Calculation of Gas Pulsation in Large Reciprocating Compressor Piping Systems: Postprint

Authors: Xiong Yijun (1); Zhang Xiaoqing (1); Zhang Dong (1); Hou Xiaobing (2); Ye Junchao (2)

Date: 2018-01-05T00:00:00+00:00

Abstract

For a large-scale reciprocating piston compressor piping system structure, calculations and analysis of gas flow pulsation were performed. Based on plane wave theory and utilizing the network transfer matrix method, a network analysis model for gas flow pulsation in this complex piping system was established, and the corresponding analysis software program code was developed. The first ten natural frequencies of the gas column in the piping system were computed, and resonance conditions of the piping system were evaluated. Additionally, pulsation pressures at various nodes within the piping system were calculated, with the results satisfying the requirements of API 618 standard. Research on gas flow pulsation in complex piping systems will provide a theoretical foundation for piping system design and for the suppression and elimination of both gas flow pulsation and piping vibration.

Full Text

Calculation of Gas Pulsation in Large Reciprocating Compressor Piping Systems

XIONG Yi-Jun[1], ZHANG Xiao-Qing[1], ZHANG Dong[1], HOU Xiao-Bing[2], YE Jun-Chao[2] [1]School of Energy and Power Engineering, Huazhong University of Science and Technology, Wuhan, 430074, China
[2]Sinopec Petroleum Machinery Co., Ltd. Compressor Branch, Wuhan, 430041, China

Abstract: This paper presents the calculation and analysis of gas pulsation for a large reciprocating compressor piping system. Based on plane wave theory, a network analytical model for gas pulsation in this complex piping system was established using the network transfer matrix method, and the corresponding analytical software code was developed. The first ten orders of gas column natural frequencies were calculated to assess potential resonance in the piping system. Simultaneously, the pulsating pressure at each node of the pipeline system was computed, with results satisfying the requirements of API 618 standard. This research on gas pulsation in complex piping systems provides a theoretical basis for piping system design and the control and elimination of gas pulsation and pipeline vibration.

Keywords: reciprocating compressor; plane wave theory; network transfer matrix; gas natural frequency; pulsating pressure

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Large reciprocating piston compressors serve as power sources for compressing and transporting media, widely used in petroleum, chemical, refrigeration, and other industries. Due to transportation size, weight, and footprint constraints, demands for structural compactness, reliability, and operational stability of large reciprocating compressors are increasingly stringent. However, the intermittent suction and discharge processes of compressors generate strong gas pulsation within pipelines, which not only affects compressor performance but also induces pipeline vibration, posing significant threats to production safety. Therefore, investigating the generation mechanism of gas pulsation and calculating the acoustic field distribution of flow pulsation hold important engineering practical value and practical significance for actively controlling and suppressing gas pulsation in compressor units, reducing or eliminating pipeline vibration, and improving operational economy.

Recent domestic and international research on gas pulsation has primarily employed the transfer matrix method [1], finite element method [2], acoustic-electric analogy method [3], CFD method [4], one-dimensional time-domain flow solution method [5], and commercial software Bentley PULS. PULS is specialized foreign software for gas column analysis in pipelines, currently monopolizing the international market. It is not only expensive to purchase but also costly to update, making the development of independent gas pulsation analysis software extremely significant. The transfer matrix method, also known as the network transfer matrix method, is currently the most practical calculation method for compressor pipeline gas pulsation suitable for engineering design and analysis. Xi' an Jiaotong University has made major contributions in this area [1], particularly in calculating gas column natural frequencies, gas pulsation, and pressure pulsation, successfully solving some practical problems. Li Zhibo [6] used the transfer matrix method to obtain the natural frequencies of arbitrarily complex piping systems and the pulsation waveforms and amplitudes at any location within the piping system. Comparison between calculated results and measured values demonstrated high accuracy of this method. Xu Bin et al. conducted in-depth and detailed research on inter-stage pipeline gas pulsation [7], modified the model to address insufficient calculation accuracy of the transfer matrix method [8], and performed actual measurements, providing highly instructive results. Fan Gen [9] calculated gas column natural frequencies based on the transfer matrix method, studied control methods for pipeline gas pulsation, effectively reduced vibration in compressor unit outlet pipelines and corresponding accessories, and ensured safe and stable operation of the unit, which is of great significance.

This paper, based on plane wave theory and employing the network transfer matrix method, developed a gas pulsation analysis program to analyze and calculate the gas pulsation in the pipeline system of a large reciprocating compressor from an enterprise. The natural frequencies of the gas column were obtained, resonance conditions were assessed, and the pulsating pressure at each node of network components was calculated. The results were compared with API 618 standard to evaluate the rationality of the piping system design, providing theoretical basis and improvement direction for compressor unit piping system design and vibration suppression.

1 Mathematical Model

Research on gas pulsation primarily consists of two parts: calculating the natural frequencies of gas columns to avoid resonance bands of compressor excitation frequencies (0.8~1.2)f, and calculating gas pulsating pressure to control it within standard limits. This paper uses the network transfer matrix method to calculate both gas column natural frequencies and gas pulsating pressure.

1.1 Mathematical Model for Gas Column Natural Frequency Calculation

Under normal conditions, the pulsation value of gas pressure in pipelines is a very small number relative to the average pressure (generally within 8% based on double amplitude), which satisfies the assumptions of plane wave theory. Therefore, gas pulsation in reciprocating compressor pipelines can be analyzed using plane wave theory.

Based on the gas continuity equation, motion equation, and wave equation, and neglecting the average gas flow velocity, the plane wave equation can be obtained [1]:

$$\frac{\partial^2 p}{\partial t^2} = a^2 \frac{\partial^2 p}{\partial x^2}$$

When the average gas flow velocity is u_0, according to the motion equation and wave equation:

$$\rho_0 \frac{\partial u}{\partial t} + \frac{\partial p}{\partial x} = 0$$

$$\frac{\partial^2 p}{\partial t^2} + 2u_0 \frac{\partial^2 p}{\partial x \partial t} = a^2 \frac{\partial^2 p}{\partial x^2}$$

where p is pulsating pressure, Pa; u is pulsating velocity, m/s; ρ_0 is gas density, kg/m³; a is gas sound speed, m/s; C_1, C_2 are complex constants determined by pipeline endpoint conditions.

A complex pipeline system can be considered as composed of basic pipeline elements such as constant-section pipes, containers, confluence points, and reducers. According to network transmission matrix theory, the relationship between pulsating pressure and pulsating velocity at nodes between components can be established. Assuming the pulsating pressure and velocity at the start and end of the compressor pipeline system are p_1, u_1, p_n, u_n respectively, the transmission matrix equation is:

$$\begin{bmatrix} p_1 \\ u_1 \end{bmatrix} = [T_1][T_2]\cdots[T_n]\begin{bmatrix} p_n \\ u_n \end{bmatrix}$$

where $[T_i]$ is the transmission matrix of each pipeline element. Based on boundary conditions, the natural frequency equation of the gas column in the piping system can be obtained, and the natural frequencies of the pipeline system can be calculated using MATLAB programming.

1.2 Mathematical Model for Pulsating Pressure Calculation

To calculate pressure pulsation magnitude, damping must be considered. The linear damping wave equation is:

$$\frac{\partial^2 p}{\partial t^2} + 2u_0\frac{\partial^2 p}{\partial x \partial t} + 2\alpha\frac{\partial p}{\partial t} = a^2\frac{\partial^2 p}{\partial x^2}$$

where α is the pipeline damping coefficient, expressed as $\alpha = \frac{\lambda u_0}{2D}$; λ is the friction coefficient between gas and pipe wall; D is pipe inner diameter, m; u_0 is average gas flow velocity in the pipe, m/s.

The solution to equation (7) is:

$$p(x,t) = e^{-\alpha t}[C_1 e^{j(\omega t - kx)} + C_2 e^{j(\omega t + kx)}]$$

where ω is pulsating angular frequency; $k = \frac{\omega}{a}$ is wave number.

First, harmonic analysis is performed on the compressor' s suction and discharge curves to obtain pulsating mass flow rates at each order. Then, based on the transmission matrices of each element, the total transmission matrix of the piping system is obtained to calculate pulsating pressure at each order. Finally, pulsating pressures at all orders are superimposed to obtain the pulsating pressure at any point in the piping system. The network transfer matrices of general elements can be found in reference [1].

2 Case Study and Calculation

2.1 Network Calculation Model

The compressor cylinder arrangement is symmetrically balanced, with 6 columns and 6 cylinders, double-acting. The medium in the piping system is mixed process gas with an average molecular weight of 17.3 g/mol. The compressor speed is 994 r/min, suction pressure is 1.6 MPa, suction temperature is 16°C, and discharge temperature is 93.9°C.

For calculation and modeling, the following assumptions were made for the compressor unit discharge pipeline system: 1. Temperature and pressure measurement devices in the system are neglected; 2. Valves are treated as variable cross-sections, while filters, scrubbers, and buffer tanks are treated as volume elements; 3. The condenser is simplified into five components: two front-end headers, a rear-end header, and two connecting pipes. The front and rear headers are treated as volume elements, and connecting pipes as constant-section pipes; 4. Flanges are not considered, and normally open valves are treated as straight pipes; 5. Drainage systems and vent pipelines are not considered for their effect on the gas column.

Based on these assumptions, the actual pipeline system was simplified. The simplified result is shown in Figure 1 [Figure 1: see original paper], which represents the network impedance model for analyzing gas pulsation in the piping system.

Figure 1. Network model of pipeline system

In the gas pulsation analysis, the pipeline system is divided into 45 elements and 46 nodes. Nodes 1, 3, 5, 12, 14, and 16 are the exhaust ports of six cylinders respectively. Node 40 is the acoustic open boundary condition (process gas outlet), and node 46 is the acoustic closed boundary condition (nitrogen replacement port; the valve on this branch pipeline is normally closed).

2.2 Gas Column Natural Frequency Calculation

For an acoustically open end, pulsating pressure $p = 0$; for a closed end, pulsating velocity $u = 0$. The endpoint with non-zero p or u is set to 1. The end connected to the cylinder is treated as a closed end. The region between 0.8~1.2 times the gas column natural frequency is defined as the resonance zone.

Based on the transmission matrix method, the natural frequencies of the gas column in the discharge pipeline system were calculated. The first ten natural frequencies f and resonance regions $(0.8\ 1.2)f$ are shown in Table 1 .

Table 1. Gas natural frequency and resonance region

Order	f / Hz	$(0.8\ 1.2)f$ / Hz
1	0.69	0.56~0.83

Order	f / Hz	(0.8 1.2)f / Hz
2	2.36	1.89~2.83
3	3.10	2.48~3.72
4	5.16	4.13~6.19
5	7.94	6.36~9.53
6	9.93	7.95~11.92
7	12.62	10.10~15.15
8	15.38	12.30~18.46
9	17.83	14.26~21.39
10	19.16	15.33~23.00

The compressor excitation frequency f_{ex} is calculated by equation (9):

$$f_{ex} = \frac{n \cdot m}{60}$$

where n is compressor speed, r/min; m equals 1 for single-acting and 2 for double-acting.

The calculated main excitation frequency of the compressor is 33.13 Hz. The results show that the compressor excitation frequency effectively avoids the first ten resonance zones of the gas column, indicating that resonance between the compressor and gas column in the pipeline will not occur.

2.3 Pressure Pulsation Calculation

2.3.1 Pressure Pulsation Control Criteria The degree of pressure pulsation is evaluated using pressure non-uniformity δ:

$$\delta = \frac{p_{max} - p_{min}}{p_m} \times 100\%$$

where p_{max} is maximum instantaneous gas pressure, p_{min} is minimum instantaneous gas pressure, p_a is maximum pulsating pressure amplitude, and p_m is average gas pressure.

According to the fifth edition of API 618 released in 2007, the pressure pulsation control criteria at various locations in the system are summarized as follows [10]:

(a) Maximum allowable pressure pulsation at compressor cylinder flanges

The unfiltered pressure pulsation non-uniformity at compressor cylinder flanges should be limited to below 7% or less than the value calculated by the following formula:

$$[\delta] = \frac{0.03 \cdot p_m}{(p_d/p_s) - 1}$$

where p_d/p_s is stage pressure ratio.

(b) Maximum allowable pulsation range at buffer device pipe side connections or beyond

Buffer devices should limit the pressure non-uniformity at the pipe side of the buffer device within the range of equation (12):

$$\delta = \frac{35 \cdot a}{P_m \cdot D \cdot f}$$

where a is gas sound speed, m/s; P_m is average absolute pressure in pipeline, MPa; D is pipe inner diameter, mm; f is pulsation frequency, Hz.

2.3.2 Harmonic Analysis of Pulsating Velocity Nodes 1, 3, 5, 12, 14, and 16 are boundary conditions at compressor exhaust ports with known pulsating velocity. First, Fourier harmonic analysis was performed on pulsating velocity and pulsating mass flow rate. The first 10 harmonics of pulsating velocity and mass flow rate at node 1 are shown in Table 2 . Although the harmonic coefficients of pulsating velocity at other exhaust nodes differ, their amplitudes are the same as at node 1 and are not listed here. For double-acting cylinders, the harmonic with maximum amplitude is the 2nd order, whose frequency is the main excitation frequency.

Table 2. The first 10 order of harmonics for the pulsating velocity and mass flow rate at node 1

Order	Pulsating Velocity Component (Real) $/\mathrm{m}\cdot\mathrm{s}^{1}$	Pulsating Velocity Component (Imag) $/\mathrm{m}\cdot\mathrm{s}^{1}$	Pulsating Velocity Amplitude $/\mathrm{m}\cdot\mathrm{s}^{1}$	Mass Flow Rate Component (Real) $/\mathrm{kg}\cdot\mathrm{s}^{1}$	Mass Flow Rate Component (Imag) $/\mathrm{kg}\cdot\mathrm{s}^{1}$	Mass Flow Rate Amplitude $/\mathrm{kg}\cdot\mathrm{s}^{1}$
1	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
2	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
3	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
4	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
5	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
6	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
7	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
8	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
9	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000
10	0.0000	0.0000	0.0000	0.0000	0.0000	0.0000

Order	Pulsating Velocity Component (Real) /m · s^{1}	Pulsating Velocity Component (Imag) /m · s^{1}	Pulsating Velocity Amplitude /m · s^{1}	Mass Flow Rate Component (Real) /kg · s^{1}	Mass Flow Rate Component (Imag) /kg · s^{1}	Mass Flow Rate Amplitude /kg · s^{1}

Figure 2 [Figure 2: see original paper] shows the pulsating velocity curve within one cycle. The curve conforms to the exhaust velocity pattern of double-acting cylinders. Additionally, the pulsating velocity amplitudes at the shaft side and head side of the compressor are asymmetric due to the piston rod occupying certain cylinder volume at the shaft side, resulting in structural asymmetry.

Figure 2. Pulsating velocity curve

2.3.3 Calculation of Pressure Non-Uniformity Based on the element transmission matrices of each component, the pulsating pressure harmonics at each point can be obtained. Due to the large number of nodes, only the pressure amplitudes at compressor exhaust nodes and buffer tank outlet are listed here, as shown in Table 3 and Table 4 .

Table 3. Pressure amplitude of node 1 at exhaust port

Order	Re/Pa	Im/Pa	Pm/Pa	d/rad
1	0.00	0.00	0.00	0.00
2	0.00	0.00	0.00	0.00
3	0.00	0.00	0.00	0.00
4	0.00	0.00	0.00	0.00
5	0.00	0.00	0.00	0.00
6	0.00	0.00	0.00	0.00
7	0.00	0.00	0.00	0.00
8	0.00	0.00	0.00	0.00
9	0.00	0.00	0.00	0.00
10	0.00	0.00	0.00	0.00

Note: Re = real part of solution; Im = imaginary part of solution; Pm = pressure amplitude; d = phase angle.

Superimposing the pulsating pressure amplitudes at all orders according to their phase relationships yields the total pressure waveform at node 1, shown in Figure 3 [Figure 3: see original paper]. The maximum pulsating pressure is 81,700 Pa, and the minimum is -102,910 Pa, occurring at rotation angles of 2.26 rad and 5.82 rad respectively. With an average internal pressure of 6 MPa, the pressure non-uniformity is 3.08%.

Figure 3. Pressure waveform of node 1 at exhaust port

Table 4. Pressure amplitude of node 7 at the outlet of the buffer

Order	Re/Pa	Im/Pa	Pm/Pa	d/rad
1	0.00	0.00	0.00	0.00
2	0.00	0.00	0.00	0.00
3	0.00	0.00	0.00	0.00
4	0.00	0.00	0.00	0.00
5	0.00	0.00	0.00	0.00
6	0.00	0.00	0.00	0.00
7	0.00	0.00	0.00	0.00
8	0.00	0.00	0.00	0.00
9	0.00	0.00	0.00	0.00
10	0.00	0.00	0.00	0.00

Note: Re = real part of solution; Im = imaginary part of solution; Pm = pressure amplitude; d = phase angle.

Superimposing the pulsating pressure amplitudes at all orders according to their phase relationships yields the total pressure waveform at node 7, shown in Figure 4 [Figure 4: see original paper]. The maximum pulsating pressure is 1,564.6 Pa, and the minimum is -1,200 Pa, occurring at rotation angles of 0.71 rad and 5.74 rad respectively. With an average internal pressure of 6 MPa, the pressure non-uniformity is 0.046%.

Figure 4. Pressure waveform of node 7 at the outlet of the buffer

Similarly, the maximum and minimum pulsating pressures at each point in the pipeline can be obtained, and the pressure non-uniformity can be calculated and compared with API 618 standard, as shown in Table 5 .

Table 5. Pressure non-uniformity of each node

Node	Location	p^*_{max} /Pa	p^*_{min} /Pa	p_m /MPa	δ /%	API 618 Allowable Value /%
1	Cylinder exhaust port	81700	-102910	6	3.08	7.00
3	Cylinder exhaust port	81700	-102910	6	3.08	7.00

Node	Location	p^*_{max} /Pa	p^*_{min} /Pa	p_m /MPa	δ /%	API 618 Allowable Value /%
5	Cylinder exhaust port	81700	-102910	6	3.08	7.00
7	Buffer outlet	1564.6	-1200	6	0.046	0.51
12	Cylinder exhaust port	81700	-102910	6	3.08	7.00
14	Cylinder exhaust port	81700	-102910	6	3.08	7.00
16	Cylinder exhaust port	81700	-102910	6	3.08	7.00
40	Buffer outlet	1564.6	-1200	6	0.046	0.51

Note: p^*_{max} = maximum pulsating pressure amplitude; p^*_{min} = minimum pulsating pressure amplitude; p_m = average pulsating pressure; δ = pressure non-uniformity; $[\delta]$ = API 618 allowable value.

The results show that after passing through the buffer tank, the pressure non-uniformity is far below the API 618 allowable value, indicating that the pressure non-uniformity in subsequent pipelines will also certainly meet standard requirements.

3 Conclusion

Based on plane wave theory, this paper established mathematical models and network transfer analysis models for gas pulsation analysis in complex piping systems of large reciprocating piston compressors, and developed a gas pulsation analysis program using MATLAB. The program was applied to analyze gas pulsation in the outlet pipeline system of a large reciprocating compressor from an enterprise, obtaining calculation results for gas column natural frequencies and pulsating pressures, and evaluating the rationality of the pipeline system design.

The research results indicate that the first ten orders of gas column natural frequencies in this pipeline system range from 0.7 to 19.16 Hz, which effectively avoids the compressor excitation frequency of 33.13 Hz, demonstrating that low-order gas column resonance will not occur in this system. Additionally, the pressure non-uniformity at each compressor exhaust port of the pipeline system is 3.08%, which is below the API 618 standard allowable value of 7%, and the pressure non-uniformity at the buffer tank outlet is 0.046%, far below the API 618 standard allowable value of 0.51%. This indicates that the pipeline system design is rational and will not cause strong pipeline vibration.

Through the developed gas pulsation analysis software, the natural frequencies of gas columns and the pulsating pressure and pressure non-uniformity at each node can be obtained, enabling effective prediction of gas pulsation in compressor pipeline systems and providing theoretical basis and improvement direction for piping system design and gas pulsation suppression, ensuring safe and efficient operation of the unit.

Acknowledgments: This research was funded by the National High-tech Ship Research Program (Project No. 0214120042).

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Note: Figure translations are in progress. See original paper for figures.