Pipe Vibration Analysis and Structural Improvements of Reciprocating Compressor

Zheng Liang, Qiangbing Dong

***Abstract*— Avoiding pipe vibration is an important aspect of reciprocating compressor system design. By modal analysis upon a secondary compressor inlet pipe to get the natural frequency of the pipe, using the transfer matrix method to calculate the natural frequency of the pipe column. Compared with the excitation frequency of the compressor, it reveals the cause of the pipe vibration is mainly due to the natural frequency of pipe column in the region of the compressor excitation frequency. Calculating the pipe stresses under the pulsating action of the exciting force and it far exceeds the allowable stress of pipe. By changing the secondary intake pipe length, increasing intake surge tank and theoretical analysis of the system show that changes in the structure of solution effectively avoids the resonance length of the pipe, significantly reduces the stress and vibration of the pipe. The field secondary inlet pipe of the reciprocating compressor after transformation doesn't lead severe vibration and break accident again. It confirms the feasibility of theory study.**

***Index Terms*—reciprocating compressor, piping vibration, modal analysis, natural frequency, exciting force**

I. INTRODUCTION

Reciprocating compressor, as a kind of universal equipment, already is widely used in petroleum, chemical, and other fields, the pipe vibration is an important aspect of the compressor reliability design, determines the overall performance of the compressor. In actual production application, the pipe vibration will do great harm to production safety: On the one hand, it will reduce the volumetric efficiency of the compressor, reduce the displacement, and can shorten the service life of the valve and control instrument; On the other hand, it can cause loose joints of pipelines and their accessories, moreover a pipeline vibration fatigue and sudden rupture during use. Pipeline vibration lightly leads to media leaks, or explode, the seeds of major production safety accidents [1]-[8].

In this paper, a reciprocating compressor used oilfield associated gas recycling is shown as an example. Using SolidWorks to establish three-dimensional solid modeling of compressor pipes, and accordingly modal analysis through ANSYS Workbench, obtain pipe natural frequency in the case of low-level. Transfer matrix method is used to calculate the air column natural frequency, compared with frequency of the compressor do a simple analysis for the causes of pipe vibration, and thus make structural improvements.

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II. CAUSE ANALYSIS OF PIPE VIBRATION

Generally think that, in addition to the unit itself caused by manufacture and installation of vibration, the main causes of the compressor pipe vibration are as follows. One is pipe mechanical resonance, namely pipe natural frequency in the resonance region of compressor excitation frequency. The other is the pipe column, while any rank innate frequency of gas column coincides with the resonance region of compressor excitation frequency, it will cause the column resonance [2]-[5].

Due to the dynamic balance of the unit performance, pipe resonance, as well as factors such as the flow fluctuation occurs in the gas pipeline, it will produce a considerable periodic exciting force to the pipes, and cause the destruction of the pipeline structure. A secondary stage inlet pipes and the cylinder intake port flange welding is shown in figure 1, this causes pipe rupture because of severe pipe vibration.
The reciprocating compressor is balanced opposed type with two stages of two column distribution, its main parameters are shown in Table 1 below.

Table.1. Reciprocating compressor main parameters

Parameter name	Value
Spindle speed (r/min)	1500
Cylinder characteristic	double-action
Pipe wall thickness (mm)	4
Pipe material	20 steel
Pipe pressure (MPa)	1.5-2.1
Pipe medium	natural gas
Medium density(kg/m3)	2.5
Medium molecular weight	20
Medium temperature (℃)	40

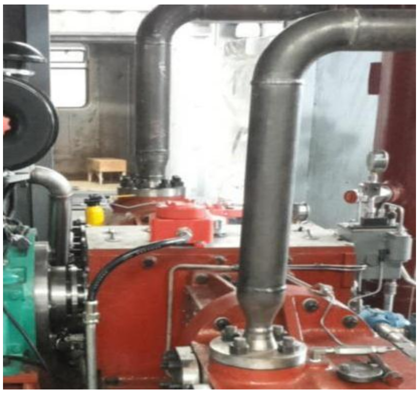


Fig.1. Secondary cylinder intake pipe

Use SolidWorks to establish secondary of the compressor inlet pipe of the three-dimensional entity modeling , as shown in Figure 2. The specification of the pipes is$\Phi 60.3\times4$—$\Phi 76.2\times4$—$\Phi 42.4\times4$, connected by uniform thickness variable diameter tube. Elastic modulus and Poisson's ratio are respectively 206GPa and 0.28.

Fig.2. Three-dimensional solid model of pipe

A. *Mechanical resonance analysis*

When the natural frequency of the pipe is in the resonance region of compressor excitation frequency, it will cause severe vibration of the pipe. The excitation frequency of compressor is calculated according to Eq. (1) as follow:

$$f_e = \frac{Nm}{60} \tag{1}$$

where: N—compressor spindle speed,

m—the order of excitation frequency, while a single-action cylinder is 1, 2 double-action.

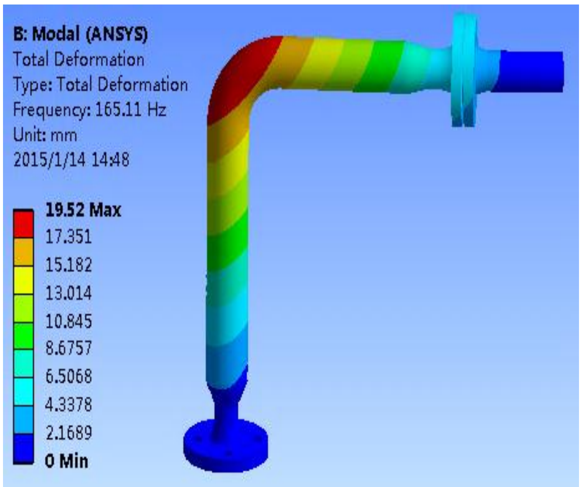
It is generally believed in engineering, the resonance region of compressor excitation frequency is calculated according to $(0.8-1.2)f_e$, and the value is 40-60 Hz.

Introduce the three-dimensional entity model established by SolidWorks into ANSYS Workbench for modal analysis. The first six natural frequencies of pipe are shown in Table 2, the first two modal shapes shown in Figure 3.

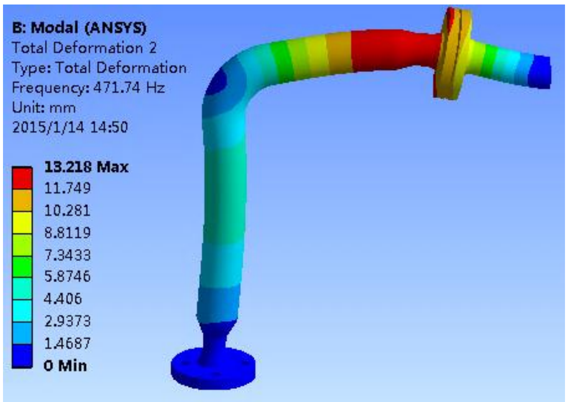
Table.2. The first six order natural frequencies of pipe / Hz

order	1	2	3	4	5	6
frequency	165.11	471.74	508.41	684.99	845.01	1000.77

Accordingly, due to the natural frequency of the pipeline is not in the resonance region of excitation frequency, and its value is far greater than the area, that it will not occur mechanical resonance.



(a) The first order modal shape



(b) The second order modal shape

Fig.3. The first and second pipe mode shape

Through the Response Spectrum module of ANSYS Workbench, the pipe stress distribution under the excitation modal frequency can be shown as clearly as in figure 4. The figure shows the maximum stress appears in the joint of variable diameter tube and flange, pipe rupture in the maximum possibility here because of the diameter of the mutation.

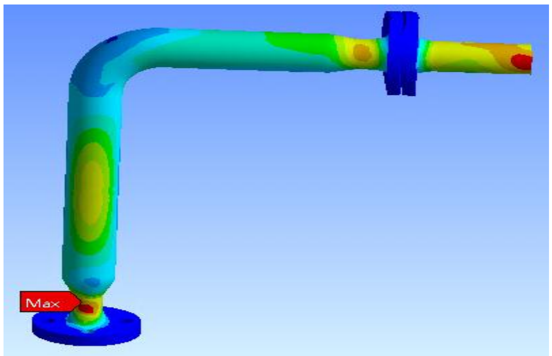


Fig.4. Stress size distribution diagram

B. *Column resonance analysis*

Pulsating air flow in the pipe elbow, variable cross-section, valves and blind pipes can produce larger alternating load—exciting force, may lead to mechanical vibrations of the pipe. In the column close to the compressor end, as a result of intermittent reciprocating compressor suction and discharge of cycles, the column is subjected to periodic exciting force. While any rank innate frequency of gas column coincides with the resonance region of compressor excitation frequency, it will cause the air column resonance, so that pipes have a strong mechanical vibration [2]. In the study, the natural frequency of the air column can be calculated through transfer matrix method, the second inlet pipe diagram as shown in figure 5.

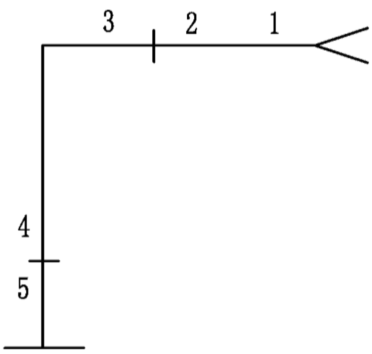


Fig.5. Secondary air inlet pipe diagram

In Figure 5, 1—2 and 3—4 are equal section pipe element, 2—3 and 4—5 are different-diameter pipe element. The transfer matrix of volume element is shown as Eq. (1) [1]-[4].

$$\begin{cases} \begin{pmatrix} p_5 \\ u_5 \end{pmatrix} = M_{4-5} M_{3-4} M_{2-3} M_{1-2} \begin{pmatrix} p_1 \\ u_1 \end{pmatrix} \\ M_{1-2} = \begin{pmatrix} \cos\frac{\omega l_1}{c} & -\rho_0 c \sin\frac{\omega l_1}{c} \\ \frac{1}{\rho_0 c}\sin\frac{\omega l_1}{c} & \cos\frac{\omega l_1}{c} \end{pmatrix} \\ M_{3-4} = \begin{pmatrix} \cos\frac{\omega l_2}{c} & -\rho_0 c \sin\frac{\omega l_2}{c} \\ \frac{1}{\rho_0 c}\sin\frac{\omega l_2}{c} & \cos\frac{\omega l_2}{c} \end{pmatrix} \\ M_{2-3} = \begin{pmatrix} 1 & 0 \\ 0 & \frac{A_2}{A_3} \end{pmatrix}, M_{4-5} = \begin{pmatrix} 1 & 0 \\ 0 & \frac{A_4}{A_5} \end{pmatrix} \end{cases} \quad (2)$$

Endpoint 1 is connected to the second intake separator, regarded opening, $p_1 = 0$, $u_1 = 1$, endpoint 5 cylinder is connected with the second stage cylinder, regarded as closed, $p_5 = 1$, $u_5 = 0$, $A_1 = A_2$, $l_1 = 0.23$ m, $A_3 = A_4$, $l_2 = 1.07$ m, so there is:

$$\begin{pmatrix} 1 \\ 0 \end{pmatrix} = M_{4\text{-}5} M_{3\text{-}4} M_{2\text{-}3} M_{1\text{-}2} \begin{pmatrix} 0 \\ 1 \end{pmatrix} \quad (3)$$

Equation (3) can be expanded:

$$\cos\left(\frac{\omega l_1}{c} + \frac{\omega l_2}{c}\right) = 0 \quad (4)$$

Equation (4) is the column natural frequency equation of the second stage compressor intake pipe. When n is a positive integer, there is:

$$\frac{\omega}{c}(l_1 + l_2) = n\pi - \frac{\pi}{2} \quad (5)$$

Then f the natural frequency of the pipe can be obtained:

$$f = \frac{\omega}{2\pi} = \frac{2n-1}{4} \frac{c}{l_1 + l_2} \; (n = 1, 2, \ldots) \quad (6)$$

where: c—Sound velocity in the gas, $c = \sqrt{kgRT}$,

k—Adiabatic exponent, $k = 1.3$,

g—Acceleration of gravity, $g = 9.8\ \mathrm{m/s^2}$,

R—Gas constant, $R = 20$,

T—Temperature, $T = 313.15$ K,

l—Column length, equal to the length of the pipe.

The first six natural frequencies of pipe column obtained as shown in Table 3.

Table 3. The first six natural frequencies of pipe column

order	1	2	3	4	5	6
frequency	54.32	162.96	271.6	380.24	488.88	597.52

As can be seen from Table 3, the first natural frequency of column falls exactly in the resonance region of excitation frequency, it shows that column resonance would be produced, and column resonance tube length should be avoided when carrying pipe improvements.

C. Calculation exciting force

Gas inside the compressor pipes will create pressure pulsation when passing elbow and variable diameter tube, a greater excitation stress will be caused, thus it may result in rupture of the pipe welding. From figure 5, the airflow from endpoint 1 to the end endpoint pipe 5 of the pipe, endpoint 1 connect with a large container (splitter), and the length of pipe from 1—2 is short enough. It could be regarded as no pulse excitation force when the airflow through the first variable diameter tube. The excitation force pulsation at the bend is shown in figure 6[6]-[8].

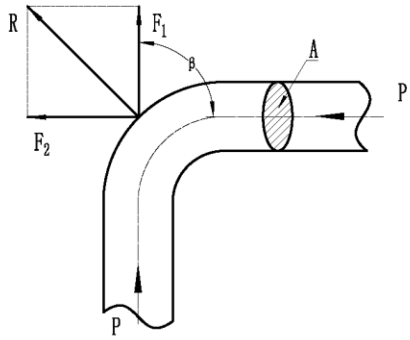


Fig.6. Excitation force pulsation at the bend

The component amplitude of excitation force pulsation at the bend is as follows:

$$\Delta F_1 = \Delta F_2 = \Delta p \cdot A \quad (7)$$

$$\Delta p = \delta \cdot p / 2 \quad (8)$$

$$\delta = \frac{p_{\max} - p_{\min}}{(p_{\max} + p_{\min})/2} \quad (9)$$

The excitation force pulsation at the variable diameter tube is shown in figure 7.

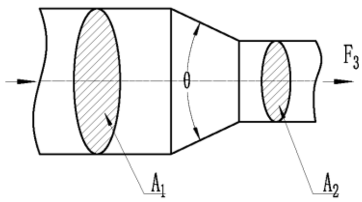


Fig.7. Excitation force pulsation at the variable diameter tube

Reference [6] shows that the component amplitude of excitation force pulsation at the variable diameter tube is as follow:

$$\Delta F_3 = \Delta p(A_1 - A_2) \quad (10)$$

In these vibrations, some may partly offset because of the force direction, so that the pipe vibration is weakened without causing harm. However, in some cases, superposition of exciting force makes the pipe vibration violently [6]. At the variable diameter tube, suppose outer diameter of the small end is D, inner diameter is d, the height from the small end to the bend is h (in the study $h = 450$ mm). As in [7], axial stress at small end of the variable diameter tube is:

$$\sigma_1 = \frac{\Delta F_1 + \Delta F_3}{\pi(D^2 - d^2)/4} \quad (11)$$

Radial stress at small end of the variable diameter tube is:

$$\sigma_2 = \frac{\Delta F_2 h}{W_z} \quad (12)$$

$$W_z = \frac{\pi d^3}{32}\left[1 - \left(\frac{d}{D}\right)^4\right] \quad (13)$$

According to the fourth strength theory, the maximum stress of the small end of the variable diameter tube suffered as follow:

$$\sigma = \sqrt{\sigma_1^2 + 3\sigma_2^2} \tag{14}$$

After calculation, the maximum stress is 376.8MPa. According to the provision of the national standard GB50316-2008, the allowable stress of 20 steel at 40℃ is 130MPa. Obviously the variable diameter tube cannot meet the strength requirements and result in a fracture accident.

III. Pipe construction improvement

Based on the analysis and calculation above, unreasonable design of the inlet pipe is the main cause of pipe rupture, causing the pipe and gas column resonance, and leading to excessive pulse exciting force. To solve this problem, the following measures should be taken:

1) Adding an intake surge tank, at the entrance of second stage cylinder. According to API618 regulations, surge tank is a pulsation suppression device, increases damping and effectively cushions system pressure fluctuations.
2) Using non-standard flange at the entrance of second stage cylinder, increasing the outer diameter to DN65 (Φ76.2 mm) and removing the variable diameter tube.

The three-dimensional solid model of the secondary intake pipe improved shown in Figure 8. At this time, the length secondary intake pipe decreased from 1.3 m becomes 0.98m, the resonance length calculated by Eq. (6) corresponding to the first-order excitation frequency is 1.13-1.7m, effectively avoid the resonance length of pipe. The natural frequency of improved pipe is shown in table 4. Now, the natural frequency of the pipe is not in the resonance region of excitation frequency, will not cause the mechanical resonance. The first order modal shape of improved pipe is shown in Figure 9. The maximum deformation of the pipe decreases more significantly than before.

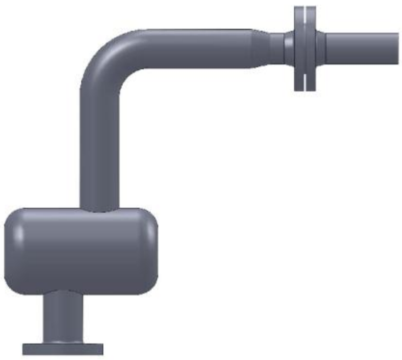


Fig.8. Three-dimensional solid model of the improved pipe

Table.4. The first six order natural frequencies of improved pipe / Hz

order	1	2	3	4	5	6
frequency	77	425.61	427.44	487.92	545.45	690.61

By adding an intake surge tank and removing the variable diameter tube, the exciting force and vibration of the pipe triggered by airflow pulsation are both reduced, and the pipe stress is also significantly decreased. The maximum stress of improved pipe is only 76.55MPa, which is far less than the allowable of pipe stress 130MPa. After rectification, the vibration of pipe is decreased obviously and there are no related accidents. It confirms the correctness of the theoretical research.

IV. Conclusions

1) The pipe resonance and column resonance can cause severe vibration of compressor pipes. By changing the length of pipe, adding a surge tank and other measures to avoid the pipe column resonance caused by the excitation frequency of compressor.
2) Because of the intermittent suction and exhaust of the reciprocating compressor, the pressure and airflow rate will change periodically. When the pulse airflow passes through the elbow, variable diameter tube and other parts, it will make a great exciting force, makes the rhythmic vibration of pipe.
3) By shortening the length of the pipe, reducing the pressure unevenness, reducing the number of elbows and variable diameter tubes, the airflow pulsation caused by vibration force can decrease obviously, and it also can reduce the accident rate of the pipe.
4) When laying pipes, reducing diameter of mutation and avoiding stress concentration as far as possible. Using larger diameter pipe to increase the cross sectional area of pipe, reduce pipe stress, but also through other methods such as increasing the pipe supports to reduce pipe vibration and enhance the stability of pipe.

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