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EXPERIMENTAL STATISTICAL ENERGY ANALYSIS APPLIED TO A ROLLING PISTON-TYPE ROTARY COMPRESSOR

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ABSTRACT

Hermetic rotary compressors are one of the most important components in air-conditioners and refrigerators since they control both the performance and noise level of these products. Noise and vibration control of hermetic compressors should start with the identification of their sources of noise and vibration since the noise sources within a compressor play a determining role in the noise radiation from the compressor shell. Many approaches have been used to identify compressor noise sources in the past. However, compressor noise source identification has proven difficult since the characteristics of compressor noise are complicated due to the interaction of the numerous components within the compressor. In particular, a compressor possesses a number of partially coherent excitation mechanisms within its small inner space. The fact that several mechanisms operate simultaneously makes the identification of the source of compressor noise more difficult since noise is generated both by pressure pulsations within the refrigerant gas and by the compressor body vibration due to unbalanced internal dynamic forces. Therefore experimental Statistical Energy Analysis has been investigated as a noise control tool for the development of low noisy hermetic-type compressors since that procedure can be used to trace energy flow through the system and identify transmission paths from the noise source to the exterior sound field.

INTRODUCTION

For noise source identification and sound level evaluation, Statistical Energy Analysis (SEA) has been applied recently to design problems in the HVAC industry since it can quantify the flow of energy through compressor components and trace their noise transmission paths. In particular, SEA can be a modeling procedure for the theoretical estimation of the dynamic characteristics of the vibrational response and the noise radiation from complex structures using energy flow relationships. Therefore SEA can be used to predict the interactions between resonant structures and reverberant sound fields in acoustic volumes. Theoretical SEA, however, has a limitation when it is applied to small, hermetic-type compressors since the individual compressor components may have insufficient modes in the frequency bands of interest. For this reason, it is necessary to use Experimental Statistical Energy Analysis (ESEA) to circumvent some of the limitations of theoretical SEA.

In the work described in this paper, the main parameters of ESEA models, such as modal density, damping loss factor, and coupling loss factor between the individual components of the rotary compressor, were obtained experimentally. The modal densities were measured by the mobility method and the damping loss factors by the half-bandwidth method. For the evaluation of the sound radiation from the compressor body, the radiation ratio was estimated from vibration measurements on the compressor surface and sound intensity measurements in an anechoic room.

MODAL DENSITY

The vibrational and acoustical response of structural elements, and the acoustical response of volume elements, to random excitations are often determined by the resonant response of contiguous structural and acoustic
modes. For example, when a structure is excited by some form of broadband structural excitation, the dominant structural response is resonant. When a structure is also acoustically excited, the dominant response is generally resonant [Norton]. Therefore the energy flow between groups of resonant modes can be a primary concern in the SEA method. The modal density is therefore a very important parameter for establishing the resonant response of a system to a given forcing function. The modal density indicates the number of modes per unit frequency and is analogous to the thermal capacity of a thermal system. Asymptotic modal density formulae are available in the literature for a range of idealized subsystems such as bars, beams, flat plates, thin-walled cylinders, acoustical volumes, etc. However, theoretical estimates are not readily available for non-ideal subsystems such as compressor components. For this reason, in practical engineering problems an experimental technique is more suitable for obtaining modal densities. To establish an appropriate modal density estimation procedure, comparisons between analytic solutions and experimental measurements for ideal models such as beams and plates were performed. After these verification tests, experimental measurements for various compressor components were performed.

Experimental Method

In SEA, the spatial and time averages of a variety of different signals, rather than their instantaneous values, are considered. Therefore it is necessary to measure the vibration at numerous locations and over some specific time interval to obtain an averaged vibration level. In this experiment, this spatial average was obtained by an arithmetic average of the responses at several measurement points.

Point mobility measurement

Mobility is a complex frequency response function that is commonly referred to as a transfer function between an output velocity and an input force. The real part of the mobility is a function of the modal density of the structure, and the real part is of primary concern since it represents the mean energy flow that can be dissipated. The imaginary part of the point mobility represents the reactive energy exchange in the region of the coupling point. Therefore the imaginary part is not considered when the modal density is considered. The modal density can be defined as,

\[ n(\omega) = 4S\rho_s \text{Re}[Y(\omega)] \]  

(1)

where: \( S \) is the surface area of the test structure, \( \rho_s \) is the surface mass (mass per unit area), and \( \text{Re}[Y(\omega)] \) is the space-average of the real part of the mobility.

Test results of modal density

Plate and beam

A flat plate and beam were chosen to verify the experimental procedure since the analytical solutions of these components are presented in many references: e.g., [Lyon]. The modal density of a flat plate and beam are shown in Figures 1 and 2, respectively. There is good agreement between experimental and analytical results. The analytical solution for the plate was obtained based on flexural vibration modes and it has a constant value. The modal density of the beam, however, is inversely proportional to the frequency.

Compressor shell

Among the compressor components, the modal density of the shell is presented here since its shape is similar to a theoretical thin-walled shell. Figure 3 shows the experimental, analytical, and ANSYS (i.e., finite element) results for this case. The difference between the experimental and analytical solution shows that the analytical solution below the ring frequency (the frequency at which the cylinder vibrates uniformly in the breathing mode) is not suitable for describing the compressor shell. The compressor shell ring frequency was 14 kHz. To verify this difference, the numerical modal density was calculated using ANSYS at the locations of the experimental measurements. The ANSYS result agrees well with the experimental data between 2 kHz and 10 kHz. Thus an ANSYS analysis based on thin shell theory agrees well with the measured modal density of a compressor shell below the ring frequency.

DAMPING LOSS FACTOR

In general, damping exists in all real systems and there are many different types of damping in practice. The most commonly encountered damping types are structural (hysteretic) damping, coulomb (dry-friction) damping, and velocity-squared (aerodynamic drag) damping. Among them, the form of damping that is most
relevant to engineering noise and vibration control is structural damping and structural damping is of two types: viscous and hysteretic. Viscous damping is usually chosen for mathematical convenience, since it yields simple solution for transient response. Hysteretic damping, based on the concept of a complex modulus, can often be utilized in the calculation of steady state response. The principal difference between a viscous-damped system and a hysteretic-damped system is that for the viscous system the energy dissipated per cycle depends linearly on the frequency of oscillation, whereas for the hysteretic case it is independent of the frequency. Energy is thus conserved only for the particular case of free undamped vibrations since there are no excitation or damping forces present. However, when considering practical engineering components such as compressor components, there should be an energy dissipation due to the damping inherent to their material properties. Therefore viscous damping is assumed in order to estimate the damping loss factor of compressor components. The two most common techniques for obtaining damping loss factors are the half-power bandwidth technique and the envelope decay technique. The half-power bandwidth technique utilizes the standard half-power bandwidth relationship which is associated with a 3 dB drop in response from the associated steady-state frequency response function peak. The other, envelope decay (reverberation), technique is based on the logarithmic decrement of the transient structural response subsequent to gating of the excitation source. For the measurement of damping loss factors in compressor components, the half-power bandwidth technique is used: i.e.,

$$\eta = \frac{\Delta f}{f_n}$$

where: $\Delta f$ is the half-power bandwidth and $f_n$ is the resonance frequency of the mode.

Damping loss factors of compressor components

The damping results for the compressor components are listed in Table 1 as percentages. Among the damping values for the compressor components, the highest value was found for the cylinder with an average value of 3. On the other hand, the lowest value was found for the top-cap at 0.34, perhaps because it is more stiff due to the attachment of the discharge pipe and the electric adaptor. When we compare the damping loss values at a specific frequency of 4 kHz, the maximum value was 1.57 for the cylinder and the minimum was 0.27 for the top-cap. Among the other components of the compressor, the average shell damping was 1.05, the average bottom-cap damping was 1.15 and the average accumulator damping was 1.27. At 4 kHz, the shell damping was 0.69, the bottom-cap damping was 0.9 to 1.0, and the accumulator damping was 0.65 to 0.75.

**COUPLING LOSS FACTOR**

The concept of the coupling loss factor is unique in SEA and it describes the link between two coupled subsystems. The coupling loss factor therefore determines the degree of coupling between the two subsystems. If a coupling loss factor is less than a damping loss factor, then the subsystems are described as being weakly coupled. It is always desirable to select the subsystems so that they are weakly coupled. There is no single way to evaluate the coupling loss factor experimentally or analytically. In particular, the couplings between different structural elements are very complicated because of the different types of wave motions generated at the discontinuity. For the calculation of coupling loss factors of compressor components, modal densities, damping loss factors, and energies of each subsystem are used: i.e.,

$$\frac{E_2}{E_1} = \frac{n_2 \eta_{21}}{n_1 \eta_2 + n_1 \eta_{21}} = \frac{n_2 \eta_{12}}{n_2 \eta_2 + n_1 \eta_{12}}$$

where: $E_1$ and $E_2$ are the energies of subsystems 1 and 2, $n_1$ and $n_2$ are the subsystem modal densities, and $\eta_1$ and $\eta_2$ are the subsystem damping loss factors. By using the reciprocity relation, $\eta_{21} = n_1 / n_2 \eta_{12}$, the final coupling loss factor between subsystems 1 and 2 can be written as

$$\eta_{12} = \eta_2 \frac{n_2 E_2}{n_2 E_1 - n_1 E_2}.$$
Coupling loss factors of the basic SEA components

As a preliminary test, the coupling loss factor of a plate-plate subsystem was obtained. When a force is applied either by an impact hammer or by a shaker through a force transducer, the velocity of each subsystem is measured by an accelerometer in order to calculate the subsystem energy. The measured coupling loss factor of a flat plate–plate joint shown in Figure 4 agrees well with the analytical solution. At a junction between two components having the same geometry and composition, the coupling loss factors 12 and 21 are exactly same since the modal densities of both subsystems are same. However, at a junction of different components, the coupling loss factors 12 and 21 are in general different.

Coupling loss factor of compressor components

Figure 5 shows the coupling loss factor between the shell and the cylinder. Here “shell-cylinder” means the energy flows from the shell to the cylinder. In the “cylinder-shell” case, the energy flows in the reverse direction. Figure 5 shows that energy is transferred well from the shell to the cylinder at all frequencies except 2000 Hz and 5000 Hz. The energy flow from cylinder to shell, however, increases with frequency up to 8 kHz. Therefore the vibration of the cylinder can be identified as one of the main vibration sources of the shell.

RADIATION FACTOR

Definition of radiation ratio

When a rotary compressor noise is evaluated in terms of radiated sound, the structural vibration of the shell and the accumulator play a very important role in determining the noise level since the sound pressure or sound power is determined by the interaction between the surface vibration of the compressor and the fluid medium around it. The radiation ratio is defined as that the ratio of actual radiation of sound energy (per unit time) compared with that from a plane surface of the same area and the same mean square normal surface velocity (averaged over both time and space) radiating perfectly: i.e.,

\[ \Pi = \pi^{-2} p_{\text{rms}} u_{\text{rms}} = \sigma \rho_0 c S \left( \frac{v^2}{\pi} \right) \]

where: \( p_{\text{rms}} \) is the root mean square radiated pressure at some point in space, \( u_{\text{rms}} \) is the corresponding root mean square acoustic particle velocity at the same point, \( z \) is the radius of the vibrating plane piston, \( < > \) represents a time average, \(-\) represents a space average, \( S \) is the radiating surface area of the structure, \( \rho_0 \) is the density of the fluid medium into which the structure radiates, \( c \) is the speed of sound in the fluid medium, and \( \sigma \) is the radiation ratio of the structure.

Acoustic radiation loss factors

The acoustic radiation loss factor of a structural element is given by

\[ \eta_{\text{rad}} = \frac{\rho_0 c \sigma}{\omega \rho_s} \]

where: \( \rho_s \) is its surface mass (mass per unit area), \( \rho_0 \) is the fluid density, and \( \omega \) is the center frequency of the band.

Experimental result of radiation ratio for compressor shell

It is very complicated to predict an analytical solution for a rolling piston-type compressor since a rotary compressor consists essentially of a body shell and an accumulator. This is similar to an object that is comprised of two noise sources. Therefore, it would be very useful if an experimental radiation ratio of a rotary compressor can be described based on the actual operating condition. For this measurement, measurements of sound intensity and surface velocity are performed since sound intensity can be interpreted as the amount of radiated sound power per unit radiating surface area.

Sound intensity and velocity measurement method

The two-microphone cross-spectral method was used to measure the sound intensity. At the same time, the velocities on the surface of the compressor shell and accumulator were measured by accelerometers while the
compressor was running under the ASHRAE-T test condition. The vibration was measured as shown in Figure 6 at 45-degree intervals on the shell surface and at 90-degree intervals on the surface of accumulator. The acceleration was also measured at five heights. The minimum measuring distance when estimating the sound intensity was chosen as 10 cm from the surface of a shell and accumulator as shown in Figure 7. Intensity data measurement points consisted of 5 intervals of height and 30-degree intervals in the circumferential direction.

Sound intensity and velocity measurement results

Among the data covering the full frequency range, only results at 1250 Hz, 1600 Hz, and 6300 Hz are presented here. The 1250 Hz component of velocity shown in Figure 9(a) shows that a high velocity value appears at the discharge hole side of a muffler. This velocity contour indicates that the main vibration component at 1250 Hz is generated by gas flowing from the discharge hole of the muffler. The velocity contour at 1600 Hz in Figure 9(b) shows the maximum vibration occurs at the bottom part of the accumulator. This component causes a dominant noise generation on the upper or the middle part of the accumulator which can also be seen in the sound intensity measurements. At 6300 Hz, the major vibration sources are located at around the welding points. These vibration results coincide with the sound intensity maps shown as Figure 8(a) to 8(c). The intensity level of 1250 Hz in the part of the compressor body between the middle and the bottom part. This result has a close relationship with the velocity distribution in Figure 8(a). The acoustic intensity associated with the accumulator is obvious at 1600 Hz in Figure 9(b). In particular, the 2000 Hz and 3150 Hz bands appear to radiate from the upper and the middle part of the accumulator. The intensity levels generated by the inner components of the compressor and by transmission through the welding points are shown at 6300 Hz in Figure 8(c). These results can be inferred from the fact that the welding points play an important role as a path of vibration transfer.

The radiation ratio and the radiation loss factor measurement result

The peaks of the averaged squared velocity level appear at 1000 Hz, 1600 Hz, and 3150 Hz as shown in Figure 10. They are due to the vibration of the lower part of an accumulator or the vibration around the muffler discharge hole. The average value of sound intensity is almost constant over the whole frequency range in Figure 10. The calculated radiation ratio and radiation loss factor in Figure 11 have the sharp peaks at 500 Hz, 1600 Hz, and 2500 Hz. This graph indicates that the rotary compressor has two main noise sources based on the compressor body and accumulator.

CONCLUSIONS

The objective here was to provide ESEA results for a rotary compressor as follows: (1) the modal density of the compressor shell illustrates the differences between the experiment and the theoretical results below the ring frequency. Therefore the analytical solution below the ring frequency is not accurate when used to describe the modal density of the compressor shell; (2) the highest damping value among the compressor components is the cylinder with the averaged value of 3.0%; and (3) the calculated radiation ratio and radiation loss factor show sharp peaks at 500 Hz, 1600 Hz, and 2500 Hz. These peaks show that the compressor body and the accumulator are the two main noise sources of the rotary compressor.

REFERENCES

Figure 6: Velocity measurement of compressor shell and accumulator

Figure 7: Sound intensity measurement

Figure 10: Averaged sound intensity and velocity square

Figure 11: Radiation ratio and radiation loss factor of rotary compressor
Figure 8(a): Sound intensity at center 1250Hz

Figure 8(b): Sound intensity at center 1600Hz

Figure 8(c): Sound intensity at center 6300Hz

Figure 9(a): Velocity at center 1250Hz

Figure 9(b): Velocity at center 1600Hz

Figure 9(c): Velocity at center 6300Hz