Research on Noise Reduction of Variable Speed Rotary Compressor with Large Capacity

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ABSTRACT

With the increasing speed and capacity of variable-speed rotary compressors, the problem of noise especially low and medium frequency noise in the air conditioning system which can't be solved by wrapping soundproof cotton has became more serious. In this paper, based on the noise problem of the rotor compressor with a working capacity of more than 80CC, the main frequency and the position of the noise source within 1000Hz are confirmed by simulation and experiment. Then on the base of this，the muffler and accumulator are respectively optimized and improved combining with Computer Aided Engineering (CAE) means. The final application results show that the optimized scheme can reduce noise by 6.1dB in 160Hz and 8.9dB in the frequency range of 500Hz to 800Hz, achieving good results.

1. INTRODUCTION

The rotor compressor has good refrigeration performance and variable condition ability, which makes it have great application advantages in small refrigeration device. It constantly breaks through the upper limit of refrigeration capacity to replace small scroll compressor, and is widely used in household air conditioner and small commercial refrigeration device. Nowadays, rapid increasing of refrigeration and air conditioning system in modern industries brings attention to the urgency of core technology development in this area. (Qian and Zhang(2018)) A variable speed rotor compressor with a working capacity of 80CC has been designed and developed, but its noise also increases significantly with the increase of refrigeration capacity, especially the medium and low frequency noise which is difficult to be suppressed by external sound-absorbing materials. At present, great progress has been made in the research on vibration and noise reduction of rotary compressors at home and abroad. Shang Zhiwu(2003) researched the identification method of the noise source of the rotary compressor and achieved the goal of vibration and noise reduction by optimizing the structure of the muffler and accumulator. Kwang ha Suh and Jin Dong Kim(2000) studied the muffler of the rotary compressor, and explored the influence of the exhaust hole and cavity capacity of the muffler in the concerned frequency band on the noise reduction. In order to reduce the noise of the newly developed rotor compressor, noise analysis and low-noise design were carried out in this study.

2. MECHANISM of NOISE GENERATION

In order to define the target of noise reduction and the mechanism of noise generation, the noise of the compressor in the single machine state and the state of wrapping sound-absorbing material in the simulation air-conditioning system are tested respectively in the semi anechoic room, and the sound power spectrum is shown in Figure 1. It can be seen that there are peaks in the frequency bands of 160Hz, 500-800Hz and 1600-2500Hz for single compressor under test, while the noise attenuation above 1000Hz is obvious in the state of analog air conditioning system, so the main noise reduction target is the peaks in the frequency bands of 160Hz and 500-800Hz.

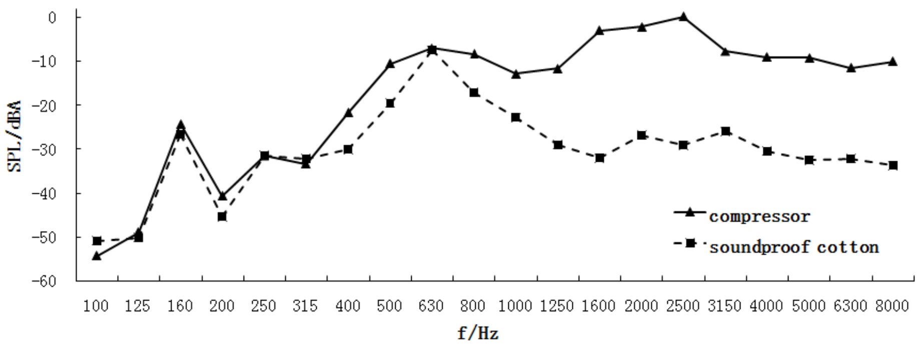
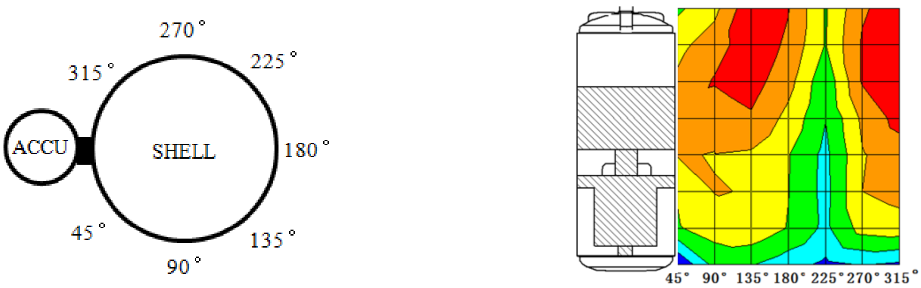


Figure 1: Noise level of rotary compressor

2.1 The mechanism of Medium-frequency Noise Generation

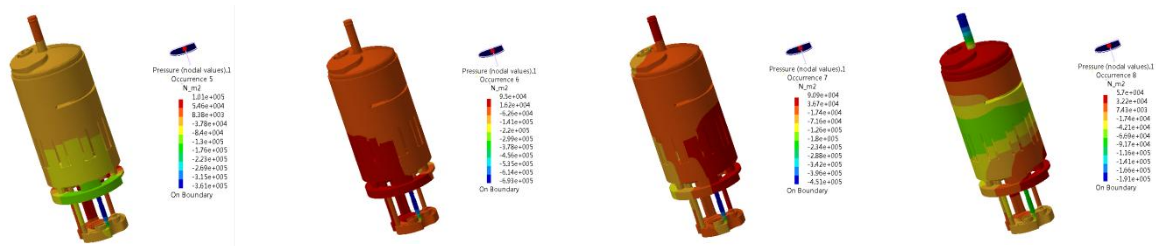
The research method of the noise source location of the rotary compressor is relatively mature that shown in Yang Weibiao (2000). In order to confirm the location of the noise source in the frequency range of 500 to 800Hz, the test of near-field noise source is carried out for the compressor. The location of the measurement point and the test results in 630Hz are shown in Figure 2. It can be confirmed that the medium-frequency noise is mainly located in the upper part of the compressor cavity, which includes cavity resonance and motor noise, etc.



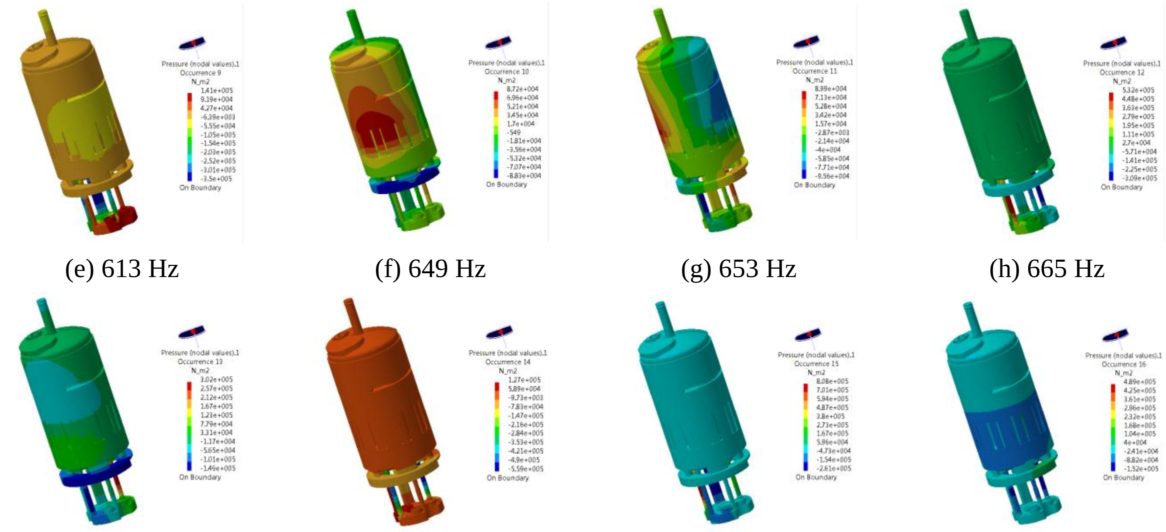
(a) (b)

Figure 2: (a)Schematic view for measurement point in detail, (b) result of experiment

In order to further confirm the mechanism of noise generation, the acoustic Finite Element Method (FEM) is used to calculate the acoustic mode of the compressor cavity. It is found that there are multi-stage cavity acoustic modes in the frequency range of 500 to 800Hz. The calculation results are shown in Figure 3. In order to verify the accuracy of the numerical calculation results, the acoustic mode of the compressor was tested. The frequency response curve obtained from the test is shown in Figure 4. It can be seen that there are multiple peaks of sound pressure response in the concerned frequency band, which indicates that there are multiple acoustic modes. The excitation frequency of fluid pulsation in compressor is broadBand, which is easy to excite acoustic mode and cause noise problems.



(a) 521 Hz (b) 551 Hz (c) 564 Hz (d) 577 Hz



(e) 613 Hz (f) 649 Hz (g) 653 Hz (h) 665 Hz

(i) 685 Hz (j) 724 Hz (k) 742 Hz (l) 799 Hz

Figure 3: The schematic view mode shape in a rotary compressor

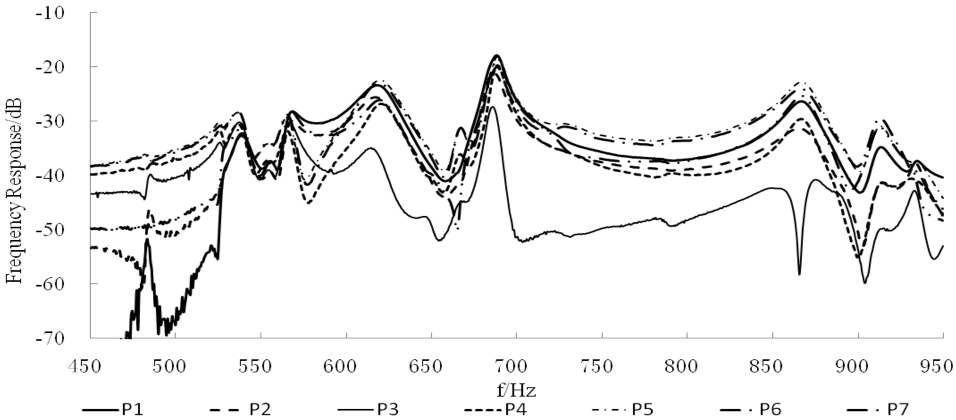


Figure 4: Frequency response function of cavity

2.2 The Mechanism of Low-frequency Noise Generation

Due to the poor effect of microphone on the sound source positioning test in 160Hz, the sound source analysis is carried out by testing the acceleration of different positions on the surface of the whole machine, and the position of the test point is shown in Figure 5. As shown in Figure 6, it can be found that the vibration points on both sides of the accumulator in the tangential direction of the compressor are significantly higher than the other measurement points, and the values of the other measurement points are similar, which indicates that the noise source in the 160Hz is located in the accumulator.

As we all know, some natural modes of the structure itself will be excited due to the working load, which will generate corresponding responses at the excited modal locations. These superimposed responses are the mechanical noise we are concerned about. In order to clarify the mechanism of mechanical noise in 160Hz, the structural mode of the accumulator was tested by the impact testing. The results are shown in Figure 7. At the same time, because the compressor uses a two-cylinder structure, the pulsating frequency of its load torque is 2 times the rotational speed frequency. The double-speed frequency in 160Hz has a strong excitation force when running at 4800 rpm. The operating load of the compressor caused the first-order tangential swing mode of the accumulator to be excited, and the structural resonance caused the generation of noise, which explained the mechanism of noise at 160Hz.

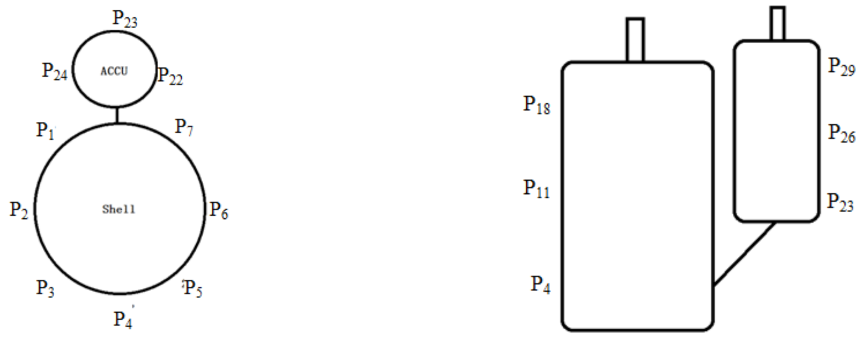


Figure 5: Schematic view location of vibration sensor

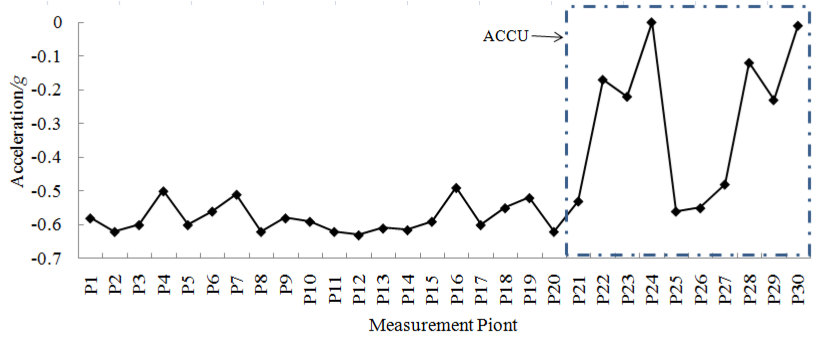


Figure 6: Schematic view experimental result of vibration

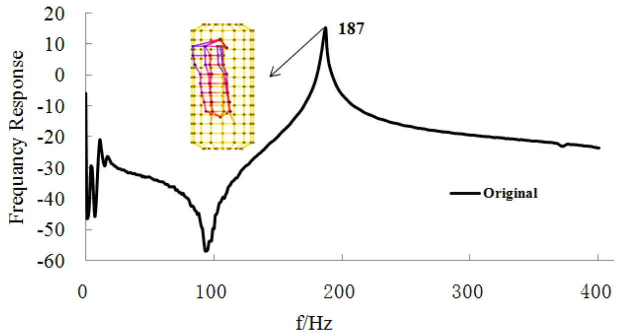


Figure7: Schematic view mode shape and frequency response function (FRF) of the accumulator

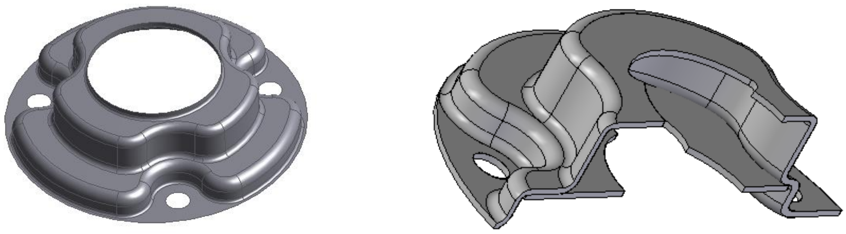
3. OPTIMITZATION OF MUFFLER

The previous section confirmed that the noise in the frequency range of 500 to 800Hz is caused by the resonance which caused by the acoustic mode of the compressor's cavity, which can be reduced by optimizing the structure of the compressor's muffler to reduce the exhaust pulsation and change the excitation frequency.(Huang and Liu(2007), Li and Jia(2009), Liu(2004)) The evaluation method of muffler is established to evaluate the accuracy of optimization design and confirm the optimization effect. Based on the muffler model, Finite Element Method(FEM) is used to calculate the transmission loss, which is also used as the basis of muffler performance evaluation.

The muffler of the rotor compressor belongs to the resistant muffler, which can suppress the noise in a specific frequency band through the reflection and superposition of sound waves. The noise reduction performance of this muffler is closely related to its volume, cavity distribution, location and area of the inlet and outlet. In this study, these parameters are optimized. Figure 8 shows the three-dimensional model of the original structure and the improved structure. The original muffler is a single-layer structure, which can be optimized by adding a layer of partition in the cavity of the muffler. At the same time, the outlet structure of the muffler is also improved.

Figure 9 shows calculation results of the transmission loss of the muffler before and after improvement. It is found that by changing the structure of the muffler, the acoustic mode of the cavity inside the muffler is adjusted, and the transmission loss performance of the muffler is improved. In the target frequency band of 500 to 800Hz, the transmission loss of the optimized muffler is significantly improved.

Original



Optimized

Figure8: Modification of muffler shape

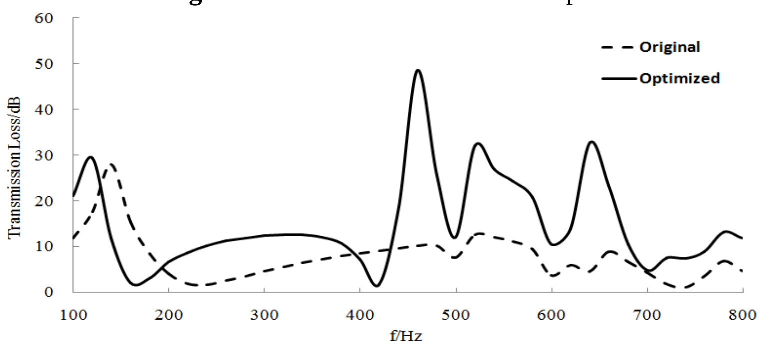


Figure 9: The schematic view transmission loss of the muffler

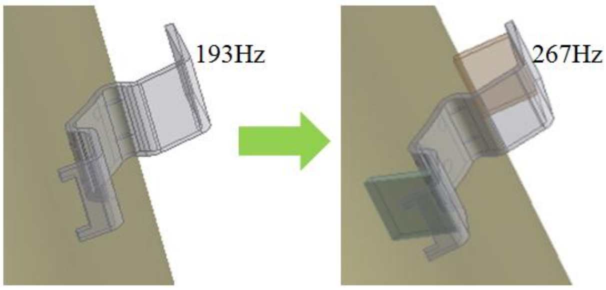
4. OPTIMITZATION OF ACCUMULATOR

The low-frequency noise in 160Hz is high, which has a great correlation with the tangential mode of the accumulator. Therefore, the natural frequency of the main structural mode of the accumulator is far from the range of second harmonic frequency of rotational speed by optimizing the structure of the accumulator, so as to achieve the goal of noise reduction.(Zhu and Gao(2008), Xiang and Ye(2004)). Considering that the operating speed range of the variable frequency compressor is broadband, which can reach 3000 to 6000 RPM (50 to 100 rps), the natural frequency can be adjusted by optimizing the tangent stiffness of the accumulator, and the first-order tangential swing mode can be increased to the area far away from the rotational speed frequency.

The natural frequency of the structure is positively related to its own stiffness. The increase of the stiffness of the accumulator system results in the increase of its modal frequency. The tangential stiffness of the accumulator is improved through the optimization design of the tank-stay of the accumulator. The CAE method is used to analyze and evaluate the optimization schemes. The results of the optimization scheme and its modal natural frequencies are compared with that of FIG 10. After the tank-stay is optimized, the first order tangential swing mode frequency of the optimal scheme is increased from 193Hz to 267Hz.

Impact testing is used to verify the structural mode of the accumulator before and after improvement. Figure 11 shows the frequency response characteristic curve of the accumulator structure. The test results are similar to the simulation results, and the modal frequency of the accumulator has been significantly improved.

Original



Optimized

Figure10: Modification of tank-stay shape

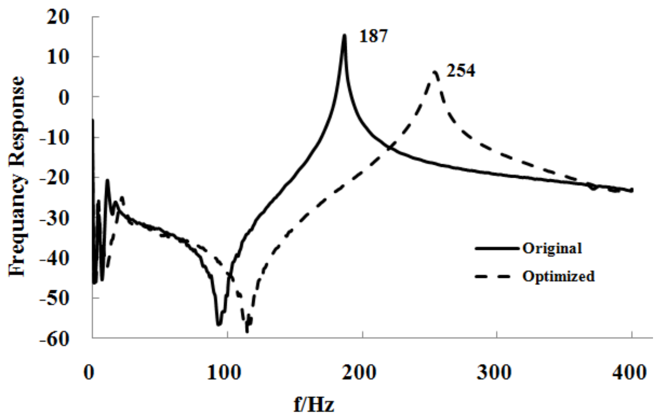


Figure 11: The frequency response function (FRF) of the accumulator

5. EFFECT of NOISE REDUCTION

Based on the optimization of the accumulator structure, the natural frequency of tangential mode is larger than the range of the second harmonic frequency of rotational speed. The actual improvement effect can be verified by testing the vibrations of the accumulator before and after the optimization, as shown in Figure 12. It can be seen that the vibration of the accumulator is significantly reduced by using the improved tank-stay, and the average vibration amplitude is reduced to about 52% at multiple rotating speeds, which has achieved good vibration reduction effect.

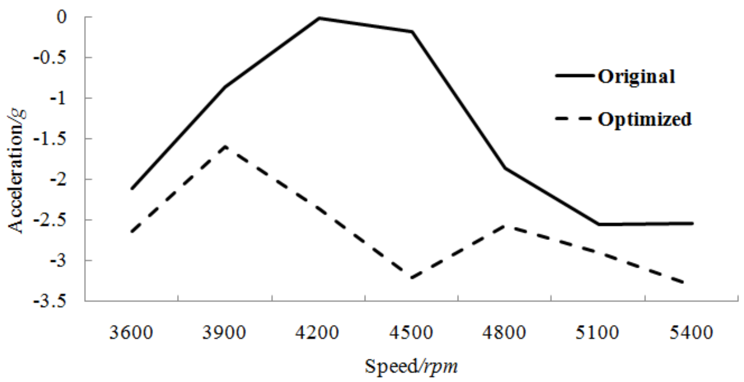


Figure 12: Schematic view experimental result of tangential vibration of the accumulator

The optimal solution was selected for trial production, and the noise test was carried out to verify the results as shown in Figure 13. Compared with the original compressor, it can be seen that the noise power of the compressor decreases by 8.9dB on average in the frequency range of 500-800Hz and 6.1dB in 160Hz, and the noise reduction effect is obvious.

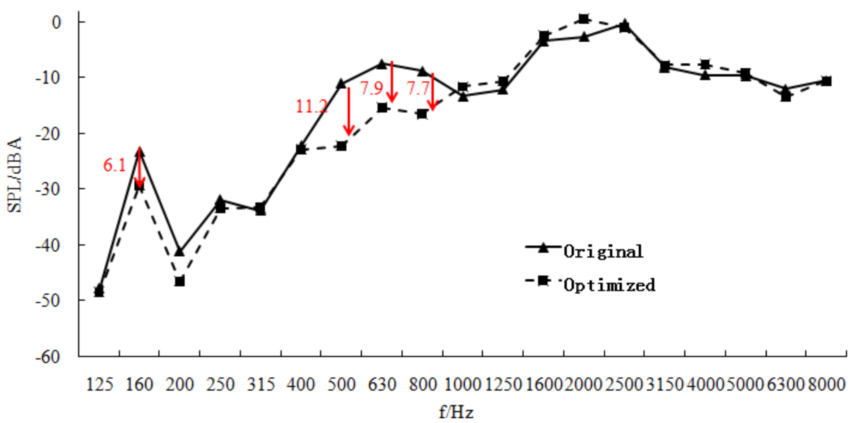


Figure 13: The schematic view noise pressure

6. CONCLUSIONS

In this paper, the high noise of the newly developed variable-speed compressor with a capacity of more than 80CC is studied. The location and generation mechanism of the main noise sources within 1000Hz of the compressor are confirmed by noise experiment, the location of noise source, impact testing and numerical analysis. On the basis of the analysis, the numerical analysis method of the structure of the accumulator and muffler is established, and the structure optimization and evaluation of the two models are carried out. The final optimization scheme test shows that the average vibration amplitude is reduced to about 52% at multiple rotating speeds, and the compressor can reduce noise by 6.1dB in 160Hz and 8.9dB in the frequency range of 500 to 800Hz, achieving good effect of vibration and noise reduction.

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