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ON THE CONTRIBUTION OF VIBRATING SHELL TO THE SOUND POWER LEVEL OF A VACUUM CLEANER

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Abstract.
The shell of a vacuum cleaner is essentially an acoustic enclosure surrounding the noise sources and the transmission paths. While the shell generally stops a significant amount of the noise transmitted to it from continuing to the outside, the shell will be respond to these inputs in a manner depend on the nature of the excitation and the structural dynamic characteristics of the shell. Thus, the contribution of the vibrating shell to the sound power level of the vacuum cleaner is considered important. In this study, the near-field sound pressure, the surface velocity and the sound intensity levels are measured. Then, by using these data, the shell structural borne sound power level is calculated and compared with the overall sound power level of the vacuum cleaner. The results presented in this paper show that the sound power level of a vacuum cleaner is depend on the frequency and distribution of the shell motion.

INTRODUCTION

In order to produce high quality vacuum cleaners and comply with EU directivity [1], the shell structural borne sound must be one of the most important concepts to be considered. Research to improve the performance of the unit at the design stage has been pursued. The study in reference [2] was concerned essentially with the vibro-acoustic behaviour of a wet and dry type vacuum cleaner.

The shell of a vacuum cleaner is essentially an acoustic enclosure surrounding the noise sources and the transmission paths. Thus, all the sound transmitted from vacuum cleaner to the outside is caused by mechanical vibration of the shell which can either be a rigid body motion or deformation of the shell. While the shell generally stops a significant amount of the noise transmitted to it from continuing to the outside, the shell will respond to these inputs in a manner dependent upon the nature of the excitation and the structural dynamics occur in terms of its natural frequencies, mode shapes, and damping coefficients. To avoid high amplitude response, the shell should be formed such that the natural frequencies of the shell are well away from the driving frequencies.

The sound generation mechanism is composed of three major parts: The excitation mechanism, the system response characteristics and the radiation conditions. The excitation results mainly from unbalance of the rotor at the rotational speed and the blade passing frequencies such that the six blades of the electric motor cooling fan and the twelve blades of the electric motor fan. The system response characteristics represent the entire transmission path including the dynamical behaviour of the shell. It is most conveniently described in terms of a frequency response function (FRF). The result of the product of the excitation and the system response characteristics is the shell vibration in the system response. The shell vibrates at each of the driving frequencies in a certain pattern that is determined by the dynamic characteristics of the shell and the nature of the excitation. Finally the shell vibration is converted into air borne sound. The amount of radiated sound depends on the vibration level of the shell and the radiation efficiency. Depending on the deformation pattern and the frequency, the radiation efficiency of each forced mode is different.

The sound radiation efficiency of a vibrating shell is dependent on the frequency and distribution of the shell motion. The frequency response of acoustic radiation from the shell is divided into two parts separated by the coincident frequency. The coincident frequency is the frequency at which the wavelength of sound in air is the same as the wavelength of the shell vibration pattern. For frequencies below coincident frequencies, the shell wavelength is shorter than air wavelength and thus adjacent parts of the shell which are vibrating out phase with each other will interfere, the radiation efficiency will be less than unity, and the shell will tend not to exhibit strong directional characteristics. For frequencies above the coincident frequency, the shell wavelength is longer than the air wavelength and thus adjacent parts of the shell will not interfere with each other, the radiation efficiency will be unity, and the shell will tend to exhibit radiation patterns which correlate with the pattern of shell motion.

In order to reduce the noise and vibration radiated from such units, it is necessary to locate accurately the dominant frequency components of noise and vibration and to trace their transmission paths and sources.

In this work a wet and dry type vacuum cleaner is investigated. The vacuum cleaner is driven by a 1600 W electric motor that includes the cooling fan with twelve blades. Speed of the motor is about 19.000 rpm (~315 Hz). The material of the vacuum cleaner shell is the moulded plastic. In the present work, contribution of vibrating surfaces to the sound power level of a sample vacuum cleaner were experimentally determined using the sound intensity, the near field sound pressure, and the surface velocity measurements. The objective of this paper is to find the effects of the vibrating surfaces on the total vacuum cleaner sound power.

THEORY

In electro-acoustic systems, the acoustic intensity is a vector quantity defined as a product of the acoustic pressure and the corresponding particle velocity at a given point. Acoustic intensity indicates the amount of energy passing through a unit area in a normal vector direction per unit time. Therefore, sound intensity is calculated by time averaging the product of the instantaneous sound pressure p(t) and the instantaneous particle velocity u(t), i.e.,

$$I = \overline{p(t)\,u(t)} \tag{1}$$

Where the bar indicates time averaging. Thus, if the sound pressure and the particle velocity can be measured, the sound intensity can be calculated. The sound pressure can be measured easily, the particle velocity can be approximated by making two closely spaced sound pressure measurements. For a unidirectional energy flow the equation of motion is

$$\rho\frac{\partial u(t)}{\partial t}+\frac{\partial p(t)}{\partial r}=0 \tag{2}$$

Where ρ is the density of air, u(t) is the particle velocity in the position r. If the acoustic pressure are $p_1(t)$ and $p_2(t)$ measured at two closely spaced points in the direction r, then the gradient of acoustic pressure is approximately represented as

$$\frac{\partial p(t)}{\partial r}=\frac{p_2(t)-p_1(t)}{\Delta r} \tag{3}$$

Where Δr is the separation between two points where the acoustic pressure is measured. This is valid as long as the separation is small compared with the wavelength of the sound being measured. The approximate particle velocity is

$$u(t)=-\frac{1}{\rho}\int\frac{\partial p(t)}{\partial r}dt=-\frac{1}{\rho\Delta r}\int(p_2(t)-p_1(t))dt \tag{4}$$

and the acoustic pressure p(t) at the center of two points can be approximated as

$$p(t)=\frac{p_1(t)+p_2(t)}{2} \tag{5}$$

Therefore, the acoustic intensity in the direction r, I becomes

$$I = \lim_{T \to \infty} \frac{1}{T} \int u(t)\, p(t)\, dt = \overline{u(t)\, p(t)} \qquad (6)$$

hence the acoustic intensity of equation (7) can be rewritten as

$$I = -\frac{1}{\rho \Delta r} \overline{\frac{p_1(t) + p_2(t)}{2} \int (p_2(t) - p_1(t) dt)} \qquad (7)$$

The sound intensity is directly related to the sound power. Hence, a convenient method of measuring the sound power of a source could be to set up some arbitrary closed test surface surrounding the source and then average the sound intensity passing through the surface. The sound power of the source can then be calculated from the average intensity using the following equation:

$$L_W = I + 10 \mathrm{Log}_{10} A \qquad (8)$$

where L_w is the sound power level in dB as reference power 1 pW, I is the sound intensity level in dB as reference sound intensity 1pW/m^2 and A is the area of the test surface in m^2.

EXPERIMENTAL SET-UP

An appropriate experiment and analysis system is very important for getting accurate results. In order to cover all related frequency range and minimize the influences from the transducers, the appropriate accelerometers, microphone and data acquisition system were employed.

Measurements were made on the vacuum cleaner itself in a semi-anechoic room. Sound intensity is a measure of the magnitude and direction of the flow of the sound energy [3]. The major point in the noise problems is an analysis of the product sound field that includes both the position and frequency content of each sound source. This analysis can be made with sound intensity measurements. To identify and evaluate the noise sources of the vacuum cleaner, the sound intensity measurements were made in a semi-anechoic room with a volume of 144 m^3 in accordance with IEC 704-2-1: 1984 [4].

The vacuum cleaner equipped with all accessories was set on the floor, and placed in a 0.6mx0.8mx1m cubic grid and set to operate under normal working conditions. The measurement surface on which the measuring points are located is the cubic grid enveloping the vacuum cleaner in order to measure sound power according to ISO 9614-1 standard [5]. Sound intensity measurements were made with a Brüel&Kjær type 3548 intensity probe with a 12 mm spacer and a real-time dual-channel frequency analyzer Brüel&Kjær type 2144. A sound intensity calibrator

Brüel&Kjær type 3541 calibrates the microphones. The measuring grid has 79 measuring points and linear averaging time for each measuring point was 16 sec. A GPIB Board then transferred the sound intensity measurements to a PC.

Both the measurements of the near-field sound pressure and the surface velocity were carried out at the normal working conditions of the vacuum cleaner in a semi-anechoic room. It was set on the floor and equipped with all accessories. 120 measurement points were defined on the surface of the vacuum cleaner drum, uniformly distributed. The near-field sound pressure measurements were made very close (approximately 20 mm) to the vacuum cleaner surface with a Brüel&Kjær type 4192 microphone. The surface velocity measurements were made with an accelerometer Brüel&Kjær type 4371.

RESUTS AND DISCUSSION

Preliminary measurements revealed the acoustic and structural behaviour of the vacuum cleaner. In order to calculate the shell structural borne sound power level of the vacuum cleaner, near-field sound pressure and surface velocity were measured on the vacuum cleaner. One of the near-field sound pressure measurements on the vacuum cleaner is shown in Fig. 1. as a sound pressure level spectrum of the vacuum cleaner with respect to the centre frequencies of the one-third octave bands. The frequency range of the sound pressure measurements is between 200 Hz and 8 kHz.

One of the surface velocity measurements on the vacuum cleaner is shown in Fig. 2. as a vibration level spectrum of the vacuum cleaner with respect to the centre frequencies of the one-third octave bands. The frequency range of the surface velocity measurements is between 200 Hz and 8 kHz.

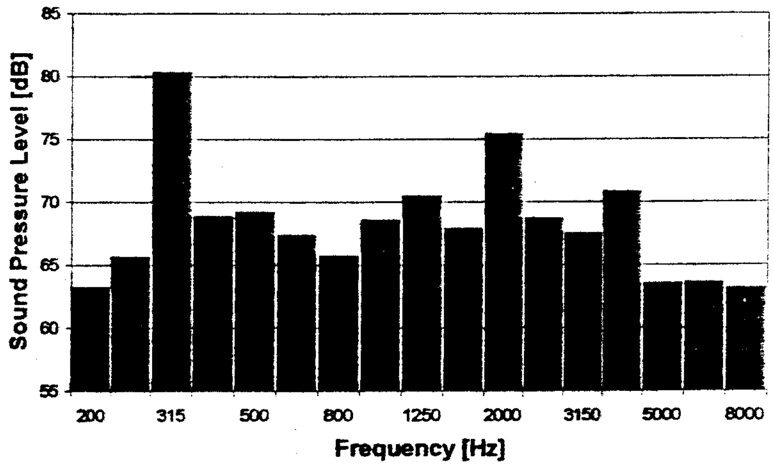

Fig. 1. One of the near-field sound pressure measurements on the vacuum cleaner. (reference pressure: 2 x 10^{-5} Pa)

Thus, the shell structural borne sound power level can be calculated at each point using the near-field sound pressure and the surface velocity values for each one-third octave center frequencies using the following equation:

$$L_W^* = \sum_{i=1}^{120} (I_i + 10\mathrm{Log}_{10} A_i) \qquad (9)$$

where L_W^* is the sound power level calculated for each one-third octave center frequencies in dB as reference power 1 pW, I_i is the sound intensity level calculated using the near-field sound pressure and the surface velocity values which are substitute into the intensity equation (1) at each of the test points on the vacuum cleaner for each one-third octave center frequencies in dB as reference intensity 1pW/m^2 and A_i is the area of every single test locations on the vacuum cleaner in m^2. If the sound intensity level from equation (1) is substituted into the sound power level equation (9), it can be expressed as

$$L_W^* = \sum_{i=1}^{120} \left(\overline{p_i\, u_i} + 10\mathrm{Log}_{10} A_i\right). \qquad (10)$$

Where the bar indicates time averaging. p_i and u_i denote the sound pressure and the surface velocity values measured for each one-third octave center frequencies, respectively.

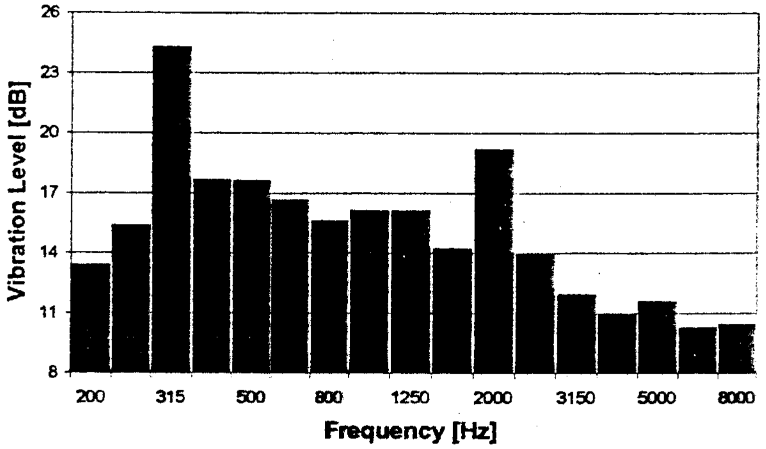

Fig. 2. One of the surface velocity measurements on the vacuum cleaner. (reference acceleration: 1.0 x 10^{-6} m/s^2)

Fig. 3. shows the sound power spectrum calculated from equation (10) on the vacuum cleaner with respect to the centre frequencies of the one-third octave bands. The

frequency range is between 200 Hz and 8 kHz. The vacuum cleaner is operating under the normal working condition during the measurements.

Fig. 4. shows the sound power spectrum measured on the vacuum cleaner with respect to the centre frequencies of the one-third octave bands according to ISO 9614-1 standard. The frequency range of the sound power measurements is between 200 Hz and 8 kHz. The vacuum cleaner is operating under the normal working condition during the sound power measurements.

The shell of the vacuum cleaner is the final component between the noise sources and the receiver. In this sense, the vacuum cleaner shell will respond in two ways to the internal noise transmitted to it: acoustic enclosure reflecting the noise energy back into the vacuum cleaner; and acoustic transmitter and radiator transmitting the noise energy to airborne noise radiation or to structure borne vibration. The comparative action of these two types of response will depend on primarily on the nature of the transmission of noise to the shell and the characteristics of the dynamic response of the shell.

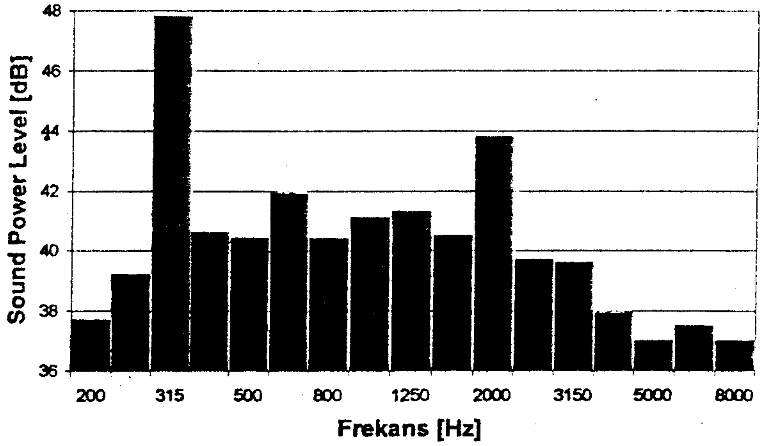

Fig. 3. Calculated sound power spectrum of the vacuum cleaner from equation (9).

The measurements presented in figures established the main components at three frequencies. 315 Hz is the fundamental frequency related to the unbalance of the rotor at the rotational speed. 2 kHz and 4 kHz could be related to the blade passing frequencies such that the six blades of the electric motor cooling fan and the twelve blades of the electrical motor fan respectively. These peaks represent the unpleasant noise perceived as pure tones especially at 315 Hz by the customer.

Shell vibration is most easily characterized in terms of the modal parameters, e.g., the shell natural frequencies, mode shapes, and damping coefficients. Ideally the shell should be designed such that all important driving frequencies would excite the shell in the mass controlled region of all its modes.

The first design modifications should be concentrated on the reduction of the side flat surfaces, the back and the top of the shell are also involved in the design variations to eliminate the abrupt changes at the bending points. The overall stiffness of the shell could be increased to raise the resonant frequencies and reduce the vibration amplitudes. It has to be avoided abrupt changes in shell curvature that act as semi-rigid boundary conditions. The basic strategy is the elimination, as much as possible, of the flat surfaces in terms of the sound and the vibration transmitted. Finally, the balance quality of the rotor as the main source could be upgraded in order to achieve a vibration reduction.

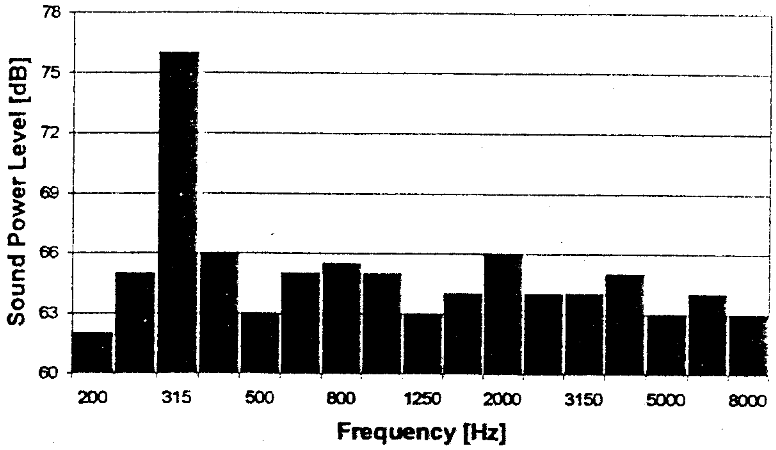

Fig. 4. Measured sound power spectrum of the vacuum cleaner.

CONCLUSIONS

In this study, the near-field sound pressure, the surface velocity and the sound intensity levels are measured. Then, by using these data, the shell structural borne sound power level is calculated and compared with the overall sound power level of the vacuum cleaner. The results presented in this paper show that the sound power level of a vacuum cleaner is depend on the frequency and distribution of the shell motion. Results obtained in this study will be usefull data for the design of the noise and the vibration reduced vacuum cleaners.

REFERENCES

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