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High Frequency Aerodynamic Noise Improvement of Variable Speed Scroll Compressor by Transient CFD Analysis

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ABSTRACT

With the variable speed scroll compressor operation frequency increasing, the compressor noise also rises. Reducing the compressor noise becomes one of the main challenges of new compressor development stage. In this paper, based on spectrum and visualization analysis, high frequency noise is in the high-pressure cavity of scroll compressor. By pressure pulsation test, the high frequency noise is caused by gas pulsation in the discharge port area of scroll set. New mufflers are designed to stabilize the gas pulsation of discharge port area and reduce the high frequency noise. By computational fluid dynamics method, the optimal muffler is chosen by the comparison of pressure pulsation simulation result. The muffler prototype is assembled in the target scroll compressor and tested with different frequency. Compared with original compressor without muffler, the compressor noise was reduced significantly, especially for the high frequency band which from 1000Hz to 3150 Hz.

1. INTRODUCTION

With the growing threat of global warming, it is more urgent to improve the energy utility efficiency of refrigeration air conditioning product. Variable speed compressor becomes more popular in refrigeration air conditioning field due to higher energy utility efficiency. In order to cover much wider operating condition, more and more manufactures enlarge the frequency scope of variable speed compressor. However, the noise also rises with the operation frequency increasing.

The growing awareness of comfort increased the need for compressor quiet operation. In response to these requirements, research institutions and enterprises have been concerned about compressor noise for decades, Takahide and Makoto (1995) analyzed the mechanism of noise produced by pressure pulse in scroll compressor [1]. Masanori (2002) put forward noise reduction technology for inverter-controlled scroll compressors [2]. Huang (2016) put forward to reduce compressor noise by controlling pressure pulsation [3]. Danfoss has been focused on the noise of compressor. Based on experimental analysis and numerical simulation, the vibration and noise of scroll compressor is systematically analyzed.

In this paper, the noise sources of scroll compressor are systematically introduced. The high frequency noise (1000Hz~3150Hz) of compressor operating at 120Hz is higher than 90Hz with noise spectrum analysis. The high frequency noise is located at the high-pressure cavity of compressor with sound energy visualization analysis. The dynamic pressure sensor is installed in the discharge port area of compressor to monitor the dynamic pressure pulsation status. The high frequency noise is increased due to big gas pulsation of discharge port area. Therefore, based on the expansion chamber and perforated plate noise reduction theory, three different mufflers are designed to reduce the gas pulsation of discharge port area. Computational fluid dynamics method is applied in this paper, twinmesh and ANSYS meshing are used to generate the mesh, and CFX is used to calculate the transient process of scroll orbiting. The optimal muffler scenario 1 is chosen by the comparison of pressure pulsation simulation result. Muffler prototype assembled and tested to verify the noise improvement effect. From the test result and spectrum analysis, the compressor noise was reduced about 1 dBA at 90Hz and 2-3 dBA at 120Hz, especially for the high frequency band, there was a significant improvement from 1000Hz to 3150 Hz.

2. SCROLL COMPRESSOR NOISE SOURCE IDENTIFICATION

2.1 The Mechanism of Noise of Scroll Compressor

Typical Danfoss scroll compressor structure is shown in Fig.1 which includes low pressure and high-pressure cavity. High pressure cavity is from the discharge port of fixed scroll to the discharge fitting of compressor. low pressure cavity from suction fitting of compressor to suction port of orbiting scroll.

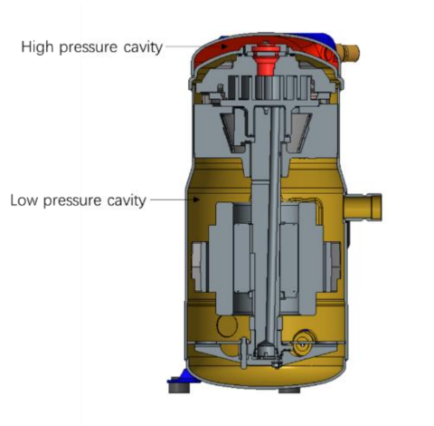


Figure 1: Typical Danfoss Scroll compressor cross section

Scroll compressor noise sources include mechanical noise, aerodynamic and electromagnetic noise. Mechanical noise is caused by crankshaft rotation unbalance, sliding friction and mechanical impact, which transmits outward through bracket, bearing and shell. The electromagnetic noise comes from radial forces between stator and rotor magnetic fields, unbalanced magnetic pull, and carrier modulation of inverter, which transmits outwards through stator, stator spacer, bearing and shell. The aerodynamic noise mainly comes from the rotation gas flow of rotor and gas pulsation, which transmits outward through the shell [4].

About compressor noise source identification, several methods are widely used in Industrial field, including spectrum analysis, acoustic vibration coherence analysis, the near field analysis, visualization analysis and so on. In this paper, spectrum and visualization analysis are used to identify the noise source of scroll compressor.

2.2 Noise Spectrum Analysis

Spectrum analysis is one of the important tools to describe the characteristics of vibration and noise, which is a good method to study the distribution of vibration and noise intensity with frequency.

Danish B&K data acquisition instrument is applied to test the sound power signals of target variable speed scroll compressor under requested operating conditions. The test operating frequency is 90 Hz/120Hz, and the test method is hemisphere method, The CPB spectrum of the sound power signal is shown in Fig.2.

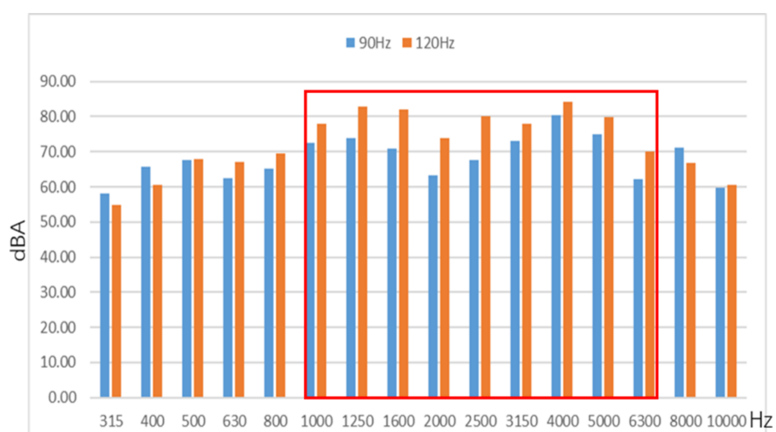


Figure 2: Scroll compressor sound power spectrum at 90Hz/120Hz

From the above spectrum analysis, for 90 Hz spectrum, the noise intensity at 1250&1600Hz is higher than others, for the rest, the high frequency (2000-3150Hz &6300Hz) noise is a little lower than low & medium frequency noise, for 120Hz, the noise intensity at 500Hz is higher than others, for the rest, the high frequency (1000-5000Hz) noise is a little higher than low & medium frequency noise. It means that the high frequency (1000-6300Hz) noise increased with the operating frequency rise.

2.3 Visualization Analysis

Visualization analysis is one of the important tools to identify the location of noise. With microphone matrix, accelerometers or a professional sound energy visualization camera, the sound energy generated by the whole compressor can be clearly seen, and the status of sound energy is distinguished by different colors. It is widely used in the noise source analysis and noise location identification.

Scroll compressor is a hermetic structure. All noise radiates to outside through the shell. Therefore, the main noise source can be identified by analyzing the sound energy distribution on the shell. In order to test the sound energy distribution on the shell, the compressor is divided into 9 layers at the direction of height, and 12 equal parts along the horizontal plane of the compressor as shown in Fig.3.

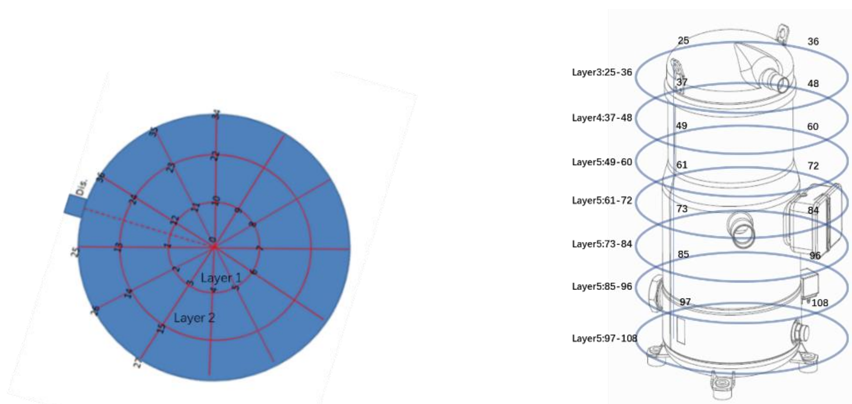


Figure 3: Test area partition graph

The target scroll compressor is tested at operating frequency 120Hz. Accelerometers are put in the center of each defined grid, and data are recorded by B&K instrument. The test data are processed by MATLAB, and the main visualization analysis graphs of the target scroll compressor are shown in Fig. 4.

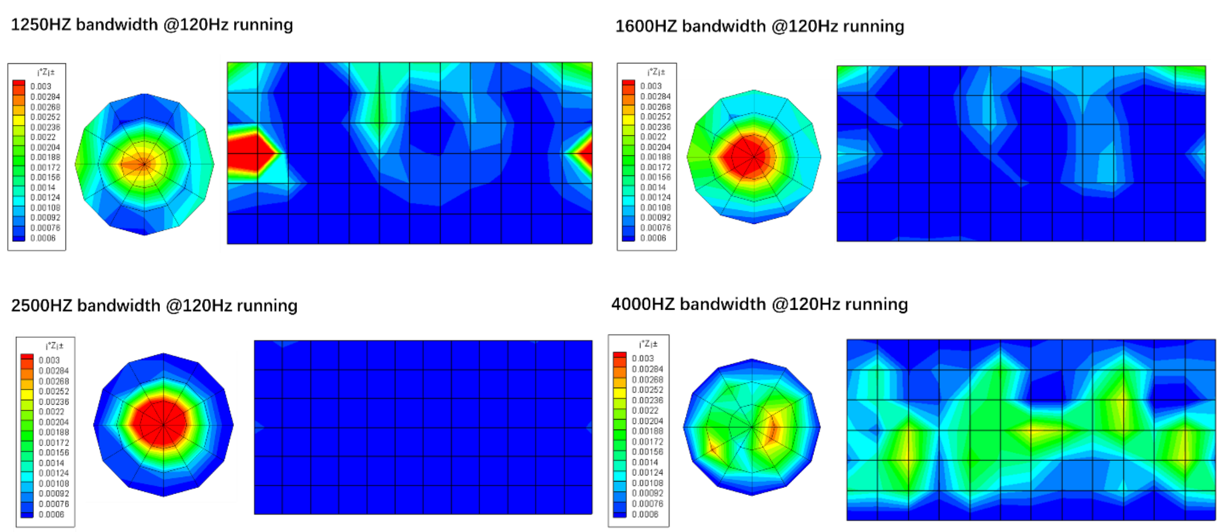
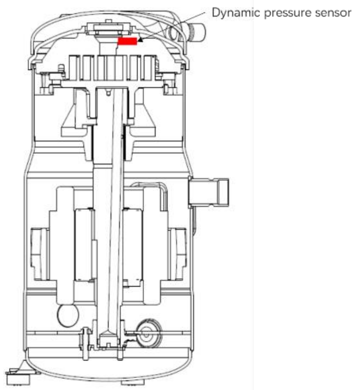


Figure 4: The Visual Figure of Signal

From the above sound energy visualization analysis, for 1600Hz and 2500Hz, the main noise source is located at high pressure cavity area of compressor. For 1250Hz, except for the high-pressure cavity area, the biggest sound energy is caused by electrical box of compressor. For 4000Hz, except for the high-pressure cavity area, the sound energy is also caused by motor.

2.4 Discharge Port Pressure Pulsation Test

Based on above analysis, the pressure pulsation of discharge port area is possible to generate much bigger noise in high pressure cavity. The dynamic pressure pulsation test of discharge port is designed, and the dynamic pressure sensor is put at the discharge port location as shown in Fig. 5. In order to analyze the reason why the high frequency noise at 120Hz is higher than 90Hz, the test operating frequency is 90Hz and 120Hz.



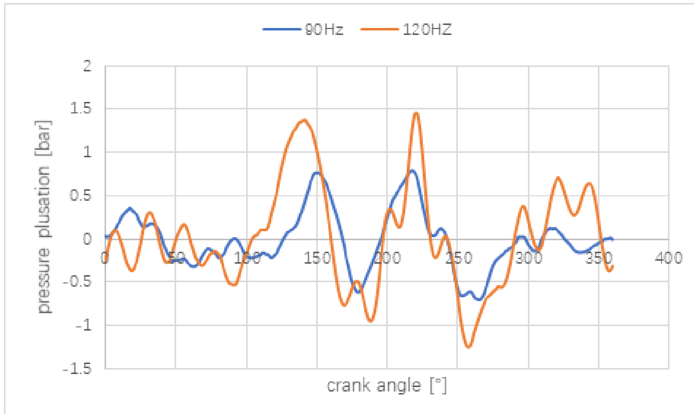


Figure 5: Discharge port dynamic pressure pulsation curve

The absolute pressure value is record by sensor, and the curves with one cycle of scroll rotation are shown in Fig. 5. To get the fluctuation value of pressure, the calculation formula as following:

$$Pressure\ Pulsation = (P_t - P_0)/10^5 \tag{1}$$

P_t: Absolute pressure.
P_0: Average pressure.

From the above pressure pulsation curve, the peak of pressure pulsation at 120Hz is much higher than 90Hz.

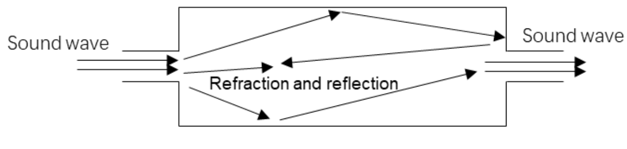
In summary, the conclusions about noise source of scroll compressor are shown as following.

1) The high frequency noise increased with compressor operating frequency rise;
2) The high frequency noise is mainly located in the high-pressure cavity of the compressor;
3) The high frequency noise is mainly caused by the pressure pulse at the compressor discharge port, it is continuous aerodynamic noise.

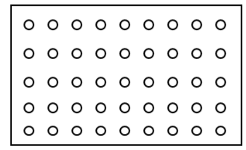
3. NEW MUFFLER MODELING

For the high frequency aerodynamic noise, two main noise reduction methods are applied widely in industrial field. One method is to weaken the noise radiation to outside of the compressor, e.g. sound hood. Another method is to optimize the internal structure of high-pressure cavity, e.g. adding muffler in the discharge port of the compressor. In this paper, new muffler is designed and applied.

Based on classical muffler structure and theory analysis, different type of muffler has different advantage and disadvantage, any single type of muffler is not suitable to be applied in scroll compressor. The new muffler is designed with the combination of resistance muffler and perforated plate. In the resistance muffler, the noise is reduced by the reflection and interference of sound energy. The main feature of resistance muffler is shown in Fig. 6 (a). For the perforated plate sound absorption structure, the gas becomes more stable due to even-distributed holes in the plate. the main feature of perforated plate is shown in Fig. 6 (b)



(a)



(b)

Figure 6: Different muffler types

The expansion chamber and perforated plate are included in the new muffler. The transmission loss TL of expansion chamber is:

$$TL = 10\lg\left[1 + \frac{1}{S}\left(m - \frac{1}{m}\right) sin^2\, nl^2\right] \tag{2}$$

m: expansion ratio of the chamber.
S: the cross-sectional area of the connecting pipe.
l: the length of the chamber
n: wave number

Expansion ratio of the chamber *m*:

$$m = \left(\frac{D}{d}\right)^2 \tag{3}$$

D: the diameter of chamber section.
d: the diameter of inlet/outlet tube

When the length of the expansion chamber is odd number times of the quarter wavelength, the transmission loss of the expansion chamber reaches the maximum value [6].

$$l = \frac{n_1 v}{4f} \quad n_1 = 1,3,5 \ldots \ldots \tag{4}$$

f: frequency.
v: sound velocity.

Based on above theory, the length of chamber is designed to reach the max. transmission loss, and the diameter of chamber section and inlet/outlet tube are defined by the structure of high-pressure cavity. The new expansion chamber structure is shown in figure 7.

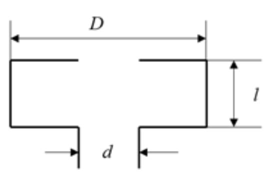


Figure 7: Expansion chamber structure

About the perforated plate design, the holes are designed to even distributed in the plate, several different perforated plate scenarios are shown in Fig. 8.

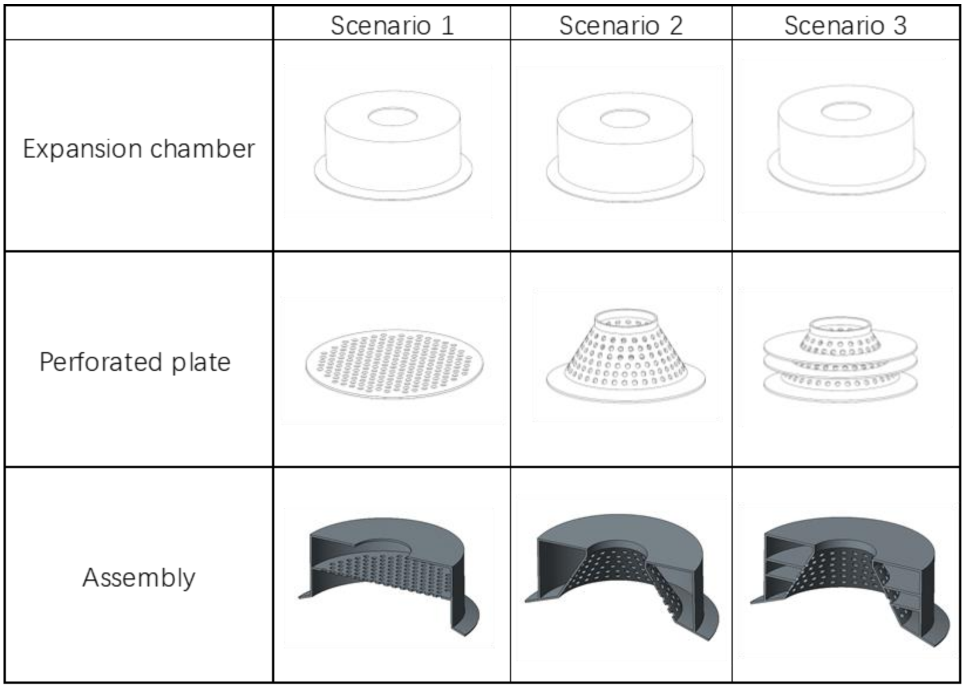


Figure 8: Mufflers design scenarios

4. COMPUTATIONAL FLUID DYNAMIC MODEL AND PRESSURE PULSATION ANALYSIS

4.1 Computational Model

These three mufflers are separately assembled in same compressor as shown in the Fig. 9. Outlet is extended to a suitable position, and the vents on housing are extruded as the simplified inlets. These models are created by two software, which are ANSYS Design Modeler for stationary part and Twinmesh for dynamic part. The whole model is transient calculated by ANSYS CFX. The pressure pulsation result at the monitor point is compared with different scenarios.

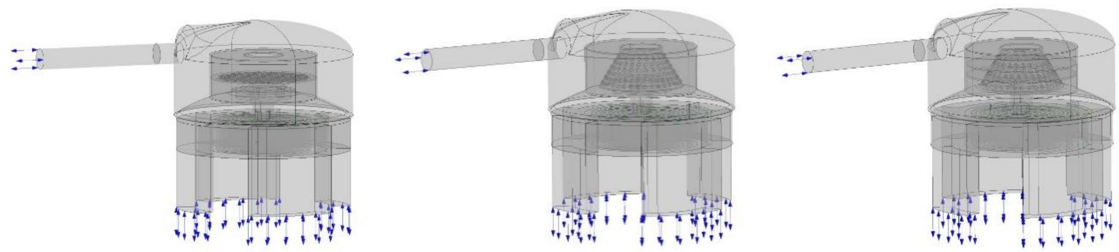


Figure 9: Computational model with different muffler scenarios

Several hypotheses are shown as following:
1) Evaporation/condensation pressure is used for boundary condition at inlet and outlet;
2) Real gas model is used for refrigerant;
3) Axial leakage is not considered;
4) Outside walls are thermal isolation.

4.2 Result Analysis and Discussion

After calculation of 6 circulations, it gets convergence. The result of pressure and mass flow rate show regular periodic tendency. The mass flow rate difference of inlet and outlet is varying in a very small range. The pressure pulsation simulation result of final circulation is extracted.

During post process, pressure pulsation is recorded at a monitor point where the pressure sensor is assembled in test. The position is in discharge plenum under discharge valve which is shown as Fig. 10.

The character of the pulsation is determined by the feature of scroll, which has a big peak and three smaller peaks. The pressure fluctuation of all the proposals are shown in Fig. 11:

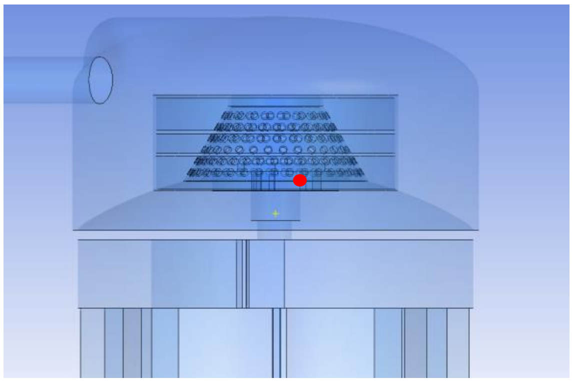


Figure 10: Monitor point in compressor

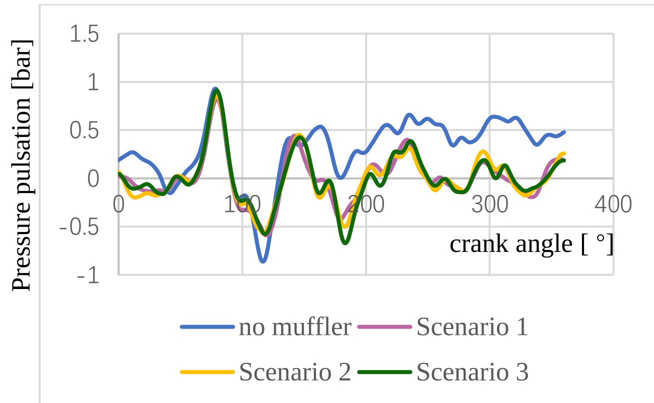


Figure 11: Pressure pulsation comparison curve

The average values of pressure pulsation are shown in table 1:

Table 1: Average values of pressure pulsation

	No muffler	Scenario 1	Scenario 2	Scenario 3
Aver.	0.24833	0.18156	0.19543	0.18965

From calculation result, scenario1 is the best of 3 muffler scenarios. The pressure pulsation of scenario 1 is the smallest. At the same time, scenario 1 is easy to be machined, so the scenario1 is selected for prototype making.

5. PROTOTYPE TEST AND VERIFICATION

After pressure pulsation simulation, the optimal proposal scenario1 is tested in R&D laboratory, the test bench is shown in Fig. 12.

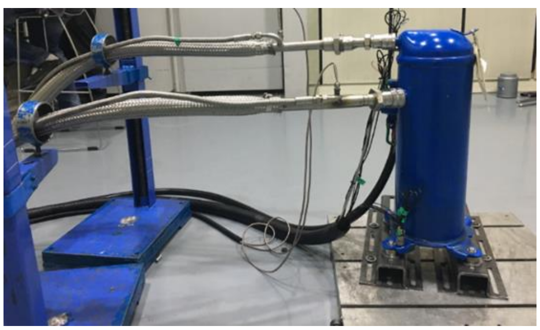


Figure 12: Noise test in R&D laboratory

In order to verify the effect of muffler, the compressor with muffler is tested @ 90 Hz/120Hz, The CPB spectrum of the sound power signal is compared with that of compressor without muffler in Fig. 13.

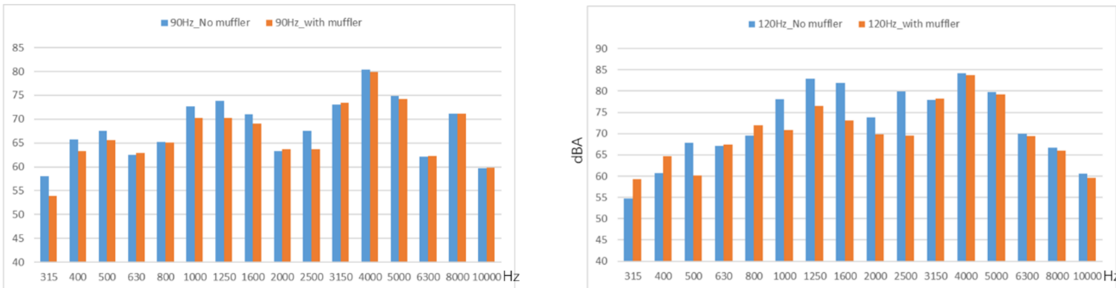


Figure 13: Test spectrum analysis

From the test result and spectrum analysis, the compressor noise is reduced about 1 dBA at 90Hz and 2-3 dBA at 120Hz, especially for the high frequency band, there is a significant improvement from 1000Hz to 3150 Hz, The result of the noise test is in good alignment with the conclusion of the previous noise source analysis.

6. CONCLUSIONS

Based on spectrum analysis, visualization analysis and pressure pulsation test, the noise source is identified. The optimal muffler is designed by CFD method and verified by prototype test. Several conclusions are summarized as following:

(1) With the scroll compressor operating frequency increasing, the high frequency noise of compressor is also increased. The high frequency noise is mainly located in the high-pressure cavity of the compressor;

(2) There is correlation relationship between the gas pulsation of discharge port area and the high frequency noise of compressor. When the gas pulsation of discharge port area is stable, the high frequency noise is reduced;

(3) Computational fluid dynamics method is used to transient simulate the gas pulsation of scroll set. The optimal muffler can be chosen by the comparison of pressure pulsation simulation result;

(4) The new designed muffler reduces the high frequency aerodynamic noise of scroll compressor, the compressor noise is reduced about 1 dBA at 90Hz and 2-3 dBA at 120Hz. Especially for the high frequency band, there is a significant improvement from 1000Hz to 3150 Hz.

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