utilized in the petrochemical industries. Piping vibration is a potential threat to the stability and reliability of the compressor system. Gas pulsation is one of the most important sources of piping vibration. Therefore, it is of great significance to investigate on gas pulsation and piping vibration of the compressor piping system.

The gas pulsation and piping vibration have drawn much attention since 1950s. The studies concerning gas pulsation and piping vibration mainly include theoretical research and engineering application. Kinsler and Frey (1962) deduced the equation of plane wave theory. Sakai *et al.* (1973a and 1973b) calculated the gas column natural frequencies of complex pipe system by the transfer matrix method. Benson (1972) proposed to simulate the gas pulsation in the piping system by using the one-dimensional non-isentropic flow theory. Soedel *et al.* (1978) investigated the acoustic characteristics of pulsation attenuator. Jia *et al.* (2015a) studied the influence of the orifice plate on gas pulsation in a reciprocating compressor piping system. Liang *et al.* Liu *et al.* (2013) investigated the acoustic attenuation and flow resistance characteristics of perforated tube silencers though numerical simulation. (2015b) estimated the main causes of the abnormal vibration in a reciprocating compressor inlet piping system through acoustic analysis, vibration analysis and field measurements, and took relevant treatments to eliminate the vibration. These research lays a good foundation for the analysis and control of vibration problems of the compressor piping systems.

This paper is devoted to investigate the main causes and elimination of the vibration problem of a screw compressor outlet piping system. An acoustic model and two mechanical models were established to simulate the gas pulsation and structure vibration of the screw compressor outlet piping system. The main causes of abnormal vibration in the outlet piping were investigated and relevant elimination treatments were put forward. The good performance of the improvement measures were verified by field vibration measurements.

2. FIELD VIBRATION PROBLEM

The serious vibration of a screw butadiene compressor outlet piping system caused the unscheduled shut down of the system. The compressor operates at rated condition most of the time. The rated speed is 2950 rpm, rated motor power is 1500 kW, rated flow rate is 15277 $m^3.h^{-1}$, the inlet pressure and temperature are 159 kPa and 318 K, and the outlet pressure and temperature are 530 kPa and 374 K, respectively. The vibration velocity at the first elbow after the outlet silencer was 38.8mm/s, much exceeding the allowable value. Worse still, the small-bore appendages attached to the mainline vibrated violently, and the fatigue fracture happened on the junction of some thermowells and mainline, as shown in Figure 1. Although the temporary measure of changing the elastic support near the outlet silencer to guide support was adopted, there were still severe vibrations. As a result, vibration caused analysis and eliminating measures are urgently needed to ensure the safety and reliability of the system.

Figure 1: Fatigue fracture of the thermowells

3. VIBRATION CAUSES ANALYSIS

3.1 Acoustic Analysis

The gas pulsation is an important excitation source of vibration in the compressor piping system. Eliminating the gas pulsation in the piping system within the allowable range is an effective measure to control the vibration in the compressor piping system. The plane wave theory and transfer matrix method were both applied to analyze the gas pulsation of the outlet piping system. Accordingly, the piping system can be divided into several basic elements such as pipe, valve, tee and volume. Each element has a transfer matrix *M*. For example, the transfer matrix of the uniform section straight pipe is given by Equation (1). The transfer matrix of the overall piping system is formulated by multiplying the transfer matrixes of all the piping elements as Equation (2).

$$M = \begin{bmatrix} \cos(\omega l / a) & -\rho_0 a \sin(\omega l / a) \\ -\dfrac{1}{\rho_0 a}\sin(\omega l / a) & \cos(\omega l / a) \end{bmatrix} \tag{1}$$

$$\begin{bmatrix} p_{end} \\ u_{end} \end{bmatrix} = M_n \cdot M_{n-1} \cdots M_2 \cdot M_1 \cdot \begin{bmatrix} p_{start} \\ u_{start} \end{bmatrix} \tag{2}$$

The acoustic natural frequency can be obtained by applying relevant boundary conditions of the piping system. Then, the pressure unevenness defined by Equation (3) and corresponding shaking forces can be calculated.

$$\delta = \frac{(p_t)_{\max} - (p_t)_{\min}}{\frac{1}{2}\left[(p_t)_{\max} + (p_t)_{\min}\right]} \tag{3}$$

The acoustic model of the outlet piping system was shown in Figure 2. The outlet piping system was divided into 84 elements and 103 nodes. The compressor boundary conditions were applied to node 1, 4, 7, and 9. The nonreflecting boundary condition was applied to node 45, and the boundary conditions of thermowells and pressure gauge branches were applied to node 46-58.

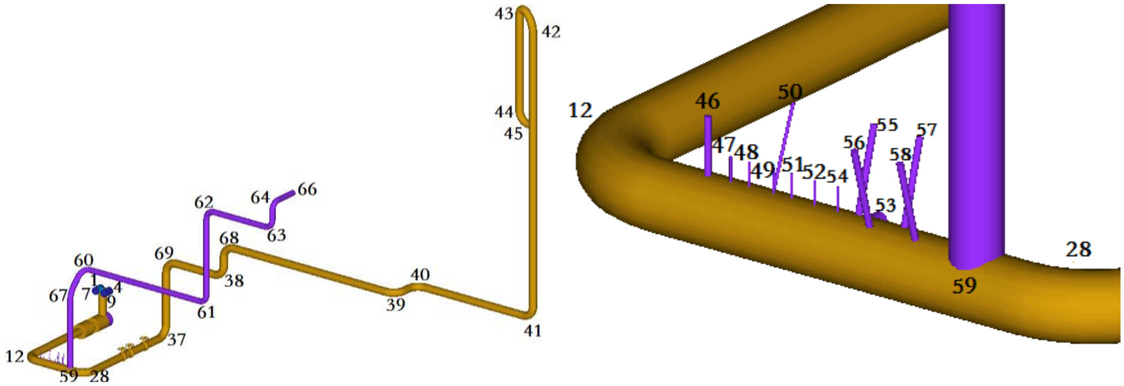

Figure 2: Acoustic model of the outlet piping system

The acoustic analysis results showed that the pressure unevenness of most nodes exceeded the allowable value of 2% under the operating condition. The maximum pressure unevenness was 15.5%, located on a pressure gauge branch (node 53). Therefore, the first step in controlling the severe vibration is to eliminate the gas pulsation of the piping system.

3.2 Mechanical Vibration Analyses

Piping system is a continuous elastic structure system, the structure resonance will be induced when the excitation frequency is within ±10% of the mechanical natural frequency of the piping system. Therefore, the mechanical vibration analysis plays an important role in the design of piping systems. Modal analysis is an effective method to investigate natural frequencies and mode shapes of a structure. Dynamic response analysis can be used to evaluate the piping system design due to the operating load cases. Since the mainline as well as small-bore appendages had severe vibration, the vibration analysis of overall outlet piping system and the thermowells were performed, respectively.

3.2.1 Vibration analysis of overall outlet piping system

The mechanical model of the outlet piping system was established, and the modal analysis and harmonic response analysis were performed. The pipes were modeled with beam elements, and the valves were simplified as rigid elements. Small-bore appendages were modeled and analyzed separately in the following and not included in the overall pipeline analysis.

Modal analysis results showed that the first order natural frequency of the outlet piping system was only 0.9Hz; the modal shape of the natural frequency 198.8Hz, close to the excitation frequency of 196.67Hz, occurred mainly on the horizontal pipe behind the first elbow that the small-bore appendages attached, as shown in Figure 3. The results indicated that the outlet pipeline was prone to vibrate due to insufficient stiffness.

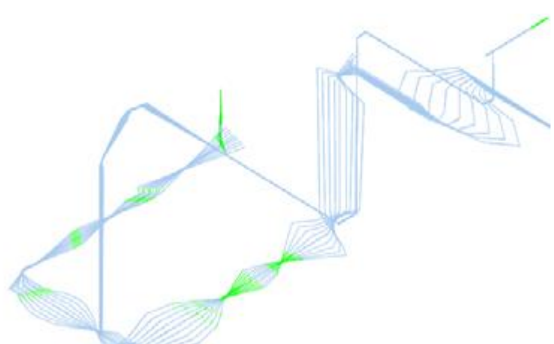

Figure 3: The mode shape at the MNF of 198.8Hz

The harmonic response analysis was performed under the action of calculated shaking forces by the acoustic analysis. The vibration velocities under the excitation frequency of 196.67Hz were calculated and compared with the field measured data, as shown in Figure 4. It can be seen that the calculated values agreed well with the measured values. Accordingly, the mechanical model of the outlet piping system and the shaking forces were reliable, which indirectly verified the reliability of acoustic model of the piping system.

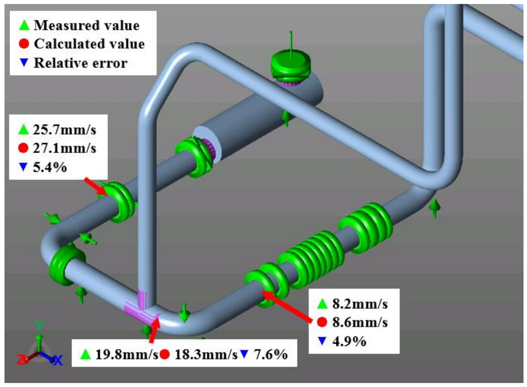

Figure 4: Comparison of vibration velocity between calculated and measured values

3.2.2 Vibration analysis of thermowells

Failures of small-bore appendages due to alternating stresses caused by vibration are typically occurred in compressor piping systems. Shifting the MNFs of the appendages away from the excitation frequency will go a long way toward avoiding such failures. A mechanical model including a thermowell, a section of the mainline and the connection flanges between them was established and the modal and harmonic response analysis were performed.

The modal analysis results showed that the first order natural frequency of the thermowell was 207.4Hz, near the excitation frequency of 196.67Hz, which means the first order structure resonance will occur on the thermowell. The harmonic response analysis results showed that the maximum equivalent stress of the model under the excitation frequency of 196.67Hz was 44.4MPa, appeared at the root of the thermowell, and prone to induce fatigue fracture of the thermowell, as shown in Figure 5.

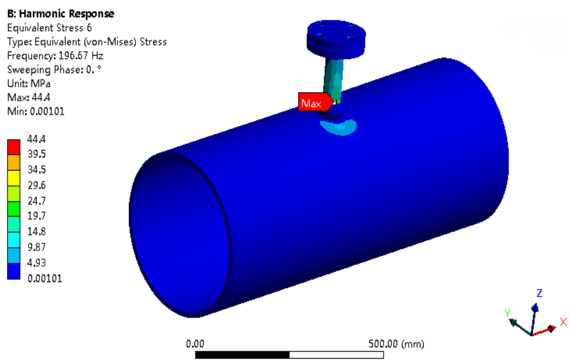

Figure 5: Stress distribution at the excitation frequency of 196.67Hz

4. VIBRATION ELIMINATING TREATMENTS

4.1 Pulsation Reduction Measures

At first, the pulsation reduction measure of adding a new pulsation attenuator was put forward. The 1650-milimeter-long straight pipe after the original discharge silencer was replaced by a pulsation attenuator of the same length, and the outer diameter of the new pulsation attenuator was 800mm, as shown in Figure 6. Acoustic analysis results of the modified model showed that the pulsation of the pipe after the new pulsation attenuator was significantly decreased, but the pressure unevenness of some small-bore appendages cannot meet the requirements of API 619.

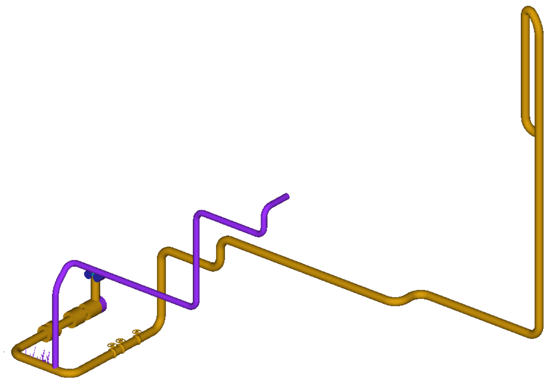

Figure 6: The first pulsation eliminating scheme

In order to eliminate the pressure unevenness and shaking forces at the small-bore appendages, another treatment including adding a new pulsation attenuator, adjusting the mainline, reflux pipeline and instrument branches was put forward. On the basis of the first scheme, the horizontal pipe after the first elbow was extended by 4000mm, the reflux pipeline was moved 1000 mm after the second elbow, and the layout of the instrument branches was adjusted, as shown in Figure 7.

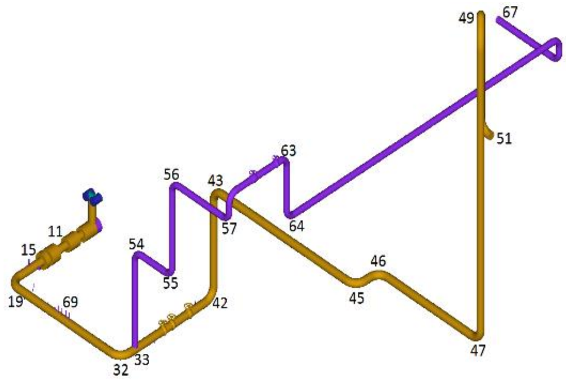

Figure 7: The second pulsation eliminating scheme

After modifications, both the pressure unevenness and shaking forces all over the piping system under designed operating conditions can meet the requirements of API 619; the maximum pressure unevenness was 1.53%, occurred on a thermowell (node 70), as shown in Figure 8.

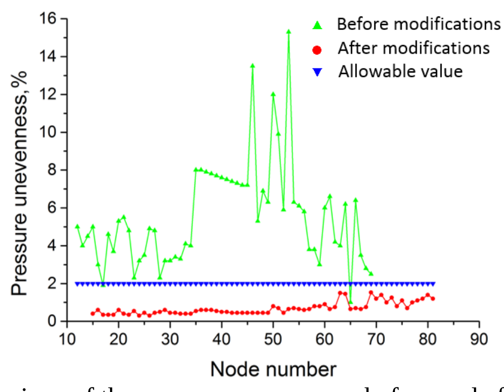

Figure 8: Comparison of the pressure unevenness before and after modifications

4.2 The piping vibration control measures

The vibration analysis results showed that the outlet pipeline with poor stiffness was prone to vibrate, and the fatigue fracture caused by the first-order structure resonance occurred on thermowells. Some supports were added to improve the stiffness of the overall piping system, as shown in Figure 9. The thermowells were reinforced by adding four gussets between the thermowells and the connection flanges, as shown in Figure 10.

- Two variable spring supports were added at the bottom of the pulsation attenuator (node 2022) to support the weight of the new pulsation attenuator (about 2 tons);
- Since the shaking forces in X direction can be generated at the first and the second elbow after the new pulsation attenuator, the poor stability temporary support before the first elbow (node 50) was changed to a guide support to improve the stiffness of the piping system in X direction.
- Two guide supports were added on the straight pipe between the first and the second elbows (node 200 and node 20260) to reduce the base motion of the mainline where the small-bore appendages attached.
- Since the shaking forces in Y direction can be generated at the tee which connected the outlet pipe and reflux pipe, the bearing support were added on node 250 to improve the stiffness of the piping system in Y direction.
- A guide support was added to the vertical pipe of reflux pipeline (node 565) to improve the stiffness of the reflux pipeline.
- A guide support was added to node 442 to improve the stiffness of the mainline.

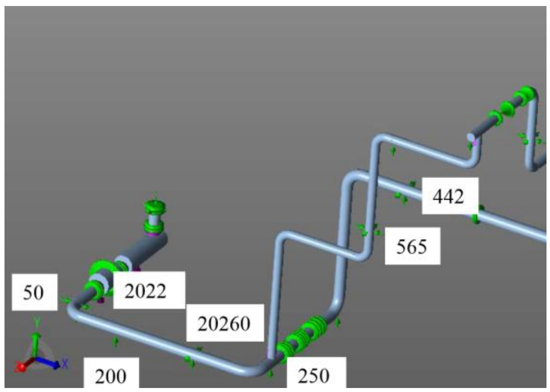

Figure 9: Vibration analysis model after modifications

After above modifications, the first order natural frequency of the outlet piping system improved from 0.9Hz to 1.67Hz, and the calculated maximum vibration velocity was 11.4 mm/s, far less than the allowable value, which indicated that the stiffness of the outlet piping system increased. The first order natural frequency of the thermowell improved from 207.4Hz to 280.7Hz, away enough from the excitation frequency of 196.67Hz, and the first order structure resonance could be avoided. The maximum equivalent stress of the modified model at the excitation frequency of 198Hz was 2.65 MPa, occurred on the welding location between a gusset and the thermowell, as shown in Figure 10, which indicated that the fatigue life of the thermowell was improved.

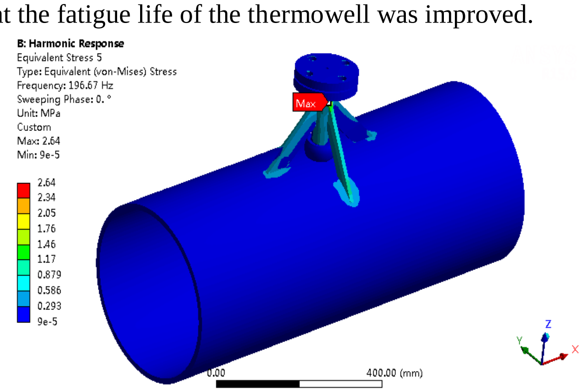

Figure 10: Stress distribution at the excitation frequency of 196.67Hz after modifications

4.3 Field Measured Results

The comparison of the field measured data before and after modifications is shown in Table 1. It can be seen that the maximum vibration amplitude dropped from 0.07mm to 0.017mm and decreased by 76%, and the maximum vibration velocity dropped from the 38.8mm/s to the 10.6mm/s and decreased by 73%. The vibration of the outlet piping system meet requirements of API 619, which indicated that the gas pulsation eliminating treatments and vibration control measures were effective.

Table 1: Comparison of the field vibration measurements data before and after modifications

	Before modifications	After modifications	Decreasing amplitude
The maximum horizontal vibration amplitude	0.046 mm	0.012 mm	72%
The maximum vertical vibration amplitude	0.07 mm	0.017 mm	76%
The maximum horizontal vibration velocity	21.4 mm/s	6.8 mm/s	68%
The maximum vertical vibration velocity	38.8 mm/s	10.6 mm/s	73%

5. CONCLUSIONS

The most important conclusions of the present paper were as follows:

- The main vibration causes of the severe vibration problem of the outlet piping system were investigated through the acoustic analysis, mechanical vibration analyses and field measurements. It was found that the excessive pulsation, low stiffness of the overall outlet piping system and the first-order structure resonance occurred on the thermowelles mainly contributed to the severe vibration.
- To control the abnormal vibration, some practical elimination treatments including adding a pulsation attenuator and some supports, changing the arrangement of the outlet piping system and reinforcing the thermowelles were carried out. The results of numerical analyses and field vibration measurements showed that the pressure pulsation, vibration amplitude and velocity after modifications satisfied well with standard requirements, which indicated that the treatments adopted indeed eliminated the vibration sources.

NOMENCLATURE

M	transfer matrix	(-)
a	sound speed	(m/s)
l	length	(m)
t	time	(s)
p	pressure	(Pa)
ω	angular frequency	(rad/s)
ρ	density	(kg/m^3)

Subscripts

start	the start node
end	the end node
max	the maximum value
min	the minimum value

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