Numerical and Experimental Research of Noise Reduction due to Low Frequency Pressure Fluctuation of Rotary Compressor

Chaole Zhang
zcl@pudut.com

Zhiqiang Deng

Hui Shen

Lichang Xie

Haishui Jin

Numerical and Experimental Research of Noise Reduction due to Low Frequency Pressure Fluctuation of Rotary Compressor

Chaole Zhang[1]*, Zhiqiang Deng[1], Hui Shen[2], Lichang Xie[2], Haishui Jin[1]

[1] Shanghai PuDu Information Technology Co., LTD,
Shanghai, China
NO.19 Jiangfa Road, Pudong new district, Shanghai, China

[2] Zhuhai Landa Compressor Co., LTD,
Zhuhai, Guangdong, China
No. 1, Longshan Avenue, Zhuhai, Guangdong, 519100, China

* Corresponding Author, email: zcl@pudut.com

ABSTRACT

The mechanism of the low frequency pressure pulsation at around 173Hz and noise of a typical rotary compressor were analyzed in this work. According to the past experiences and the test results of the rotary compressor, the rotational flow caused by the balance block of the shaft in the top chamber should be the major reason of the low frequency pressure pulsation at the frequency. In order to improve the pressure pulsation problem of the rotary compressor, an effective solution was proposed in this work, which adds a guide pipe at the outlet of bearing hole in the rotary compressor in order to reduce the influence caused by the balance block and to improve the pressure pulsation problem. Numerical and experimental methods were used to validate the effectiveness of the proposed solution. According to the pressure pulsation and noise test results, the related low frequency pressure pulsation at around 173Hz and the noise level at the frequency were improved obviously as compared with the original rotary compressor. The work conducted in this paper can provide a reference for the similar pressure pulsation and noise problem in rotary compressors.

1. INTRODUCTION

The low frequency pressure pulsation and the correlated noise problem is the major research focus in the field of rotary compressor. The mechanism of the low frequency pressure pulsation was researched by lots of researchers (Istvan and Beranek, 2006).

Generally, the research mainly focuses on the transitive relationship of pressure and velocity in the different compartments of the compressor, and the calculation accuracy improvement and the methods to predict the pressure pulsation in the compressors (Cyklis, 2001; Jeongil *et al.*, 2008).

As for the numerical simulation used to study the pressure pulsation problem in the compressor, traditional CFD method was used to predict and to monitor the pressure pulsation at the area of concern. Han and his colleagues studied the applicability of different turbulence models in the numerical simulation for the gas pulsation in the reciprocating compressor piping system. Based on the research results, two equations k-epsilon turbulence model is a better and cheaper choice for the simulation of the gas pulsation in the piping system as compared with other high accuracy turbulence models (Han *et al*. 2013).

Based on the past experiences and associated research reports in rotary compressor manufacture, even a small change in the structure can improved the performance and noise level of the compressor significantly. For example, Keith's researches show that it is important to include the cylinder bore volume in the flow path analysis in order to accurately calculate pressure pulsations in the hermetic reciprocating compressor (Keith, 2014).

Therefore, in this work, with some kind of rotary compressor with a worse low frequency pressure pulsation problem at around 173Hz as the research object, the mechanism of the pressure pulsation was analyzed, and effective solution was proposed and verified in numerical and experimental methods. As a consequence, the noise level of the rotary compressor was improved too.

2. PROBLEM DESCRIPTION

There are some kind of low frequency buzz during the operation of a typical rotary compressor. In order to resolve the problem associated with the rotary compressor, a pressure pulsation test was conducted and the test fixture used in this work is shown in Figure 1. And according to the test result, the typical peak values of the pressure pulsation at the low frequency band of 173Hz and other low frequency bands are listed in Table 1. According to the test result, the pressure pulsation at the third resonance frequency (173Hz) has a higher peak value in the bottom chamber as compared with the peak value in the top chamber for this rotary compressor.

Figure 1: Pressure pulsation test fixture of rotary compressor

Table 1: Typical pressure pulsation peak values and associated frequency

Position	**57.5 Hz**	**115.0 Hz**	**173.0 Hz**	**230.5 Hz**	**288.0 Hz**
Top chamber / Pa	5896.3	3858.9	1114.3	460.3	74.0
Bottom chamber / Pa	3548.6	2285.9	2587.6	2038.7	532.7

3. MECHANISM ANALYSIS AND SOLUTION

Based on the pressure pulsation test results and past experiences for this type rotary compressor, the low frequency pressure pulsation peak value at the bottom chamber should be caused by the rotation of refrigerant pushed by the balance block in the rotary compressor. In order to suppress the low frequency pressure pulsation, it is necessary to find a way to weaken the influence of the rotation of refrigerant due to the balance block on the pressure pulsation. Therefore, based on the structure of the rotary compressor, a solution by adding a guide pipe at the top of the shaft was proposed, as shown in Figure 2.

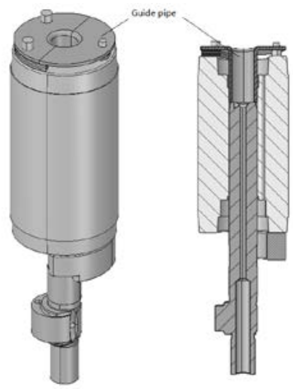


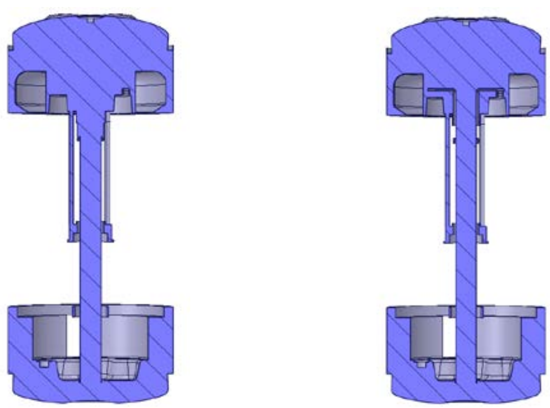
Figure 2: Proposed solution for the rotary compressor

4. VALIDATION AND RESULT ANALYSIS

The solution for the current rotary compressor proposed in this work was validated by using numerical and simulation methods respectively.

4.1 Numerical Validation

4.1.1 Numerical analysis models:
The computational models for the original and current models are shown in Figure 3. The total grid node for the original and the current model is about 2.5M.



(a) Original (b) Current

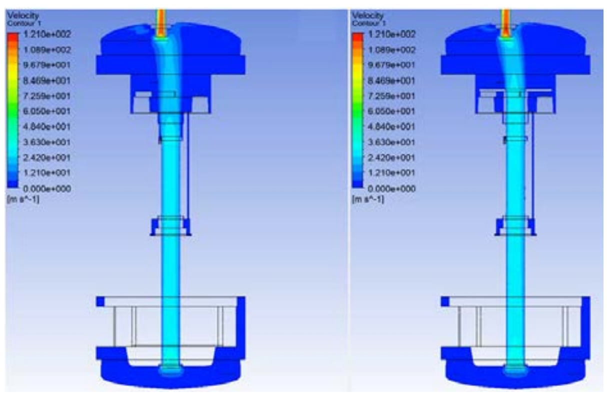
Figure 3: Computational models for original and current configurations

As for the original (Original) and the improved rotary compressor (referred to as Current later), an unsteady CFD computational model were established in this work. An unsteady analysis method was used in this work and the compressible real-gas medium was used. The Numerical analysis was conducted using commercial software Ansys Fluent.

In the fluid region, the steady compressible Reynolds-Averaged Navier-Stokes (RANS) equations are solved. The two equation Realizable k−epsilon turbulent model are used as turbulence closure. The scalable wall function is adopted. The PRESTO! Pressure discretization scheme is used because this scheme can avoid the interpolation errors and pressure gradient assumptions on the boundary. The second-order upwind scheme is used to discretize the convection terms, and the second-order central difference scheme is used to discretize the diffusion terms. The SIMPLEC scheme is used to solve the pressure-velocity coupling. The central rotational parts corresponding to the shaft and other accessories was treated by using sliding mesh method. For the bottom region of the computational models, an inlet boundary condition was setup in order to simulate the input pressure pulsation. And the outlet pipe of the rotary compressor was extended about 2000mm with a same diameter, and an averaged static pressure boundary condition was given there. For the inlet boundary condition, a specified quarter sine pressure pulsation signal with a 57Hz was given. And averaged static pressure boundary condition was given for the outlet boundary condition. The time step is 1e-5s. And for the simulation, total time steps of 10,000 were iterated in this work. The pressure value of the working medium at the real outlet position of the rotary compressor was monitored during the simulation.

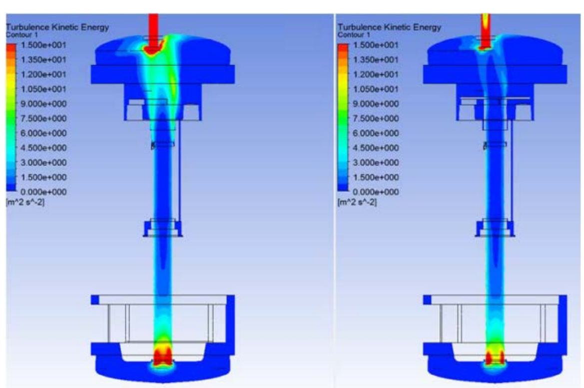
4.1.2 Results and discussion:

According to the simulation results, the comparison of the velocity magnitude distribution in the middle section along the shaft was shown in Figure 4. And the comparison of the turbulence kinetic energy distribution along the shaft was shown in Figure 5. It can be seen from the figures, as compared with the original configuration, the flow situation was improved for the configuration with the added guide pipe.



(a) Original (b) Current

Figure 4: Comparison of velocity magnitude distribution



(a) Original (b) Current

Figure 5: Comparison of turbulence kinetic energy distribution

Based on the monitored pressure pulsation signal at the near the outlet pipe position, an FFT was conducted. And the result was shown in Figure 6. Based on the comparison between the original and the current configurations, at the 3rd harmonic frequency (173Hz), the peak value was reduced by 20% as compared with the original configuration. And the peak value at the other low frequency band is almost same for the two configurations.

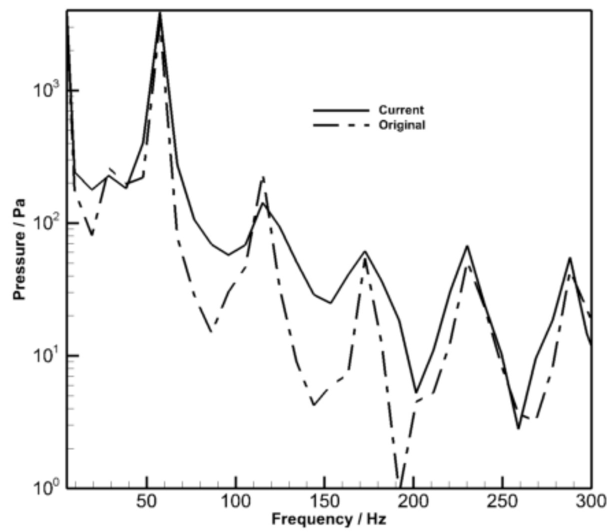


Figure 6: Comparison of monitored pressure signal near the outlet of the rotary compressor

4.2 Experimental Validation

4.2.1 Pressure pulsation test and analysis:

As for the pressure pulsation test, a test fixture was designed and installed on the shell of the rotary compressor as shown in Figure 1. And the test fixture was mainly used to get the pressure pulsation signal of refrigerant in the top and bottom chambers for the current rotary compressor. The pressure sensors were fixed on and protected by the test fixture. And the test fixture can ensure no leakiness for the rotary compressor. Based on the test results, the typical peak values for the low frequency pressure pulsation in the top and the bottom chamber were shown in Table 2 and Table 3.

It can be seen from Table 2, as for the top chamber, same with the simulation result, at the 3rd harmonic frequency (173Hz), the peak value was improved obviously as compared with the original configuration. Also, for the bottom chamber, the pressure pulsation was improved as compared with the original configuration after adding the guide pipe.

Table 2: Comparison of typical peak value of low frequency pressure pulsation in the top chamber

Top chamber	**57.5 Hz**	**115.0 Hz**	**173.0 Hz**	**230.5 Hz**	**288.0 Hz**
Original / Pa	5896.31	3858.92	1114.25	460.33	74.01
Current / Pa	6148.45	3665.89	406.03	71.00	15.32

Table 3: Comparison of typical peak value of low frequency pressure pulsation in the bottom chamber

Bottom chamber	**57.5 Hz**	**115.0 Hz**	**173.0 Hz**	**230.5 Hz**	**288.0 Hz**
Original / Pa	3548.56	2285.85	2587.58	2038.66	532.74
Current / Pa	3961.51	1814.98	819.65	293.45	105.77

4.2.2 Noise test:

The noise tests were performed for the original and the current configurations by using the standard ten points sound power test method in a semi-anechoic room as shown in Figure 7. And the noise test results were listed in Table 4 and Table 5 respectively.

It can be seen from Table 4 and Table 5, the noise level at the 173Hz was improved as compared the original configuration after adding the guide pipe at the bottom and the top chambers. Therefore, the proposed solution in this work can improve effectively the noise level at the 173Hz by decreasing the low frequency pressure pulsation.

Table 4: Comparison of typical peak value of low frequency noise in the top chamber

Top chamber	**58 Hz**	**173 Hz**	**178 Hz**	**182 Hz**	**231 Hz**	**288 Hz**	**298 Hz**
Original / dB(A)	25.67	**27.20**	35.25	22.67	30.52	30.56	29.20
Current / dB(A)	24.75	**23.67**	29.04	16.52	30.32	28.27	25.57

Table 5: Comparison of typical peak value of low frequency noise in the bottom chamber

Bottom chamber	**58 Hz**	**173 Hz**	**178 Hz**	**182 Hz**	**231 Hz**	**288 Hz**	**298 Hz**
Original / dB(A)	26.05	**26.91**	32.95	15.18	31.03	24.97	26.31
Current / dB(A)	25.47	**18.09**	27.24	7.34	28.59	18.60	21.41

Figure 7: Noise test fixture

5. CONCLUSIONS

The mechanism of the low frequency pressure pulsation of a typical rotary compressor was studied and an effective solution was proposed and validated in this work.

Based on the work, we can obtain a few conclusions as follows:

- The peak value at the third resonance frequency is mainly caused by the rotation flow of the refrigerant pushed by the balance block.

- The proposed solution can be used to improve the flow field near the outlet of the shaft hole, consequently, the low frequency pressure pulsation can be improved.

- The numerical and experimental method used during resolving the problem in this work can provide a reference for the similar problem for the rotary compressor.

REFERENCES

Istvan, L. & Beranek, L. (2006). *Noise and Vibration Control Engineering - Principles and applications*: John Wiley & Sons.

Cyklis, P. (2001). Experiment identification of the transmittance for any element of the pulsating gas manifold. Journal of Sound and Vibration, 244(5): 859-870.

Jeongil, P., Jeonghoon, K., Jihoon, C., Philsoo, J. & Minchul, Y. (2008). Reduction of the low Frequency Noise Due to the Discharge Pressure Pulsation of a Reciprocating Compressor. *Proceedings of 2008 the International Compressor Engineering Conference.*

Han, W. L., Han, S. L., & Bai, C. Q. (2013). Numerical and Experimental Research on Gas Pulsation in Reciprocating Compressor Piping System. Journal of Xi'an Jiaotong University, 47(5), 61-71.

Keith, A. N. (2014). Influence of Cylinder Bore Volume on Pressure Pulsations in a Hermetic Reciprocating Compressor.

ACKNOWLEDGEMENT

This work was supported by the Zhuhai Landa Compressor Co., LTD., Project No. PJ015.