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Lanlan Yang
Xi'an Jiaotong University, China, People's Republic of, aranya@stu.xjtu.edu.cn

Wenkai Wang
Xi'an Jiaotong University, China, People's Republic of, wangwenkai@stu.xjtu.edu.cn

Xiaohan Jia
jiaxiaohan@mail.xjtu.edu.cn

Xueyuan Peng
Xi'an JiaoTong University, China, People's Republic of, xypeng@mail.xjtu.edu.cn

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Pressure Pulsation in the Reciprocating Compressor with Stepless Capacity Regulation

Lanlan Yang¹, Wenkai Wang¹, Xiaohan Jia¹*, Xueyuan Peng¹

¹ No.28 Xianning West Road Xi'an Jiaotong University, Xi'an, Shaanxi, China
jiaxiaohan@stu.xjtu.edu.cn

ABSTRACT

The stepless capacity regulation by means of delayed closure of the suction valves plays an important role in energy conservation for the reciprocating compressor. Serious gas pulsation usually occurs during the capacity regulation, and its analysis and suppression is a challenge due to the varied components of the pulsation excitation under different conditions. This paper presents a numerical study and experimental verification of the gas pulsation in the compressor piping with stepless capacity regulation. An approach was proposed to calculate the pulsation excitation in the case of stepless capacity regulation, which was further incorporated into the frequency-domain modeling of the gas pulsation based on the plane wave theory. A test rig was built up and an experimental verification was conducted. The amplitudes as well as waves of the pressure pulsation in positions of suction and discharge valve chambers were measured as the compressor capacity was regulated by changing the closing time of the suction valves. The test data were compared with the simulated results, which indicated a good agreement. The validated model was applied to conduct the parametric study. The results showed that the gas pulsation of main excitation frequency increased in the suction pipeline system, and decreased in the discharge pipeline as the opening time of the suction valve increased.

1. INTRODUCTION

The capacity control method by delayed closure the suction valves is often used for large reciprocating compressor in petrochemical and petroleum industry. An analogy method was proposed to analyze the capacity control regulation for large reciprocating compressor by Gu Z L and Hou X P. This method simplified the piston motion as pulse signal and neglected valve motion to study the capacity of reciprocating compressor. Considering the variation of the capacity and the delayed closure time of the suction valve in a compression stroke, an ideal thermal cycle of reciprocating compressor in the case of the stepless capacity regulation by delayed the closure of the suction valve is obtained by W Hong and R Wu. Another thermal cycle for the stepless capacity regulation by delayed the closure of the suction valve was proposed by JIN Jiangming and HONG Weirong; this thermal cycle considered the effect of the real gas and used the L-K equation to deduce the thermodynamics cycle. In addition, a device based on a novel rotary control valve and an adaptive regulation system for reciprocating compressor was proposed and designed by Dacheng Li and Haiqi Wu. This regulation system was able to realize stepless capacity control within the range of 0-100% and energy saving correspondingly with the load condition.

At present, literatures focused on the theory calculation and optimizing the capacity regulation, but there has been little research on the gas pulsation in the reciprocating compressor with the capacity regulation. Frequency-domain using the plane wave theory and time-domain are the two traditional aspects to analyze the gas pulsation. The analysis for pressure pulsation based on the frequency-domain is introduced in detail by Dang XQ and Chen SW. This methodology neglects the valve motion and demands only single flow direction for one model. An optimized method was proposed by J. T. Sanford and S. J. Schoonmaker. This method considered the real valve motion and the interaction between the valve motion and the pressure pulsation in the piping. CFD analyzing the time-domain of the gas pulsation is more accurate and often used for scientific research. A 3d-CFD model with respect to the interaction between valve motion and pressure pulsation was built by Bin Zhao and Xiaohan Jia. But this method is time-consuming and not suitable for engineering filed.

Considering that the traditional frequency-domain method cannot change flow direction on one model, this paper proposed a numerical method to calculate the pulsation oscillation in the case of stepless capacity regulation, which
was incorporated into the frequency-domain modeling based on the plane wave theory. Furthermore, a test rig was built up to verify the numerical method.

2. Numerical Model

2.1 Theoretical Analysis
The stepless capacity regulation by means of delayed closure of the suction valves leads to the reverse flow in the suction pipeline system. The pressure pulsation in the suction pipeline system would be more complicated.

Take a double-acting reciprocating compressor for example. In order to analyze the effect of delayed closure of the suction valves, an assumption was proposed that the cylinder volume rate was proportional to the velocity of piston during the suction or discharge valve opening. The volumetric flow rate versus the crank angle in the suction head cylinder can be described as equation (1).

\[
\begin{align*}
    u_t &= 0 & 0 \leq \omega t \leq \theta_0 \\
    u_t &= \lambda_p \rho \sigma (\sin \omega t + \frac{\lambda}{2} \sin 2\omega t) & \theta_0 \leq \omega t \leq \theta_1 \\
    u_t &= 0 & \theta_1 \leq \omega t \leq 2\pi
\end{align*}
\]

(1)

As illustrated in Figure 1(b), with the capacity of the compressor from 100% decreased to 0, the amplitude of the 2nd time frequency is increased. Therefore, it is important to control the gas pulsation for reciprocating compressor system in capacity regulation by means of delayed closure of the suction valve.

![Figure 1: Gas volume rate and amplitude-frequency characteristics in the suction cylinder](image)

2.2 Numerical Model
Traditional pulsation analysis methods based on the frequency-domain demand single flow direction for one model. Then the whole compressor system is divided into suction piping system and discharge piping system. However, the direction of the gas pulsation is changed during the delayed closure of the suction valves in the suction pipeline. A new numerical method is proposed to calculate the pulsation in the case of stepless capacity regulation, which was complicated frequency-domain modeling based on the plane wave theory.

2.2.1 Suction Piping System
As illustrated in Figure 2, for the suction pipeline system, the suction stroke and reverse flow stroke are the main factors for the gas pulsation. The core idea of this approach is established two different models, in which the piping layout are the same as the actual suction model. The first model is used to calculate the gas pulsation for the full-load state and the second model is used to calculate the gas pulsation for the period of the delayed closure of the suction valves. The parameters of the compressor boundary for the second model are determined by the first model.
and the reverse flow stroke. The principle is to ensure that the fluid exhausted volume in the second model is equal to the reverse volume during the period of delayed closure of the suction valve.

\[ \frac{dP_{d\theta}}{d\theta} = \frac{kP_{d\theta}(-\sin \theta + \frac{\lambda_{d\theta}}{2} \sin 2\theta)}{1 + \cos \theta + \frac{\lambda_{d\theta}}{2} \sin^2 \theta + 2\alpha} \]

The crank radius, speed and connecting rod length of the second model could be evaluated by the core idea.

\[ r_{d\theta} = \frac{\alpha(v_1 - v_2) r_t}{2}, \quad n_{d\theta} = \frac{60}{t_5 - t_3}, \quad l_{d\theta} = \frac{r_{d\theta}}{\lambda_{d\theta}}, \quad \lambda_{d\theta} = \lambda_{s\theta} \]

The flow rate could be evaluated from the above-calculated parameters, as following. The flow rate for the crank-end were assumed equal to the head-end.

\[ Q_{d\theta} = A_l l_{d\theta} \frac{n_{d\theta} EV_{d\theta}}{60}, \quad EV_{d\theta} = 100 - \alpha \frac{P_{d\theta}}{P_{d\theta}^{1/4}} \frac{Z_{d\theta}^{1/4} (\frac{P_{d\theta}^{1/4}}{P_{d\theta}^{1/4}} + 1)}{Z_{d\theta}^{1/4}} \]

After simulating, the pressure pulsation could be calculated through the reverse Flourier, as following.
\[ p_{d*} = \begin{cases} 
    p_0 + \sum_{n=1}^{N} c_{dn} \cos(n \sigma_n t + \varphi) & 0 \leq t \leq T_1/2 \\
    p_0 & T_1/2 \leq t \leq T_1/2 \\
    p_0 + \sum_{n=1}^{N} c_{dn} \cos(n \sigma_n t + \varphi) & T_1/2 \leq t \leq (T_1 + T_2)/2 \\
    p_0 & (T_1 + T_2)/2 \leq t \leq T_1 
\end{cases} \] (5)

2.2.2 Discharge Piping System
The direction of the gas pulsation in the discharge pipeline is single and the discharge stroke is less as the delayed closure in the suction valve. In the discharge model, the crank radius and connecting rod length are given in as equation (6). The flow rate is similar to the second model.

\[
\begin{bmatrix}
    r_d = r_s - r_{2d} \\
    l_d = \frac{r_d}{\lambda_d}
\end{bmatrix}
\] (6)

The traditional pulsation analysis method is used to calculate the pressure pulsation of the discharge pipeline.

2.2.3 Computer Program
The computer program for the reciprocating compressor with stepless capacity regulation by delayed closure of the suction valves was developed to calculate the pressure pulsation of the whole compressor pipeline system. A flow diagram of this numerical methodology was given in Figure 3.
Figure 3: A flow chart for pulsation analysis of compressor system with stepless capacity control

3. Experiment Setup

In order to validate the numerical model, a test rig including the identical compressor system was conducted, and the dynamic pressure inside the suction and discharge chamber were monitored, as shown in Figure4. The geometrical data and the thermodynamic conditions of the compressor were shown in Table.1.

Table 1: Specification of the compressor system

<table>
<thead>
<tr>
<th>Parameters</th>
<th>1st stage</th>
<th>2nd stage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crank radius (mm)</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>Connecting rod length (mm)</td>
<td>500</td>
<td>500</td>
</tr>
<tr>
<td>Speed (rpm)</td>
<td>369</td>
<td>369</td>
</tr>
<tr>
<td>Piston diameter (mm)</td>
<td>400</td>
<td>240</td>
</tr>
<tr>
<td>Clearance volume (%)</td>
<td>16.5</td>
<td>23.5</td>
</tr>
<tr>
<td>Suction/Discharge pressure (kPa)</td>
<td>100/310</td>
<td>309/900</td>
</tr>
<tr>
<td>Suction/Discharge temperature(°C)</td>
<td>30/144</td>
<td>40/150</td>
</tr>
</tbody>
</table>
The gas pulsation in the suction and discharge chambers in the range of capacity between 10% and 100%, with a step of 10%, were tested. Two pressure sensors with the measuring ranging from 0 to 3.5bar were used to monitor the dynamic pressure in the suction and discharge chamber of the first stage respectively. Two pressure sensors with the measuring ranging from 0 to 7bar were used to monitor the dynamic pressure in the suction and discharge chamber of the second stage compression. An additional flowmeter was installed to monitor the outlet flowrate, which can be correlated to the angle of the delayed closure in the suction valve. A data acquisition system was used to simultaneously record the dynamic pressure signal.

4. Results and Discussion

4.1 Pressure pulsation in the suction pipeline
As illustrated in the Figure 5. The pressure pulsation of the main excitation frequency obtained by test rig and simulation model is present. It can be seen that the relative error less than 13.2% and 13.7% in the node p1 and p3 respectively. The results indicates the validity of the numerical model. The deviation between the simulation and experiment is caused by the algorithm. The gas pulsation in the main excitation frequency decreases with the flow rate nearly corresponded with the movement of piston. While the flow rate is 20% of the full-load condition, the pressure pulsation in the main excitation frequency increases about 58% and 82% in the node p2 and p4. The capacity regulation caused more serous effect in the 2s pipeline system.
Figure 5: Amplitude-frequency characteristics and the amplitude-2\textsuperscript{nd} time frequency characteristics in the suction pipeline system: (a) node p1, (b) node p3

4.2 Pressure pulsation in the discharge pipeline

Figure 6 illustrates the amplitude-frequency characteristics versus the compressor capacity. For the amplitude-2\textsuperscript{nd} time frequency, the simulated results were consistent with the experimental results. It could be observed that the relative error less than 16.8\% and 13.2\% at the node p2 and p4 respectively. The lower pressure pulsation in the main excitation frequency can be attributed to the higher reverse flow in the suction valves. The capacity of compressor from full-load to 10\% part-load led to the lowest pressure pulsation in the main excitation frequency at the node p2 and p4. For discharge piping system with capacity regulation, it suggests that the main focus of attention should be on analyzing the pressure pulsation of full-load.
Figure 6: Amplitude-frequency characteristics and the amplitude-2"nd time frequency characteristics in the discharge pipeline system: (a) node p2, (b) node p4

5. CONCLUSIONS

In this paper, we have presented a more complicated numerical analysis method to analyze the pressure pulsation in the reciprocating compressor system with stepless capacity by delayed closure the suction valves. There was a high level of agreement between the simulated data and the test data. The results suggest that it should be emphasized analysis the pressure pulsation of lowest part-load in the suction pipeline system but the pressure pulsation of full-load in the discharge pipeline system.

NOMENCLATURE

\[ A_p \] Piston area \( (m^2) \)
\[ a \] Part load \( () \)
\[ r \] Radius \( (m) \)
\[ w \] Compressor rotation speed \( (1/s) \)
\[ t \] Time \( (s) \)
\[ \theta_0 \] Angle in the suction valve at suction start \( (rad) \)
\[ \theta_u \] Angle in the suction valve at the end of reverse flow \( (m^3/s) \)
\[ \theta_s \]
Angle in the suction valve at the end of reverse flow 

\[ EV \]
Volumetric volume 

\[ \alpha \]
Clearance volume 

\[ k \]
Ratio of specific heat of gas 

\[ p \]
Pressure 

\[ p' \]
Dynamic pressure 

\[ p_0 \]
Constant pressure 

\[ T \]
Temperature 

\[ n \]
Speed 

\[ \lambda \]
The crankshaft-connecting rod rotation 

\[ Z \]
Compressibility 

\[ Q \]
Average volumetric flow rate 

\[ Q_{\text{max}} \]
Average volumetric flow rate in the full-load condition 

\[ \varphi \]
Angle 

\[ \delta \]
Relative change of pressure pulsation in the main excitation frequency 

\[ c_{\text{sh}} \]
The amplitude of the second model 

\[ \text{Subscript} \]

1 to 5 
State points 

s 
Suction model 

ss 
Suction side of the suction model 

sd 
Discharge side of the suction model 

dv 
The second model 

dvs 
Suction side of the second model 

dvd 
Discharge side of the second model 

d 
Discharge model 

REFERENCES


