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NOISE REDUCTION OF ACCUMULATORS FOR R410A ROTARY COMPRESSORS

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ABSTRACT

Since the accumulator is one of the largest components in R410A rotary compressors, its contribution to the total noise is substantial. Because of its importance, it deserves our attention to analyze it.

In this paper, the effects of accumulator to the total noise are analyzed from two aspects of theory and experiment. The finite element method (FEM), MATLAB software and SYSNOISE software are used in analysis. The test observations for the accumulators are presented to demonstrate the contribution. Several experimental trials and analytical approaches are used to show that the dominating noises in R410A rotary compressors are the resonance noise of suction flow noise of the accumulator from 425Hz to 475Hz.

Some valuable suggestions for design of the accumulator are given. Consequently, the structural modifications of the accumulator result in the reduction of noise level by approximately 14dB from 425Hz to 475Hz. And the total noise has reduced 1.5dB.

1. INTRODUCTION

R410A which has been considered to be suitable as alternative refrigerants of R22 has a serious noise problem because of the large relative increase in operating pressure and the change of acoustic characteristic of the refrigerant. However, few published papers showed its acoustic characteristic in rotary compressors. It is necessary to consider the acoustic characteristic of R410A rotary compressors in order to reduce noise.

Wei Zhou et al. (2000) demonstrated a typical process to investigate the vibro-acoustic behavior of R410A rotary compressors. And possible noise sources of the accumulator were discussed and the noises radiated from accumulators were predicted using three dimensional vibro-acoustic analysis (Wei Zhou et al., 1998).

The accumulator is one of the largest components in R410A rotary compressors. Its contribution to the total noise may be substantial. But Wei Zhou et al. (1998) had a result that gas pulsation in the suction cavity is not considered an important contributor to the total sound level of the noise when rotary compressors are under normal operating conditions.

The objective of this paper is to study the effects of accumulator on the total noise of the R410A rotary compressors. Jin-dong Kim et al. (1998) have measured the surface vibration and sound intensity and used the modal analysis techniques to identify the noise sources, made it possible to predict the relationship between the noise and the accumulator. K. W. Yun (1998) defined the functions of an accumulator, listed major design parameters and their common ranges, introduced a couple of major accumulator designs with an enhanced function.

In this paper, the test observations for the accumulators were presented to demonstrate the contribution. Based on the results, the dominating noises in R410A air-conditioning rotary compressors were the resonance noise of suction flow noise of the accumulator from 425Hz to 475Hz.
The finite element method (FEM) and SYSNOISE were used in acoustic analysis. Noise signal process program was developed based on MATLAB. Important dynamic and acoustic characteristics of the accumulator were also analyzed and estimated. To be effective in reducing noise in R410A compressors, a comprehensive understanding on the vibro-acoustic characteristics of compressors needs to be undertaken.

2. NOISE AND VIBRATION TEST RESULTS

The sound and vibration tests were carried out in the semianechoic room with R410A rotary compressor. Figure 1 shows the A-weighted spectra of the averaged sound level at 1 meter from the compressor center with the compressor operating at ARI condition. It is found that R410A rotary compressor is louder than that using R22. The maximum sound level peaks are located in the frequency range 425Hz-475Hz, as shown in Figure 1. Therefore, it is the necessary to consider the frequency range of R410A rotary compressor in order to reduce noise.

Accumulator vibration was also measured. A comparison of vibration amplitude of accumulator for R22 and R410A compressors is shown in Figure 2. The vibration amplitude in R410A rotary compressor is larger than those in R22 in Fundamental and Harmonic Frequency. An obvious difference is the large increase in 450 Hz band, which is a primary contributor to the noise of R410A compressor. In addition, the following phenomenons are observed in the tests.

- The vibration amplitude in the frequency band from 425 Hz to 475 Hz increases significantly whiles it close to the accumulator of R410A compressors.
- The variation tendency of R410A compressors in the frequency band from 425 Hz to 475 Hz is consistent with that of R22 compressors in the frequency band from 387.5 Hz to 434.4 Hz.
- Natural frequency of the accumulator cavity using R410A is higher than that using R22.

These phenomenons can be confirmed in the vibration amplitude as shown in Figure 2. The dominating noises from 425Hz to 475Hz in R410A rotary compressors are related to the vibration of the accumulator.

3. MODAL ANALYSIS

The finite element method (FEM) and SYSNOISE software are used in acoustic analysis. Noise signal process program is developed base on MATLAB software. The effects of accumulator to the total noise are analyzed from two aspects of theory and experiment.

3.1 Structural Model and Cavity Model of Accumulator
The model of the structural and the accumulator cavity model are shown in Figure 3. Because of the influence of the filter screen is little at low frequencies, the filter screen can be cut off when the structural and cavity analysis of accumulator is done.

Additionally, since refrigerant flow always contains lubrication oil, oil is also collected in accumulator as refrigerant/oil mixture enters the accumulator chamber. So the cavity model of accumulator must cut off the volume of oil when the cavity analysis is carried out as shown in Figure 3.

3.2 Acoustic Models
A model of accumulator was established based on three-dimensional object modeling software, and then used the FEM software Ansys for the mesh generation of the accumulator model, and the acoustics analysis software SYSNOISE was used for calculating and analyzing the characteristics of the accumulator model that using R410A, such as transmission loss, the acoustic modality of accumulator cavity. The gas properties of refrigerant at 18°C are shown in Table 1.

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The first four acoustic modalities of the cavity calculated by software SYSNOISE are shown in Figure 4. It was found that the frequency of first acoustic modality of the cavity that cut off the volume of oil was 456 Hz. It is very close to the test. The second acoustic modality is complex modal synthesis of accumulator and pipes. The frequency is almost two times comparing with that of the first acoustic modality.

Figure 5 shows the transmission loss of the accumulator calculated by three-dimensional object modeling software. Actually, the accumulator is an expansion chamber muffler with Inside Pipe, but it doesn’t like other general mufflers. As a result of the structural frequency is very low, it is easy to call the resonance of the structure. It is necessary to avoid the resonance while the noise reduce is considered.

### 3.3 Modal Analysis Using FEM

The FEM software Ansys was used to construct a dynamic model and to analyze characteristics of the accumulator model which was established based on three-dimensional object modeling software. Above all, reliable material properties, such as density and modulus of elasticity, are essential for accurate modal analysis. And the pipes are made from pure copper while the other parts of the accumulator are made form low-carbon steel. The material properties are shown in Table 2. First, establish models of all parts of the accumulator, mesh generation of the models, give the material properties, then used the Ansys coupling technology for assembling for all the parts in order to take the modal analysis. Figure 6 shows the model assembled based on the software Ansys coupling technology. And the filter screen also can be cut off for model simplification because of its influence is little at the interested frequencies.

Figure 7 shows the structural modals of the accumulator calculated by software ANSYS. There are three structural modals at frequencies below 1000 Hz. The first two structural modals at 312 Hz and 327 Hz are basically associated with the swinging motion of the L-tube (the suction pipe) in the accumulator. The frequency of the third modal is 396 Hz. It is the vibration modal of the accumulator shell. Obviously, the frequencies of first three structural modals are very close to the frequency of the second acoustic modality of the cavity. Especially, the frequency of the third structural modal is exactly matching. As a result of that, it has significant influence on acoustic response in the cavity. Additionally, the gas pulsations developed in the accumulator cavity may cause the resonance vibration of the L-tube as the frequency at 327 Hz is almost seven times comparing with the frequency of the compressor operation at 47.3 Hz.

## 4. ENHANCEMENT OF ACCUMULATOR

In this paper, the effects of accumulator to the total noise are analyzed from two aspects of theory and experiment in order to give some valuable suggestions for design of the accumulator. In one version, the enhancement is that one baffle plate is added in the accumulator to limit low-frequency vibration of L-tube and avoid the resonance.

Consequently, the structural modifications of the accumulator result in the reduction of noise level by approximately 14dB from 425Hz to 475Hz. And the total noise has reduced 1.5dB.

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**Table 1: Gas properties of refrigerant at 18°C**

<table>
<thead>
<tr>
<th>Gas Property</th>
<th>R410A</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas Density (kg / m²)</td>
<td>31.15</td>
</tr>
<tr>
<td>Speed of Sound (m / s)</td>
<td>176.91</td>
</tr>
</tbody>
</table>

**Table 2: The material properties**

<table>
<thead>
<tr>
<th>Material Properties</th>
<th>pure copper</th>
<th>low-carbon steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>density(Kg/m³)</td>
<td>8930</td>
<td>7900</td>
</tr>
<tr>
<td>modulus of elasticity(×10¹¹Pa)</td>
<td>1.27</td>
<td>2.0</td>
</tr>
<tr>
<td>Poisson ratio</td>
<td>0.33</td>
<td>0.25</td>
</tr>
</tbody>
</table>
5. CONCLUSIONS

An accumulator is more than just a component in R410A rotary compressor. Its contribution to the total noise is substantial. The characteristics of the accumulator are analyzed by using the software such as Ansys, MATLAB and SYSNOISE. Based on the experimental trials and analytical approaches, it shows that the dominating noises in R410A rotary compressors are the resonance noise of suction flow noise of the accumulator from 425Hz to 475Hz.

The noise reduction for the accumulator is tried by adding one baffle plate in the accumulator to limit low-frequency vibration of L-tube and avoid the resonance. After the modification, the noise level of the accumulator from 425Hz to 475Hz is reduced nearly 14dB. And the total noise has reduced approximately 1.5dB.

REFERENCES


Figure 1: Averaged sound level at 1 meter (ARI condition)
Figure 2: A comparison of vibration amplitude of accumulator for R22 and R410A compressors

(a) R410A                                                                                      (b) R22

Figure 3: Structural Model and Cavity Model of Accumulator

(a) Structural Model                     (b) Cavity Model of Accumulator

(a) First acoustic modality 188 Hz

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(b) Second acoustic modality 456 Hz

(c) Third acoustic modality 570 Hz

(d) Fourth acoustic modality 899 Hz

Figure 4: The first four acoustic modalities of the cavity
Figure 5: the transmission loss of the accumulator

(a) The mesh model

(b) The sectional of the mesh model

Figure 6: The assembled model of the accumulator (the ring parts mean coupling restraint)
(a) First structural modal 312 Hz

(b) Second structural modal 327 Hz

(c) Third structural modal 396 Hz

Figure 7: The structural modals of the accumulator