Study on Noise from Air-Cooled Heat Exchanger by Compressor Pulsation

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Noise problems caused by gas pulsations in the tubes of air-cooled heat exchangers have become more serious with the advance of the techniques for fan noise reduction in air conditioners. The prediction and the reduction of noise caused by the gas pulsations were investigated. The predictions of sound radiation and tube vibrations of air-cooled heat exchangers were carried out using Statistical Energy Analysis (SEA). The value of the loss factor of gas pulsations becomes high, when two-phase flows occur in the tubes. Therefore, the values were investigated and used for prediction. For the exciting forces, the forces of pressure at U bends and in straight pipes of heat exchangers were used. Air conditioners developed using the above method generate very low noise levels and they have been well received by customers.

1. Introduction

The noise level of air conditioners has been rapidly reduced in both indoor and outdoor units. With regard to indoor units, the main reduction has been in the fan noise, so that it has now become more important to reduce noise from other sources. A wall mounted type air conditioner is shown in Fig. 1. Since the heat exchanger has many long tubes as shown in the drawing, it is hard to analyze its vibration from the wave propagation approach, and a technique from an energy approach such as Statistical Energy Analysis is necessary. In this study, vibration and noise of heat exchangers were predicted by SEA method. As exciting force of piping by pressure pulsation, the exciting force of a U-bent tube at the end of the heat exchanger, and the exciting force of a straight tube having fins were considered. The radiation of noise by tube vibration was calculated by using the radiation ratio obtained experimentally. Attenuation by divergence of pressure pulsation in the heat exchanger varies with the flow pattern of the refrigerant in the tube, and it was therefore determined by measuring the distribution of pressure pulsation in the heat exchanger. After inspection of the accuracy of this method of analysis, applications to actual appliances and low noise level structure were investigated.

2. Modeling of radiation noise by pressure pulsation

2.1 Types of pulsation exciting force

Exciting force by pressure pulsation comprises two types, as shown in Fig. 2. These are (1) exciting force acting on bending part and (2) exciting force acting on straight part. The first type is an exciting force generated because the vector sum of forces is not zero in the bending part, and the second type is a hoop force which attempts to swell the cylindrical part. In most conventional cases, only the exciting force acting on the bending part was considered. However, the heat exchanger to be analyzed has a very long straight part, the wall thickness of the tube is small, and deformation must be taken into account, for which reasons the radiation noise owing to exciting forces (1) and (2) was investigated in this study.

2.2 Radiation noise by pulsation exciting force in bending part

A model of a heat exchanger is shown in Fig. 3. A heat exchanger is generally composed of a number of circuits. The

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refigerant gas flowing into the heat exchanger is uniformly separated in the distribution area at the inlet; it then flows in each circuit, gathers together and flows out of the heat exchanger.

In this case, the pulsation exciting force $F_i$ of each bending part is expressed as follows.

$$F_i = \sigma A P_i$$

where,

$\sigma$: conversion efficiency of pressure pulsation-exciting force

$A$: section area of tube

$P_i$: pressure pulsation of each bending part

$i$: bending part number of each circuit

The piping length of each circuit is often more than 10 m. In such a case, the number of modes is 8 in 100 Hz band or 25 in 1,000 Hz band, and hence it is appropriate to model the vibration as an energy flow. In this study, the spatial mean square velocity of the heat exchanger surface is determined by SEA method, and the radiation noise is predicted. Supposing the attenuation by divergence of pressure pulsation in the tube to be $\exp(\alpha t L)$, the spatial mean square force $\langle F^2 \rangle$ of an entire heat exchanger at a certain frequency is given in the following equation.

$$\langle F^2 \rangle = N t_0 \sum_{i=1}^{n} F_i^2$$

$$= \frac{n_0 \sigma^2 A^2 P_0 \exp(2aL)[1-\exp(2akL)]}{[1-\exp(2aL)]}$$

where,

$n_0$: number of circuits

$P_0$: pressure pulsation at heat exchanger inlet

$\sigma$: attenuation factor of pressure pulsation

$L$: length between bending parts

$k$: total number of bending parts of circuits

The spatial mean square velocity $\langle V^2 \rangle$ of the heat exchanger piping surface is calculated in the following formula by using $\langle F^2 \rangle$.

$$\langle V^2 \rangle = \pi \langle F^2 \rangle N(O)/(2M^2 \eta \Omega)$$

where,

$N(O)$: number of modes in 1/3 octave band

$M$: mass

$\eta$: loss factor

$\Omega$: 1/3 octave band center frequency $\times 2\pi$

The power $W_i$ of radiation noise from the heat exchanger can be calculated as follows by multiplying $\langle V^2 \rangle$ by radiation ratio $\sigma$.

$$W_i = \rho c S \langle V^2 \rangle$$

where,

$\rho c$: characteristic resistance of the medium ($\rho$: density of air, $c$: sonic speed)

$S$: radiation area

The radiation ratio $\sigma$ of heat exchanger will be reported in a subsequent paper, and is omitted here. Regarding the radiation ratio, it is known that only the acoustic radiation from the copper tube should be taken into consideration, without regard to the acoustic radiation from the fins attached to the tubes.

2.3 Radiation noise by pulsation exciting force in straight part

The straight part can be calculated as follows. In (b) in Fig. 2, the displacement $x_0$ on the tube surface is:

$$x_0 = P_0/\left[\rho h (\omega_0^2 - \omega^2)\right]$$

where,

$\omega_0$: ring circular frequency $[E/\rho_k(1-0.5\nu)]^{1/2}/R$

$E$: Young's modulus of tube

$\rho_k$: density of tube

$\nu$: Poisson's ratio of tube

$R$: radius of tube

$h$: wall thickness of tube

The radiation power $W_2$ using volume velocity $U$ and its spatial mean square volume velocity $\langle U^2 \rangle$ is

$$W_2 = \omega_0^2 \pi R L n_{\omega_0}/(4 \pi c)$$

and hence in the region of $\omega < \omega_0$ $W_2$ is determined as follows from equation (5).

$$W_2 = \frac{\rho c R^2 (2 \pi R L)^2}{[1 - \exp(2akL)]/[1 - \exp(2aL)]}$$

3. Fundamental experiment

3.1 Measurement of attenuation constant of pressure pulsation

For calculation of radiation power, values of $\alpha$ and $\sigma$ are needed. The flow in the heat exchanger is a two-phase flow, and $\alpha$ is different from that in a single-phase flow. In this study, it was determined by an experiment using a heat exchanger. The test method is shown in Fig. 4. The compressor was a reciprocating compressor having a large pressure pulsation, the pressure pulsation was measured at the end of the heat exchanger, and pressure pulsation distribution.

Fig. 4 Experimental apparatus used for exciting test

Using a compressor of a large pressure pulsation, the heat exchanger is excited, and the generated vibration and noise are measured.

Fig. 5 Pulsation distribution in an air-cooled heat exchanger

The pressure pulsation of bending parts is measured, and the damping coefficient of pulsation is determined from this curve.

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Table 1 Specifications of test air-cooled heat exchangers

<table>
<thead>
<tr>
<th>Heat exchanger</th>
<th>Area (m²)</th>
<th>Aspect ratio</th>
<th>Number of circuits</th>
<th>Number of stages</th>
<th>Number of rows</th>
<th>Tube diameter (mm)</th>
<th>Tube wall thickness (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>No.1</td>
<td>0.416</td>
<td>4.61</td>
<td>2</td>
<td>7</td>
<td>3</td>
<td>9.52</td>
<td>0.35</td>
</tr>
<tr>
<td>No.2</td>
<td>0.189</td>
<td>5.99</td>
<td>4</td>
<td>7</td>
<td>3</td>
<td>9.52</td>
<td>0.35</td>
</tr>
<tr>
<td>No.3</td>
<td>0.322</td>
<td>5.15</td>
<td>5</td>
<td>10</td>
<td>3</td>
<td>9.52</td>
<td>0.35</td>
</tr>
<tr>
<td>No.4</td>
<td>0.178</td>
<td>2.76</td>
<td>4</td>
<td>10</td>
<td>2</td>
<td>9.52</td>
<td>0.35</td>
</tr>
<tr>
<td>No.5</td>
<td>0.178</td>
<td>2.76</td>
<td>6</td>
<td>10</td>
<td>3</td>
<td>9.52</td>
<td>0.35</td>
</tr>
<tr>
<td>No.6</td>
<td>0.083</td>
<td>1.03</td>
<td>4</td>
<td>32</td>
<td>2</td>
<td>9.52</td>
<td>0.35</td>
</tr>
<tr>
<td>No.7</td>
<td>0.161</td>
<td>2.36</td>
<td>1</td>
<td>10</td>
<td>2</td>
<td>9.52</td>
<td>0.35</td>
</tr>
<tr>
<td>No.8</td>
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<td>1.37</td>
<td>2</td>
<td>20</td>
<td>1</td>
<td>9.52</td>
<td>0.35</td>
</tr>
</tbody>
</table>

(Attenuation by divergence) in the heat exchanger was determined. Thermocouples were attached to bending parts, and the onset situation of two-phase flow was measured. Fig. 5 shows the results of measurement of pressure pulsation distribution in the case of heat exchanger No. 7 in Table 1. The attenuation characteristics varies significantly between the case of not generating two-phase flow without blowing air and the case of generating two-phase flow by blowing air. The attenuation factor \( a \) is about \(-0.15\) in the former case and about \(-0.50\) in the latter case, or about three times larger. It is known that the attenuation factor is low in frequency dependence. These values were used in prediction.

3.2 Measurement of conversion efficiency of pressure pulsation-exciting force

Using the apparatus shown in Fig. 4, the spatial mean square velocity of the heat exchanger tube and inlet pressure pulsation were measured. When these values and the value of \( a \) in paragraph 3.1 are known, the pressure pulsation-force conversion factor \( \sigma' \) can be obtained. This is expressed as follows from equations (2) and (3).

\[
\sigma' = \frac{2 M^2 \omega Q <V^2>}{\pi N(\omega) n_u A_f P_s^2} \tag{7}
\]

The experiment was conducted using heat exchangers having the specifications shown in Table 1. The spatial mean square velocity of the heat exchanger surface was calculated from the acceleration at five surface positions. The diagram has been omitted because of limited space, but the logarithmic value of the spatial mean square velocity divided by the pressure pulsation \( P_s \) at the heat exchanger inlet was a straight line upward to the right with an inclination of 20 log in each heat exchanger. The result of determining \( \sigma' \) from equation (7) is shown in Fig. 6. The value of \( N(\omega) \) necessary for determining \( \sigma' \) can be easily obtained by using a theoretical equation for determining the beam mode number from the wave propagation approach or a equation obtained by experiment [the mode number is specifically described in the reference materials relating to solid-borne noise, and is omitted herein; see references (2) and (4)]. It has been ascertained that \( \sigma' \) is a constant value regardless of frequency. The cause of an error of \( \pm 10 \) dB is that the boundary treatment differs in each heat exchanger and that the temperature is not uniform among circuits. For prediction, this mean value was used.

4. Prediction and verification

The prediction equation in chapter 2 was programmed in order to calculate the noise level from 50 Hz to 10 kHz (the total power level of \( W_1 \) and \( W_2 \)). The radiation ratio of \( W_1 \) was the value already used in prediction of solid-borne noise of heat exchangers, as mentioned above. In the experiment, as shown in Fig. 4, the heat exchanger was installed in a reverberant room, and the compressor was installed in a different room.

In addition, to prevent the compressor vibration from propagating to the heat exchanger, flexible tubes were attached to the inlet and outlet of the heat exchanger. Furthermore, to make the blower noise less than the noise generated by pulsation excitation, and to generate a two-phase flow in the heat exchanger in the same state as in an actual system, the air flow rate of the blower and the temperature in the reverberant room were adjusted during the experiment.

Prediction and experiment of heat exchanger No. 5 in Table 1, which is used in the large cassette type air conditioner made by Mitsubishi Heavy Industries, Ltd., are compared in Fig. 7. The result showed the prediction and experiment coinciding very well from low frequency to high frequency. Similar results were obtained in other heat exchangers, and it was confirmed that this method of prediction is valid in a heat exchanger larger in modal density.

5. Review

5.1 Contribution of pulsation exciting force

By using this method of prediction, the magnitudes of exciting force in the bending part and exciting force in the
straight part were studied. In an example of applying a unit input of pressure pulsation to the heat exchanger of No. 7 in Table 1, at a frequency of 3 kHz or less, the radiation noise by exciting force in the bending part was greater by 20 dB or more. At extremely high frequencies, it is known that the contribution of exciting force of the straight part is very large. In rotary compressors or scroll compressors used in actual air-conditioning, the pulsation components are greater at lower frequencies, and the actual problem is believed to be almost entirely attributable to radiation noise by exciting force in the bending part.

5.2 Tolerance of pressure pulsation

The blowing sound of an air conditioner is a random noise in a broad band, and hence belongs acoustically to sound of favorable quality. Therefore, the noise by pulsation excitation, electromagnetic noise from the fan motor, and the noise of refrigerant flow are usually designed to be masked by this blowing sound. By reverse analysis of this prediction method, the value of pressure pulsation that is required in order to be below the blowing sound was studied. The relevant diagram is omitted, but it was found that a smaller pressure pulsation is required at higher frequencies owing to the characteristic of the blowing sound in all air conditioners both large and small, and it was necessary to keep the pulsation value at around 0.1 kPa at 1 kHz in small-sized indoor units where a low noise level is required.

5.3 Low noise structure

From the theoretical equation in chapter 2, as measures for reducing the radiation noise by pulsation excitation, the following methods may be considered, aside from reduction in the incoming pulsation level by means of a silencer.

(1) In a structure large in aspect ratio, the number of bent tubes is decreased.
(2) By damping at the heat exchanger end, $\eta$ is increased.
(3) The tube diameter of the heat exchanger is reduced.

Testing of these methods (1), (2), (3), confirmed that all are effective means of reducing radiation noise by pulsation excitation. They are suited to application in actual products, and contribute to further lowering the noise level of air conditioners.

6. Conclusions

The analysis technique by SEA method was applied in prediction of noise by pulsation excitation in heat exchangers of air conditioners, and the following results were obtained.

(1) By applying the exciting force acting in the bending part of a heat exchanger and the exciting force acting in the straight part to an SEA vibration model of a heat exchanger, it is possible to predict the vibration and radiation noise caused by excitation of a heat exchanger as a result of the pulsation generated in a compressor.

(2) In a heat exchanger high in the vibration mode number in each frequency band, it was verified that analysis is possible with practical precision.

(3) In actual air conditioners, except for the high frequency region over several kHz, the exciting force acting in the bending part was found to be a dominant factor of radiation noise.

(4) By reverse analysis of this technique, study of tolerance of pressure pulsation and study of low noise structure of heat exchangers are known to be possible.

In future, the constants of this method of prediction obtained by experiment will be determined analytically and improved so as to be applicable to all heat exchangers.

References


