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Research on Low Frequency Vibration of Rotary Compressor

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ABSTRACT

Low frequency noise which sounds annoying is easy to penetrate walls. To a certain domestic air-conditioner, the abnormal low frequency noise of the outdoor unit is detected when the compressor operating at certain low RPM's. Experiment and simulation analysis are taken, and it is determined that the 1st order rigid modal resonance of the accumulator is excited at certain operating RPM's which causing structure-borne noise passes through suction pipes. To solve this problem, two schemes are produced to optimizing the connection property between the accumulator and compressor shell: 1) increasing connection damping to reduce resonance region response; 2) increasing connection stiffness in order to promote modal frequency to avoid being excited resonance.

Finally, the vibration experimental verification of the two schemes is carried out, and scheme 2 is better than scheme 1 which is also better than the original one, more than 30m/s² reduced.

1. INTRODUCTION

Generally, rotary compressor is a main noise source of air-conditioner outdoor unit, which is composed of two components, airborne noise and structure-borne noise. The airborne noise is directly radiated from the compressor shell. The structure-borne noise is caused by the vibration transmitted through mounting system, suction pipes and
discharge pipes that are linked to the compressor [1].

Many researches has been done on the noise caused by the accumulator [2][3][4], but the low frequency vibration caused by accumulator were studied rarely until now, which may cause low frequency boring noise to the outdoor unit.

Usually the accumulator of rotary compressor is connected to the main shell by two parts, suction pipe and holder, as shown in Figure 1. The connection stiffness between the accumulator and the main shell is usually not so high which cause the accumulator easily resonating at low frequency by unbalanced forces and moments.

The abnormal low frequency noise of a domestic air-conditioner is studied in this paper. It is determined that the 1st order rigid modal resonance of the accumulator is excited at certain operating RPM's which causing structure-borne noise passes through suction pipes. The connection structure between the accumulator and the main shell is optimized in order to solve this problem.

![Figure 1: A sketch of accumulator connection](image)

### 2. ABNORMAL NOISE AND IDENTIFICATION

#### 2.1 Characteristic of the Abnormal Noise

The abnormal low frequency noise appears when the compressor operates at about rotation frequency 30 to 40Hz. It sounds annoying because the difference between the peak value and the total value of the noise is small and the frequency of the peak value is very low, as is shown in Figure 2.
2.2 Noise Source Identification

To solve this problem, FEM and experiments analysis are taken to identity the noise source.

Firstly, it can be confirmed that the abnormal noise is cause by the rotary compressor according to the spectrum analysis. The frequency of the peak noise is exactly 6 times the rotation frequency of the compressor and changes with the change of the compressor operating frequency.

Secondly, it can be concluded that the abnormal noise is a structure-borne noise rather than an air-borne noise according to the result of comparative test which is taken to compare the noise of the unit with and without the side panel. The peak value of abnormal noise reduced about 10 dB when the side panel of the unit is removed, as shown in Figure 3.

Finally, it is validated that the accumulator of the compressor is resonated at about 200Hz by taking vibration test on the compressor and modal analysis with FEM and experimental methods. The accumulator is excited resonance at about 200Hz by the 2nd, 4th and 6th order exciting forces, as shown in Figure 4. The modal frequency of the first rigid mode of accumulator is at about 206Hz according to the result of FEM and experimental modal analysis, as shown in figure 5.

It is clear that the abnormal noise of the unit is a structure-borne noise which is transmitted from the huge low frequency vibration of accumulator caused by resonating. So the problem of low frequency vibration of rotary compressor caused by the resonance of accumulator is study below.
Figure 3: Noise spectrum comparison before and after removing the side panel of unit

Figure 4: Order spectrum of accumulator tangential vibration

(a) FEM result   (b) Experimental result

Figure 5: Modal analysis of the accumulator
3. LOW FREQUENCY VIBRATION ANALYSIS AND IMPROVEMENT

3.1 Modal Property of the Accumulator
The influence of compressor rigid mode on the low frequency vibration of compressor has been studied before [1]. Besides, the local rigid mode of the accumulator also has a great influence on the low frequency vibration of compressor, because its first order modal frequency is relatively low. The pipes of the outdoor unit may probably operating breakout and abnormal noise will arises if these low frequency modes are excited to resonance.
To the accumulator, the whole body mode which is vibrated as a whole rigid body can be called local rigid mode [2]. Six rigid modes are composed of modal shapes that translate and rotate along the 3 axes, as shown in Table 1. Attention should be paid to the first rigid mode as defined below, because it is easily exited to resonance by unbalanced forces and moments.

| Table 1: Six local rigid modes of the Accumulator to a dual rotary compressor |
|---|---|---|
| ![1st](image1) | ![2nd](image2) | ![3rd](image3) |
| ![4th](image4) | ![5th](image5) | ![6th](image6) |

3.2 Discussion of Improvement Scheme
The following improvement schemes are discussed from the angle of reducing the vibration of the accumulator.
1) Reducing the exciting forces and moment
As shown in Figure 3, to a dual cylinder compressor, the first rigid mode is easily be excited resonance by dynamic magnetic force and torque at low rotation frequencies, and be excited resonance by unbalanced 2nd and 1st-order gas force and torque at middle and higher rotation frequencies. Obviously, if we want to reduce the vibration from the angle of reducing the exciting force, we need to optimize the design of motor, and we need to compensate the gas torque, which is more difficult to carry out.

2) Improve the modal damping in order to reduce the response of the resonance region
The modal damping can only be increased by increasing the connection damping of the suction pipes and the support system. A better solution is to use high damping rubber material. However, from the modal shape of the first rigid mode, the use of this scheme may not improve modal damping significantly.

3) Improve modal frequency and avoid resonance
Theoretically, to avoid resonance is the most effective solution. The modal frequency of the first rigid mode should be improved sufficiently high when taking account the widely operation frequency range of compressor, which may lead to increased costs.

4) Change the modal frequency and avoid operating at the rotation frequencies that may resonance.
From the discussion above, we can find that the scheme 1) and 3) will solve this problem thoroughly although it is difficult and costly to carry out, meanwhile scheme 2) may not useful. Then, changing the modal frequency and avoiding operating at the rotation frequencies that may resonance will be the most appropriate scheme to carry out. The modal frequency of the first rigid mode can be adjusted to a certain extent according to the demand of the system development department.

3.3 Sensitivity Analysis
Sensitivity analysis is taken to find out which is the most sensitive factors that influence the modal frequency. As shown in table 2, where the DESAR 1 is thickness of the holder, the DESAR 2 is thickness of the baffle, the DESAR 3 is modulus of elasticity of the rubber pad, the DESAR 4 is thickness of the suction pipe 1, the DESAR 5 is thickness of the suction pipe 1, the DESAR 6 is thickness of the accumulator shell.

<table>
<thead>
<tr>
<th>Design Variables</th>
<th>Sensitivity</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>DESAR 1</td>
<td>2.1994E+01</td>
<td>Hz/mm</td>
</tr>
<tr>
<td>DESAR 2</td>
<td>6.6295E-01</td>
<td>Hz/mm</td>
</tr>
<tr>
<td>DESAR 3</td>
<td>8.5020E-02</td>
<td>Hz/Mpa</td>
</tr>
<tr>
<td>DESAR 4</td>
<td>1.0686E+01</td>
<td>Hz/mm</td>
</tr>
<tr>
<td>DESAR 5</td>
<td>2.6743E+01</td>
<td>Hz/mm</td>
</tr>
<tr>
<td>DESAR 6</td>
<td>-1.3074E+01</td>
<td>Hz/mm</td>
</tr>
</tbody>
</table>

According to the results of sensitivity analysis, the variables which have great influence on the modal frequency are the suction pipe 2 and the holder. The sensitivity of the shell thickness of accumulator is negative, which indicates
that the increase of the thickness of the shell will increase the mass of accumulator and leads to the decrease of the modal frequency. The improvement of the suction pipe will result in the great change of the structure of accumulator. In addition, the results show that the sensitivity of the thickness of holder to this mode is only 22Hz/mm, that is, the increase of the thickness of 1mm can only increase 22Hz. Therefore, further improvements should be taken to the structure of holder to obtain higher stiffness.

4. IMPROMENT AND VALIDATION

According to the above analysis, two improved schemes are identified.

Scheme 1: improved holder and rubber pad using high damping rubber material.

Scheme 2: optimized high stiffness holder and normal rubber pad. The frequency of the first rigid mode increased to 588Hz due to a higher stiffness holder, as is shown in Figure 6 and Figure 7.

The effect of the two schemes is verified by the vibration test of the compressor, as shown in Figure 8. Owing to the improved holder and high damping rubber material, the modal frequency of Scheme 1 is higher than the original Scheme and the resonance region vibration is lower. No excited resonance appears to Scheme 2 during the whole rotational frequency range because of the higher modal frequency. Scheme 2 is better than Scheme 1 which is also better than the original one, more than 30m/s² reduced.

(a) Original  (b) Optimized

Figure 6: Optimized structure of the holder

Figure 7: FEM result of the first rigid mode of accumulator
5. CONCLUSIONS

The accumulator is connected with the main shell through the suction pipe and the holder. Six rigid modes exist due to these connections, and the first rigid mode is easily excited resonance by the dynamic magnetic and gas forces and moments. The low frequency vibration of the compressor is easily transmitted to the outdoor unit to cause structure-borne noise. It is very easy to cause discomfort because of its high energy and low frequency.

To solve the resonance problem, two verification schemes have been developed in this paper. The results show that although the high damping rubber material can reduce the resonance region response, the effect is not obvious. It is more effective to improve the modal frequency to avoid being excited resonance by improving the stiffness of the holder. After using this scheme, the mode is avoided being excited resonance during the whole operating frequency range, and the vibration of accumulator is reduced by more than 30 m/s².

REFERENCES


