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SOURCE IDENTIFICATION AND REDUCTION OF NOISE FOR THE OUTDOOR UNIT OF ROOM AIR CONDITIONER

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ABSTRACT

The noise sources in the outdoor unit of room air conditioner are identified by the sound intensity method. The main noise is classified into two sources, air-born noise and structure-born noise. First, the air-born noise is reduced through the design of new fan and shroud, the reduction of the system resistance by rearrangement of heat exchanger, and the optimization of the complex parameter between the fan and the shroud. Next, in order to reduce the structure-born noise, the new shape of compressor mount and soundproof material were applied. As a result, the overall noise was reduced by 4–5dB(A).

1. INTRODUCTION

As the concern about the noise of household electric appliances is rising and so many people use the air conditioner, the requirements for noise reduction is necessarily on the increase and the noise level is competitively lower and lower.

The main noise sources in the outdoor unit of RAC (Room Air Conditioner) are from two components, which one is air-born noise and the other structure-born noise. The air-born noise is about 40–50% distribution of total noise energy, and has low frequency characteristic and broad band spectrum. This is mainly related to axial fan, interaction between the fan and the shroud, and system resistance by heat exchanger (condenser) and flow path. Structure-born noise is about 50–60% distribution of total noise energy, and is characterized by discrete frequencies which would stimulate and annoy the hearing sense. The main sources of this are the noise radiating directly from compressor shell and secondhand noise from vibration of sub-structures by the compressor. The fundamental step to reduce noise is basically to reduce each component noise of the system, for examples to reduce the noise of fan and compressor itself without performance deterioration. This paper adjusted the focus mostly into how to obstruct or enclose the noise effectively from the various sources by experimental process in view of system engineer.

In the first place, sound source identification of the rotary compressor and the outdoor unit of RAC was carried out by two microphones sound intensity method, so the effective process for the noise reduction could be set up. In order to reduce the air-born noise, several experimental ways was executed. New types of fan and shroud were designed to carry out the adequate performances, such as noise, flow rate and static pressure. And the relative distance between fan and shroud was optimized, and the system resistance that would be associated with flow loss and then decide the flow rate was decreased by rearrangement of heat exchanger. The reduction of structure-born noise was studied through the isolation of compressor noise, which included set-up angle of compressor and position of the soundproof material, as well as the shape and materials of the compressor mount that was made up of natural rubber.

2. NOISE SOURCE IDENTIFICATION

The consideration of noise source distributions is important to reduce the total noise speedily and effectively, because the reduction process depends on which component contributes more.

Fig.1 represents the noise distribution about the two split types, one is RAC and the other is packaged air conditioner (PAC), and the other wall type air conditioner (WAC). In the case of PAC and WAC, the air-born noise dominantly contributes to total sound energy, because these are in need of much more flow rate in comparison with RAC. However, it is necessary to reduce air-born and structure-born sound at the same time because two components have nearly same energy distribution in RAC.
Sound intensity level (SIL) was measured on the lateral face of outdoor unit of RAC and rotary compressor. Sound intensity that uses two microphones is generally more useful to identify the noise sources than sound pressure level that uses one microphone, because it is able to measure vector quantity such as magnitude and direction of sound. Sound intensity means sound power per area and is also expressed multiplication of pressure and particle velocity as follows.

\[
\text{SIL} = \frac{\text{Power}}{\text{Area}} = \frac{\text{Energy}}{(\text{Area} \times \text{Time})} = \frac{\text{Force} \times \text{Distance}}{(\text{Area} \times \text{Time})} = \text{Pressure} \times \text{Velocity}
\]

The pressure is measured by one microphone, but the particle velocity is calculated by pressure gradient from the Euler's Equation using two microphones, because it is very difficult to measure directly by one. Sound intensity level (SIL(\omega)) in frequency domain is acquired from imaginary part of cross power spectrum (G_{AB}) from two microphones, particle density (\rho), frequency (\omega), and gap (r) between microphones.

\[
\text{SIL(\omega)} = -\frac{1}{\rho\omega r} \text{Im}(G_{AB})
\]

Fig.2 shows the results of SIL contour surrounding the rotary compressor in 1/1 octave band. In the center frequency (Cf) 500Hz - 2000Hz range, there are high noise sources in the near of right and left side of accumulator where two phase refrigerant from condenser would be separated into liquid and gas which is desirable to inflow to compressor. The noise sources in the range above Cf 2000Hz exist on the just below the injection pipe of the accumulator. Therefore, it is suitable to setup the soundproof materials around accumulator in order to obstruct and enclose the radiating noise from compressor.

Fig.3 shows the SIL contour in the direction of front, rear and left side around the old model of outdoor unit. In the case of front side, the contour at Cf 500Hz - 8000Hz represents that the noise sources exist widely near around the fan, shroud and compressor. As the frequency is higher, the shape of noise distribution is more similar to that of shroud because the lower frequency noise is directly radiated through the wall (in this case, front panel) from the compressor, but the higher frequency noise is blocked up by the wall. And there are relatively a little high noise sources in the right part of shroud and in the bottom side near the compressor.

From this result, it is necessary to improve the shroud shape and obstruct the compressor noise properly.

The sound intensity contour at the rear side shows that lower frequency noise such as 500Hz band is broadly around the condenser where air inflows. This is because of inlet flow noise from fan and heat exchanger in view of experiences. This means that the low frequency noise is associated with system resistance, because fluid noise related to heat exchanger mostly depends on the pressure drop, for example, the less the pressure drop, the more the flow rate goes up. Most of noise sources are on the right side of compressor, and this is also left side of grill. This noise sources is radiated from compressor through the holes of grill.

The experimental results by sound intensity about the rotary compressor and outdoor unit of RAC indicate that in the center frequency region 500 - 1000Hz, air-born noise sources exist near the heat exchanger and outside of the shroud, in the high frequency region over that, there are high noise sources in the vicinity of inlet pipe and lower part of the compressor, and on the right and left side of accumulator.

3. REDUCTION OF AIR-BORN NOISE

Fan noise in view of frequency characteristics breaks into two parts, one is broad band noise which is random turbulent noise, the other discrete frequency noise which is pure tone and peak noise. The discrete frequency noise is compose of blade passing frequency (BPF) and its harmonics which is able to be calculated from blade numbers and rotating speed, and this sometimes annoys the human hearing sense. However this peak noise would occasionally be ignored, because the 1st BPF is under the 100Hz in case of outdoor unit of RAC and insensitive frequency region to human.

For the reduction of the broad band noise with high acoustic energy density, we tried to design new fan with low noise and high flow rate, minimize system resistance by heat exchanger, optimize the parameters related to the shroud, and consider the system matching. The system matching means the appropriate combination of
system resistance and aerodynamic performance of fan.

Fig. 4 shows the relation between system resistance and PQ (P: pressure, Q: flow rate) curve what one called the aerodynamic performance curve of fan. (a) means the aerodynamic performance of fan system including shroud in specific rotating speed, where the pressure and flow rate is in inverse proportion. The curve in (b) means the pressure drop so called system resistance between suction and discharge of the resistant system including heat exchanger and flow path, and is in proportion. The curves in (c) represents that when the fan-shroud system with the PQ curve is applied to the resistant system with the pressure drop, the intersection point is called the operating point (OP) in which the flow rate and static pressure of total system is determined.

Fig. 5 indicates the very easy and fundamental ways to increase the flow rate of the total system. In matters in relation to air-born noise reduction, the flow rate and noise level is in correlation and alternative, because the noise would be reduced as much as flow rate be increased. One is to reduce the system resistance as (a), and the other to rise the fan performance as (b). As you know in (a) if the system resistance (Rs) is decreased in case of same fan system, the operating point will be moved from OP1 to OP2 and then the flow rate will be increased from Q2 to Q1. In spite it is very difficult to level up the PQ curve of fan in same conditions, most of fan designers used to focus their studies on this and so it takes much time to get the magnificent results. Therefore it is advisable to choose the speedy and appropriate way to the engineer of these.

In addition, another way to consider is system matching in the relation to system resistance and flow rate. There is a boundary or a limit to increase the fan performance and to decrease the system resistance continuously. Fig. 6 represents the appropriate combination of two components, in other words the fan with high static pressure and low flow rate is applied to the low resistant system as (a), and the fan with low static pressure and high flow rate to the high resistant system as (b). Although designing the fan to be used for both high and low resistant system is very difficult, It is easier to design the fan suitable for the resistant system alternatively. Therefore system matching is sometimes effective to increase the flow rate and then reduce the air-born noise.

Fig. 7 shows the experimental results of system resistance according to the various domestic and foreign products. The principal parameters to influence the system resistance is the heat exchanger, and its area, depth, tube diameter, fin shape, installation of form, and etc. In this study because the area, the depth, the tube diameter and etc. of heat exchanger would directly influence the cooling capacity, rearrangement of installation was under consideration. The injection flows to the heat exchanger in the outdoor unit of RAC are from rear and left side. The air from rear side runs directly into the heat exchanger and fan in straight line with discharge flow, but the air from left side runs perpendicularly and is bent to the direction of discharge flow. Therefore the former has high air velocity, on the contrary the latter runs slowly by the fluid loss.

The system resistance was reduced to 50% by installation Improvement of heat exchanger that transfers the air quantity from left to rear side with same area of heat exchanger. This resulted in another advantage that the outline of outdoor unit could be thin and slim. The reduction of system resistance about 50% is the same result as 10% increase of flow rate or 2.0 - 2.5dB(A) noise reduction.

Noise source also exists near the outside of shroud from the SIL result about outdoor unit. Three parameters about shroud were also considered to reduce the air-born noise: The shape of shroud was improved by visualizing and analyzing the turbulent intensity of the flow near the shroud by the LDV (Laser Doppler Velocimeter) equipment, consequently the noise was reduced by 1.0dB(A).

Fig. 8 displays the noise variation along the parameters of shroud at the same flow rate. The relative distance between fan and shroud was optimized as (a), and the depth, one of the specific parameters of the shroud, was also optimized as (b). The noise was reduced by 1.0dB(A) through these optimizations.

In addition to these studies, new type fan was developed to be suitable for the improved system and so the noise was reduced by 2.0dB(A). From the above mentioned results, the air-born noise was reduced effectively by 4 - 5dB(A), and the improved noise spectrum was compared with old one in Fig. 9.

4. REDUCTION OF STRUCTURE-BORN NOISE

The effective enclosing ideas were proposed for the noise reduction of structure-born noise that was the discrete frequency noise in the outdoor unit of RAC by rotary compressor.
The rubber mounts (so-called seat rubber) support the compressor and prohibit the vibration of compressor from transmitting to the connected sub-part, and are very essential to protect secondary structure-born noise. In general the studies about these mounts have been focused on the avoiding the resonance of the compressor driving frequency with natural frequency, the minimizing the transmissibility of vibration of the compressor and non-linear structure analysis. In this study we considered the various shapes and materials of the mounts and the vibration level was reduced fairly, but the results did not meet expectations because the structure-born noise by the secondary vibration did not seem to contribute greatly to total acoustic energy.

The shape of mounts was redesigned not to reduce the transmissibility of the vibration, but to enclose the radiating noise from the bottom of the compressor that was identified to be noise sources by SIL. And to obstruct the radiating noise from the both right and left side of the accumulator, the installation position of the accumulator was rotated to the corner of the outdoor unit, and the noise was enclosed sufficiently by the panels. As the result, the radiating noise from the compressor was reduced by 2.0 ~ 2.5dB(A).

Another way to reduce the noise is making the most of the soundproof material that absorbs and encloses the noise. Soundproof material is almost always used for the most of outdoor unit of RAC because it is very easy and a little inexpensive to apply and appropriate to obstruct the radiating noise from the compressor. The noise would be reduced as much as the absorbing and enclosing capacity of the material, but this capacity has a boundary and is associated with cost. The installation method of the material was improved so as to obstruct the radiating noise from the bottom and the top of the compressor. And it is also effective to make air layer between the compressor and the material because the noise diminishes naturally and the material is separate from direct influence of compressor vibration.

Fig. 10 shows the noise reduction of outdoor unit of RAC to apply the experimental outputs about air-born and structure-born noise. The broad band and the peak components of discrete frequency noise was reduced by 4 ~ 5dB(A).

5. CONCLUSIONS

This paper does not deal with the profound research but presents a little easy applications about air-born and structure-born noise reduction in the outdoor unit of room air conditioner. First of all, the noise source identification by sound intensity contour using two microphones was very useful as a landmark to set up the noise reduction process.

The air-born noise was decreased through the design of new fan and shroud suitable for the system, the reduction of system resistance, and system matching. The significant consideration is to obtain the much more increase of the flow rate in proportion to that of noise because the flow rate and noise are always alternative.

We focused the noise reduction process for the structure-born noise on the effective methods to enclose and obstruct the radiating noise from compressor. It is very useful to redesign and improve the soundproof material and the compressor mounts to help the noise reduction.

6. REFERENCES

1. Lewis H. Bell, Douglas H. Bell, "Industrial Noise Control", Marcel Dekker Inc., 1994
Fig. 1 Noise energy distributions of outdoor unit

Fig. 2 Sound intensity contour of rotary compressor

Fig. 3 Sound intensity contour of outdoor unit of RAC
Fig. 4 Flow rate at the operating point
(a) PQ Curve    (b) Rs    (c) OP

Fig. 5 How to get the high flow rate
(a) Reduction of Rs    (b) Progress of fan

Fig. 6 System matching problem
(a) In case of high Rs    (b) In case of low Rs

Fig. 7 Reduction of system resistance

Fig. 8 Noise variation along the parameter of shroud
(a) Distance between shroud and fan    (b) Geometric depth of shroud

Fig. 9 Spectrum of reduced air-born noise

Fig. 10 Spectrum of reduced total noise