PARAMETER ANALYSIS OF VIBRATORY ENERGY FLOW IN A HERMETIC COMPRESSOR BY STATISTICAL ENERGY ANALYSIS

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This work deals with the problem of vibratory energy flow from the internal electrical motor-compressor assembly (kit) to the shell of a hermetic compressor. The theoretical model is based on Statistical Energy Analysis. Shell and body vibratory energies have been measured at a standard operation condition, and the coupling loss factor thus obtained agreed well with predicted values calculated from impedance data. Several parameters, such as body and shell impedance, spring and shell loss factor, and model of the spring (tension and compression) were varied and their effects upon the compressor noise analyzed. Good agreement between theoretical and experimental results was observed.

1. Introduction

Noise generated by domestic appliances such as refrigerators and freezers has been a matter of increased concern in recent years to their manufacturers due to the annoyance caused on people. In these products the hermetic compressor is responsible for most of the noise which is partially radiated directly from its shell. Vibrations induced in the frame an other refrigerators structural components radiate also a significant amount of sound power. This study is aimed at the internal power flow analysis of a typical hermetic compressor, particularly from the compressor and electric motor assembly, to the shell through the three metallic helicoidal springs. Other energy flow paths to the shell include the refrigerant fluid discharge tube, the layer of lubricant oil resting at the shell bottom which is usually in contact with the stator of the electric motor, and the

intense sound field generated in the cavity excited by the suction muffler pulsation and by the structural vibrations of the compressor-motor assembly. These paths represent also a significant contribution to overall power transferred to the shell. However, they will not be discussed in this work.
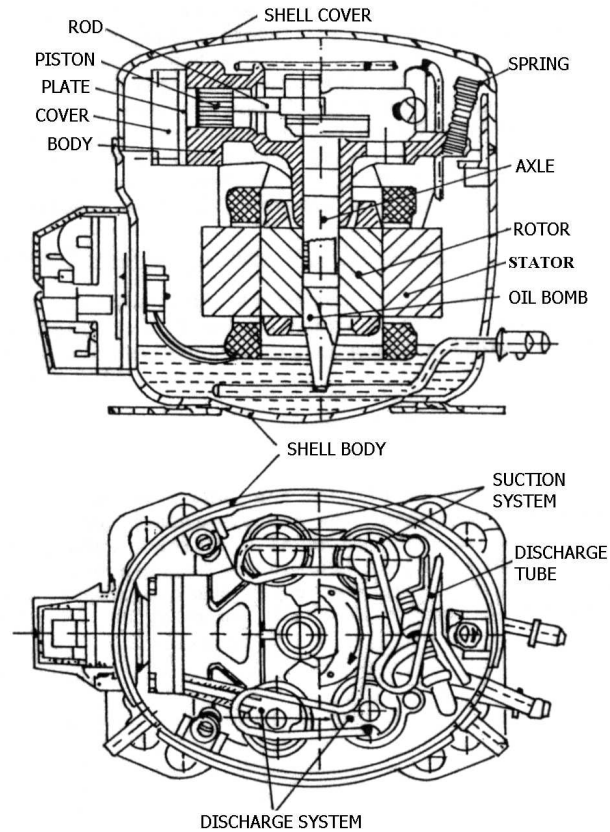

Fig. 1. Cutting section of the EMBRACO FFE 8A hermetic compressor.

The compressor model used in this study, shown in Fig. 1, is a single piston compressor (60 Hz, 115 V), power consumption less than 1 HP. Figure 2 shows a typical average sound power spectrum for this compressor model. The peak observed in the 500 Hz region is attributed to the very intense gas pulsation in the suction muffler which strongly excites the first cavity resonances. The energy flow through springs and discharge tube contributes to a great extent to the high levels observed above 1.6 kHz. This high frequency region is responsible for most of the overall sound power level. It becomes clear that in order to be effective, any noise control procedure must be able to reduce the noise radiated in this frequency region. It is therefore important to quantify the influence of the several parameters related to the energy flow to the shell, which in this study is focused on the springs only.
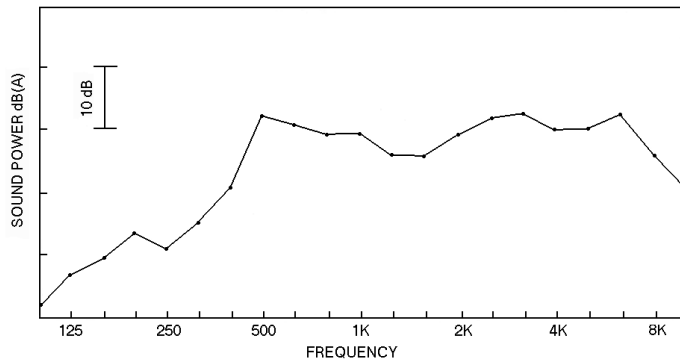
Fig. 2. Typical sound power spectrum of a FFE 8A 115V 60Hz compressor, in 1/3 octave band.

2. SEA Model

Statistical Energy Analysis was used for modelling the power flow considering two subsystems: the shell and the compressor-motor assembly. The latter, now referred to as subsystem 1 (SST$_1$), is the only subsystem receiving external excitation, attributed to large forces resultant from the pressure variations inside the cylinder, particularly during the opening of the gas discharge value. Excitations caused by the electromagnetic field in the stator of the electrical motor and by impacts between moving and rotating components due to gaps are also responsible for generation of vibratory energy in the assembly. Experiments indicate that excitations produced by the high gas pressure through flow the discharge tube, including expansion chambers, can be considered negligible.
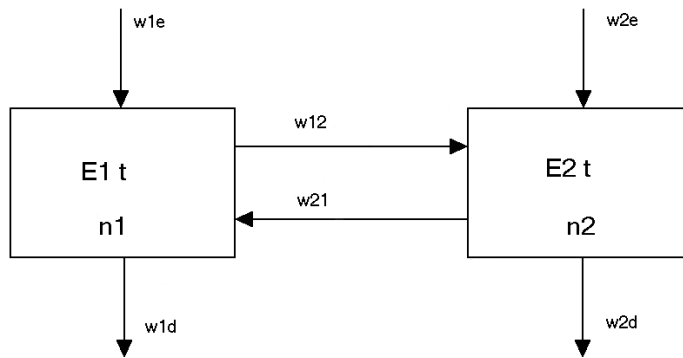

Fig. 3. Energy flow between vibrating systems.

A typical two subsystems SEA model representation is shown in Fig. 3 where respectively SST$_1$ and SST$_2$ represent the two subsystems; E_1 and E_2, the total vibratory energies per frequency band; n_1 and n_2, the modal densities (modes/Hz); η_1 and η_2,

the loss factors; η_{12} and η_{21}, the coupling loss factors; W_{diss} is the power dissipated, defined as $W = \omega\eta E$ and W_{ij}, the power transferred to the j-th subsystem, defined as $W_{ij} = \omega\eta_{ij}E_i$. The circular frequency $\omega(2\pi f)$. A power balance equation for SST_2 leads to the following relationship between the total energies of both subsystems, assuming only SST_1 is externally excited [1–3]:

$$\frac{E_2}{E_1} = \frac{\eta_{12}}{\eta_2 + \eta_{12}\dfrac{n_1}{n_2}}. \tag{1}$$

Loss factors values were obtained for both subsystems by the decay rate method. Each subsystem was hung by long and very thin wires, excited by an impulsive excitation and the response was filtered in 1/3 octave bands.

The first shell resonance was observed in the 1.6 kHz band, and the first shell resonances responsible for peaks on the sound power spectrum are above about 2.5 kHz [2, 4–6]. For this reason the analysis was carried out for the 1/3 octave bands ranging from 2.5 kHz up to 6.3 kHz. These five bands mainly dictate the overall sound power level value.

3. Coupling loss factors

Two procedures were used for the determination of the coupling loss factors. In the first one, the coupling loss factor η_{12} was obtained from the measured energies E_1 and E_2, using Eq. (1), in the form:

$$\eta_{12} = \frac{\left(\dfrac{E_2}{E_1}\right)\eta_2}{1 - \left(\dfrac{E_2}{E_1}\right)\left(\dfrac{n_1}{n_2}\right)}. \tag{2}$$

Figure 4 shows η_{12} results calculated from the above expression and compared with results obtained by a second procedure, as described below.

The power flow between two subsystems connected at a point can be determined as function of their respective point impedances Z_1 and Z_2, obtained at the connecting point. This can be used for coupling loss factor determination.

It has been shown [1] that the ratio between the vibratory energies can be expressed as:

$$\frac{E_2}{E_1} = \frac{2}{\pi\,n_1\,\eta_2\,\omega}\,\frac{R_1\,R_2}{|Z_1 + Z_2|^2}, \tag{3}$$

where R_1 and R_2 represent the real part of the complex impedances, Z_1 and Z_2, respectively. Substituting into Eq. (1), one obtains η_{12}. The impedances that enter Eq. (2) were obtained through measurements of force and acceleration signals by means of an

impedance head. The exciting point force was of random type, generated by an electro-dynamical shaker. The acceleration signal was later digitally converted into velocity in a two-channel FFT analyzer for impedance calculation.

The compressor-electric motor assembly is suspended by three springs, so that each of them will have to be considered to be part of one of the subsystems. Springs were then assumed to be part of the compressor-electric motor assembly. The impedance Z_1 as "seen" by the shell then results from the sum in series of the impedances of the spring and of the compressor block (attached to the electric motor), the latter being elevated at the spring's attachment points. The overall coupling loss factor resulted from the sum of the values calculated through Eqs. (1) and (2), for each spring and each spring direction (longitudinal and two transversal).

Figure 4 shows a reasonably good agreement between the coupling loss factors results determined by the described procedures. Maximum errors are limited to 2–3 dB.
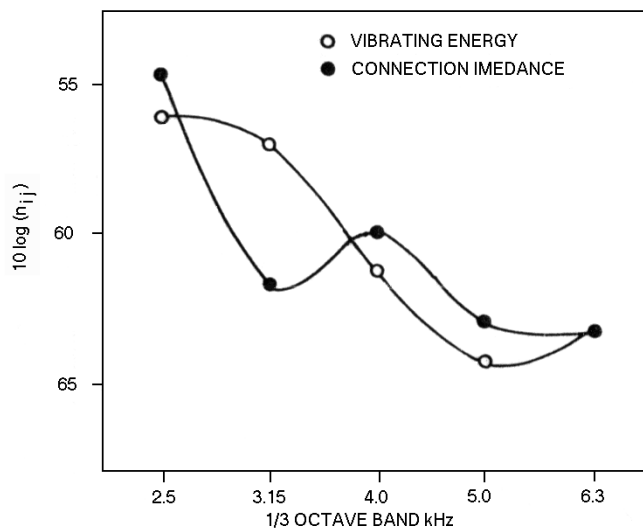

Fig. 4. Body-shell coupling factor.

4. Sound power level radiated

The sound power radiated from a vibrating structure can be determined as function of its radiation efficiency, σ_{rad}, as follows [7]:

$$W_{\mathrm{rad}} = \rho_0 c\, S\, \sigma_{\mathrm{rad}} \left(\frac{E_2}{m_2} \right), \tag{4}$$

where $\rho_0 c$ is the air impedance (415 Rayl); S is the structure radiation surface; E_2 is the vibratory energy and m_2, its mass. The ratio E_2/m_2 represents space and RMS squared

averaged velocity ($< \overline{v}^2 >$) for SST_2 (shell), where $S = 0.141$ m^2 and $m_2 = 3.44$ kg. The radiation efficiency was experimentally obtained by BARBOSA [2]. Using Eq. (3) one can calculate the radiated sound power, W_{rad}, shown in Fig. 5, and compared with the averaged sound power of this compressor model, measured in a reverberant chamber (210 m^3 volume) by the comparison method [8]. The agreement is reasonably good, indicating maximum differences of the order of 3 dB. In the overall level the difference is 0.1 dB only.
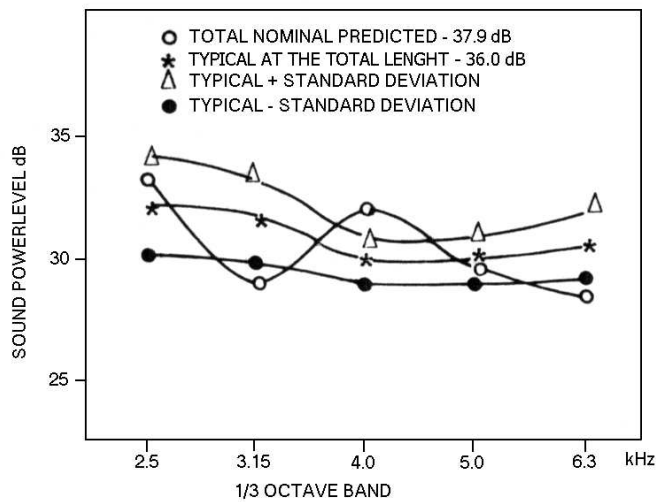

Fig. 5. Predicted sound power level.

5. Analysis of parameters

The influence of several parameters related to the vibratory energy flow to the shell and its consequent conversion into sound energy is analysed in this section. Energy of SST_1 (compressor-electric motor assembly) was assumed invariant in this analysis. Shell sound power radiation was calculated as function of parameters listed below:

Shell impedance: shell impedance values were considered in the SEA model ranging from 0.25 to 4.0 times its nominal value. Shell sound power results were normalized with respect to its nominal values, as shown in Fig. 6. It can be seen that the increase in the impedance of the shell by a factor of 4 reduce the sound power by 6 dB, which is very significant.

Compressor impedance: compressor block impedance has shown to have little influence on the sound power radiation. Variations of the order of 100 times were considered in the model presenting no significant variations in the sound power. This indicates that its impedance is much larger than that of the springs, which in this SEA model are part of SST_1.

Shell loss factor: variations from 0.25 to 4.0 times the nominal value of the shell loss factor were also considered in SEA model indicating that higher values produce lower sound power radiation. Its influence is very similar to that of shell impedance. Shell is the only component responsible for the sound radiation. The lesser the vibratory energy is transferred to it, the lower is the sound radiation. This can be achieved by increasing its impedance. At the same time, the higher the energy dissipation rate, which is achieved by increasing its structural damping, the lower is also the sound radiation. This explains the similar tendencies observed when loss factor and impedance are increased in the shell.

Springs loss factor: increasing springs loss factors reduces also the sound radiation, as shown in Fig. 6. Springs represent the path flow of energy to the shell such that damping has the important dissipation effect in these components. BASTOS [9] analysed the reduction in the transmitted energy through helicoidal springs at their resonances by adding damping. This was verified experimentally by LENZI [3]. Small rubber inserts were placed in the inner springs diameter and sound power was measured. Significant reductions above 2 kHz were observed.
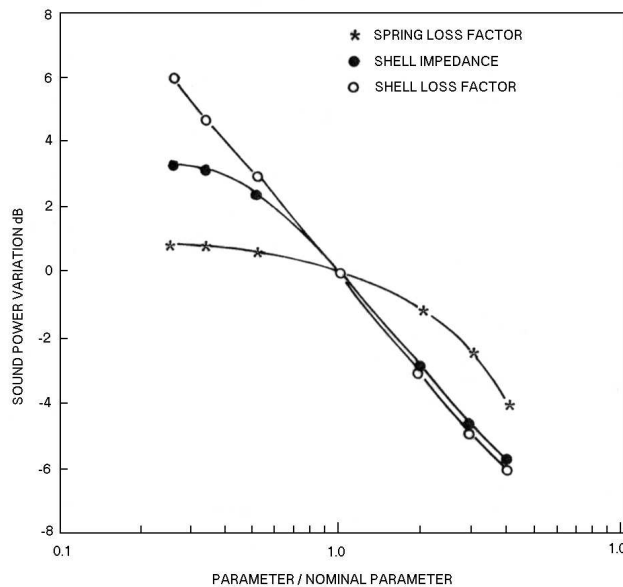

Fig. 6. Influence of the shell impedance, shell loss factor and spring over the total sound power.

Springs mounting: springs placed at the lower part of the shell, supporting the compressor-electric motor assembly are considered a better configuration from the noise radiation point of view [5, 10]. Preliminary studies indicated that in the frequency band of interest the shell impedance modules at the bottom, for this compressor model, is

approximately 3.5 times larger compared to values measured at the usual suspension supports. In case of springs of exactly the same geometry and material such an increase in the impedance would result in 5.0 dB reduction in sound power. However, for this new configuration four springs are recommended, with new specifications in order to attend vibration levels requirements, resulting in 2.0 dB reduction, approximately, in sound power. Analyses for springs under tension or compression resulted in identical behaviour.

6. Conclusions

The results presented in this work are restricted to one particular compressor model (Embraco FFE 8A) and for 1/3 octave band ranging from 2.5 kHz to 6.3 kHz. Discharge tube was neglected in the SEA model since it was not considered an important energy flow to the shell.

The importance of shell impedance and damping, and damping of springs can be seen in Fig. 6. Increasing the shell impedance or its loss factor by a factor of two, a 3 dB reduction in the sound power is achieved, which is very significant. This has stimulated development engineers to design new shell geometries and searching for improved structural damping addition techniques.

Compressor block impedance at the spring's attachment points is considered to be sufficiently large from the energy flow point of view. Studies seeking to minimize energy absorption due to the action of gas pressure resultant forces on the cylinder and valves plates are now under way, as well as at the journal bearing produced by the shaft-rotor assembly.

Springs are also important components to be considered in compressor noise reduction works since they are responsibly for most of the overall vibratory energy transferred to the shell, including other mechanical paths, such as the discharge tube and through the lubricant oil.

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