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ABSTRACT

The paper deals with the direct sound radiation of the hermetic shell of the reciprocating refrigeration compressor. The sound power level of compressor can be evaluated by applying well known conventional methods: using sound pressure levels measured in reverberant or in anechoic chamber or by integrating the sound intensity readings over a closed surface containing the source.
An alternative method using 6 accelerometer measurements for evaluating the sound power level radiated by the hermetic compressor is presented in the paper. The method is based on modeling the acoustic radiation of compressor shell by 4 simple acoustic sources – 1 monopole and 3 dipoles. The monopole accounts for the change of the volume of the compressor shell, known as shell "breathing", while each of 3 dipole stands for the "rigid body" vibration movements in 3 directions. The sound power level is then evaluated by the superposition of the sound power levels of the 4 simple acoustic sources. The "6 accelerometers method" is applied on a number of small and medium size hermetic compressors. The obtained results are compared and discussed. The accuracy as function of frequency range and dimensions of source are examined.

INTRODUCTION

The main vibroacoustic source of refrigerating and air-conditioning machinery is compressor. One of two principal components of the noise generated by compressors are: the noise transmitted to the machinery in the form of structure-borne and fluid-borne vibroacoustic energy and the airborne noise which is directly radiated by the compressor shell. The airborne noise, which is predominant in numerous cases is generated by shell vibration. Consequently by measuring the shell vibration the sound radiation of compressor can be evaluated. Since the compressor shell elastically deforms while radiating/vibrating a simple "one number" descriptor of shell vibration does not exist. The straightforward but tedious way to measure such "complex" vibration of the shell consists of use of a fine mesh of measurement points, similar to those used in FEM analysis. In order to define the vibrations in operating conditions, complex amplitudes of vibration at all measurement points are needed, for each measurement frequency. The number of data for such complete definition of shell vibration is prohibitive.

In order to overcome this point, shell vibration is to be developed in series of simple vibration modes and then truncated to a acceptable/useful number of terms. Only the most efficient modes are to be taken into account. The most efficient radiation modes are associated with the simplest vibration movements. The simplest modes of the compressor shell vibration are rigid body modes. There are 3 independent rigid body modes which can efficiently radiate the sound. Each corresponds to the displacement of the center of gravity of the shell in one of 3 principal directions. The rest of the shell vibration of the shell can be approximated by breathing deformation of the shell.

Since none of these 4 vibration modes can be expressed as a linear combination of others, they can be considered as orthogonal. The 4 principal modes correspond to 4 simple acoustic sources: 3 dipoles and 1 monopole. The total acoustic power is equal to the sum of the acoustic powers of 4 elementary sources.

"SIX ACCELEROMETER METHOD"

Acoustic radiation of compressor shell

When acoustic radiation is considered, compressor shell can be modeled using an equivalent sphere. In such a case the acoustic radiation of rigid body modes is to be computed using the expressions for vibrating sphere (dipole radiation), while the breathing of shell is to be accounted for using pulsating sphere formula (monopole radiation).

Such a model allows a relatively accurate estimation of radiated sound power of the hermetic shells of small and medium size refrigerant compressors for the whole range of audible frequencies. The acoustic radiation is evaluated using the vibration amplitude and shell dimensions. In order to use the measured vibration amplitude a 6 accelerometer antenna/array presented is needed.

The method consists of simultaneous measurement of acceleration at six points of compressor shell. In the Fig-1 the measurement positions A1, A2, A3, A4, A5 and A6 are shown on the equivalent sphere, which is associated to the compressor shell. Moreover, this method allows for the measurement of radiated acoustic power in noisy environment. The accelerometer readings are denoted by a_1, a_2, a_3, a_4, a_5 and a_6 .

"Rigid body" vibration of the compressor shell in X, Y and Z directions are given by the following expressions:

$$a_x = \frac{1}{2}(a_1 - a_2); \quad a_y = \frac{1}{2}(a_3 - a_4); \quad a_z = \frac{1}{2}(a_5 - a_6)$$

"Breathing" of the shell can be computed using average value of in-phase accelerometer readings:

$$a_b = \frac{1}{6}(a_1 + a_2 + a_3 + a_4 + a_5 + a_6)$$

The compressor shell is modelled by an equivalent sphere radiating in infinite space.

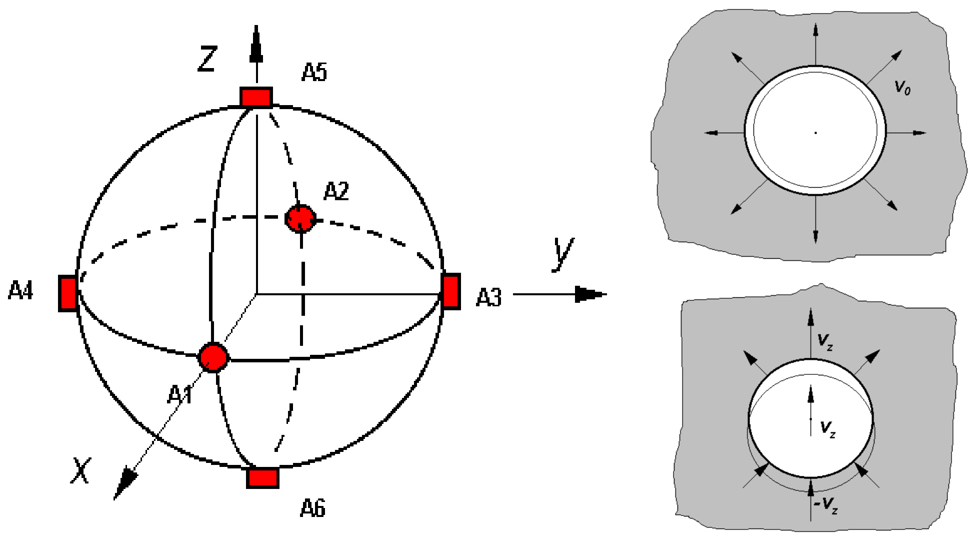

Figure 1 – *Vibroacoustic model of compressor shell.*
Left: position of measurement points on the compressor shell.
Right-up: acoustic model of "pulsating sphere".
Right - down: acoustic model of "vibrating sphere".

The acoustic pressure $p_0(r)$ generated by breathing vibration are given by the following expression:

$$p(r) = \alpha_0 \frac{e^{-jkr}}{r} a_b \quad ; \quad \alpha_0 = \rho \frac{r_0^2}{(1 + jkr_0)} e^{jkr_0}$$

where r_0, ρ, c, k are sphere radius, mass density, speed of the sound in the air and wavenumber.

The acoustic pressure $p_x(r,\theta_x)$ generated by rigid body vibration in X direction are given by the following expression :

$$p(r,\theta_x) = \beta_0 (1+1/jkr) \frac{e^{-jkr}}{r} \cos\theta_x \quad a_x \quad ; \quad \beta_0 = \frac{\rho c}{j\omega} \frac{r_0}{1+2/jkr_0 + 2/(jkr_0)} e^{jkr_0}$$

where θ, ω angle of the receiver position with respect to the vibration direction and frequency.

Once the acoustic pressure field is known, the radiated sound power can be computed.

Measurement of sound power using 6 accelerometers

A FFT analyser, which permits simultaneous processing of at least 6 channels is needed in order to use the 6 accelerometer technique. The analyser evaluates the power and cross spectra defined as follows:

$$S_{ij} = \frac{a_i a_j^*}{2} \quad \begin{cases} i = j & \text{power spectrum} \\ i \neq j & \text{cross spectrum} \end{cases}$$

, where $(\cdot)^*$ denotes complex conjugate.

In the paper the first channel is the reference channel. The FFT analyser evaluated simultaneously the power spectra corresponding to the 6 measurement channels and cross spectra with the respect to the reference channel.

$$S_{11}\ ;\ S_{22}\ ;\ S_{33}\ ;\ S_{44}\ ;\ S_{55}\ ;\ S_{66}\ ;\ S_{21}\ ;\ S_{31}\ ;\ S_{41}\ ;\ S_{51}\ ;\ S_{61}$$

The other spectral data can be computed using the measured data and the following expression:

$$S_{jk} = \left[S_{ji} S_{ki}^* \right] / S_{ii} \quad ; \quad S_{jk} = S_{kj}^*$$

For the most of the industrial applications the number of averages have not to exceed 32. But, if the signal to noise ratio is particularly low, the number of averages could be increased. These quantities are to be expressed using the measured spectra. Using some basic algebra the power spectrum of rigid body vibration in X direction can be defined as follows:

$$S_{xx} = \frac{a_x\, a_x^*}{2} = \frac{1}{2} \left\{ \frac{1}{2}(a_1 - a_2) \frac{1}{2}(a_1 - a_2)^* \right\} = \frac{1}{4} \left(S_{11} + S_{22} - 2\Re[S_{21}] \right)$$

The analogous expressions can be derived for the "rigid body" vibrations in Y and Z directions and for the "breathing".

$$S_{yy} = \frac{a_y a_y^*}{2} \quad ; \quad S_{zz} = \frac{a_z a_z^*}{2} \quad ; \quad S_{bb} = \frac{a_b a_b^*}{2}$$

The "rigid body" vibration can be used to compute the orbit/trajectory of the centre of gravity of compressor. The phase relation between different types of vibration are defined by the following cross spectra.

$$S_{xb} = \frac{a_x a_b^*}{2} \quad ; \quad S_{yb} = \frac{a_y a_b^*}{2} \quad ; \quad S_{zb} = \frac{a_z a_b^*}{2}$$

The power spectra and cross spectra defined previously are be expressed using the measured spectral data. The power spectrum of acoustic pressure $S_{pp} = \left(pp^*\right)/2$ and the radiated acoustic power P_0 are:

$$S_{pp}(r) = \frac{\alpha_0 \alpha_0^*}{r^2} S_{bb} \quad ; \quad P_0 = \frac{4\pi}{\rho c} \alpha_0 \alpha_0^* S_{bb}$$

The power spectrum of acoustic pressure and the radiated acoustic power P_x are :

$$S_{pp}(r, \theta_x) = \beta_0 \beta_0^* \frac{1 - 1/(jkr)^2}{r^2} \cos^2 \theta_x \quad S_{xx} \quad ; \quad P_x = \frac{4\pi}{3\rho c} \beta_0 \beta_0^* S_{xx}$$

Similar expressions can be derived for vibration in Y and in Z direction.

The total sound power generated by the complex vibration of shell is obtained by superposition of sound powers of elementary source components.

$$P_x = P_0 + P_x + P_y + P_z$$

The acoustic power in 1/3 octave band can be recomputed from the narrow band results using an appropriate integration technique.

The standardised 1/3 octave band filters (Butterworth filters of the third order) are used. The filters are then applied to the narrow band frequency spectrum.

The integration is carried out over the whole frequency domain. In order to respect the "stabilisation time" of filters, the lowest accurate 1/3 octave band should contain at least 5 frequency beams within the passing band. The upper integration limit of the highest 1/3 octave band is the central frequency of the next 1/3 octave band.

Validation of the developed method

In order to validate the developed measurement procedure the following experimental set-up is realised. The compressor is mounted on a rigid steel base. The mass of the base is approximately 400 kg. The set-up is placed inside the acoustically treated measurement room. The compressor is connected to the calorimeter which can handle the cooling capacity of the compressor.

High and low pressure have to be controlled accurately. Superheat temperature and liquid sub-cooling are adjusted and regulated using the regulation system which controls heat exchange of evaporator and condenser. The thermodynamic parameters have to be perfectly stable before carrying out the measurements.

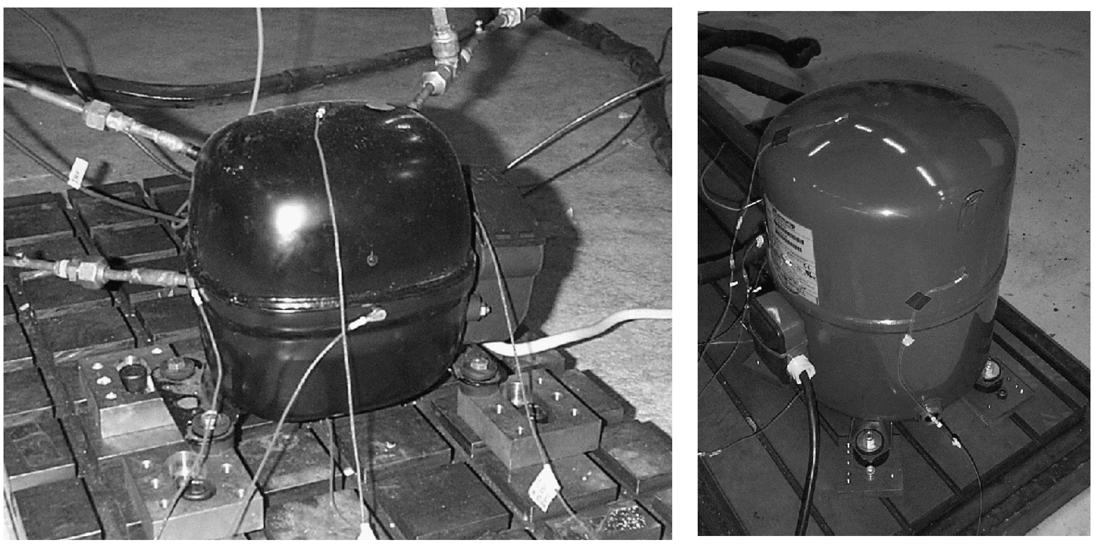

Fig. 2 : *The 6 accelerometer antenna/array applied to small and medium size refrigerant compressors.. Left – small size refrigerant single piston compressor. Right – medium size two-piston compressor.*

The connection to the calorimeter circuit is realised using the pipes of nominal diameter defined for each compressor. The standard rubber grommets isolators, which are supplied with the compressor, are used. The screw have to be tightened following manufacturer specifications.

Sometimes the connecting pipes can undergo resonant phenomena. To avoid this it is recommended to apply some commercially available damping material on suction and discharge pipes.

The six accelerometers are glued directly on the compressor shell. The six accelerometer antenna is connected to a dynamic signal analyser. The amplitudes of vibration corresponding to the 4 simple acoustic sources are computed using a home made post-processing software.

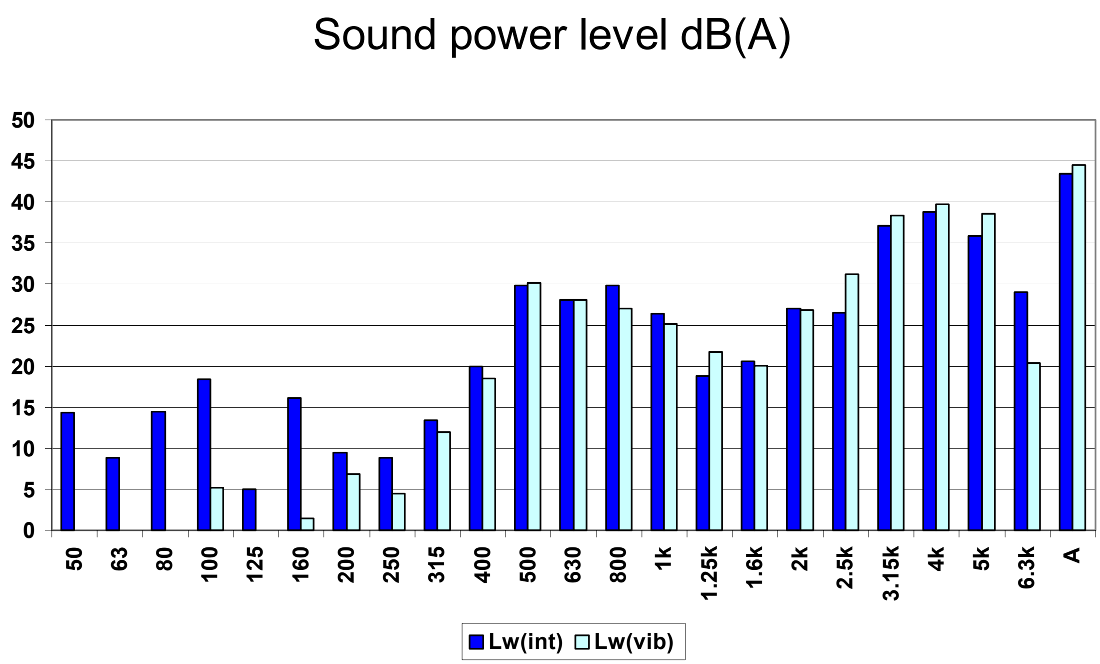

Fig. 3: *Sound power level radiated by a small size single piston refrigerant compressor; dark columns – results obtained using the intensity measurement; light columns - – results obtained using the vibration measurements and the computation procedure described in the paper.*

The sound power level is then evaluated by the superposition of the sound power levels of the 4 simple acoustic sources.

The method is applied on a more than 15 different types of small and medium size hermetic compressors. Single piston, two-piston and four-piston compressors were tested.

The same quantity is measured using the intensity probe and a measurement procedure described in ISO – 9614-1 standard.

The obtained results are compared using 1/3 octave A weighted spectra. Two results are represented in Fig. 3 and Fig. 4. The method seems to give the “A” weighted radiated sound power very accurately.

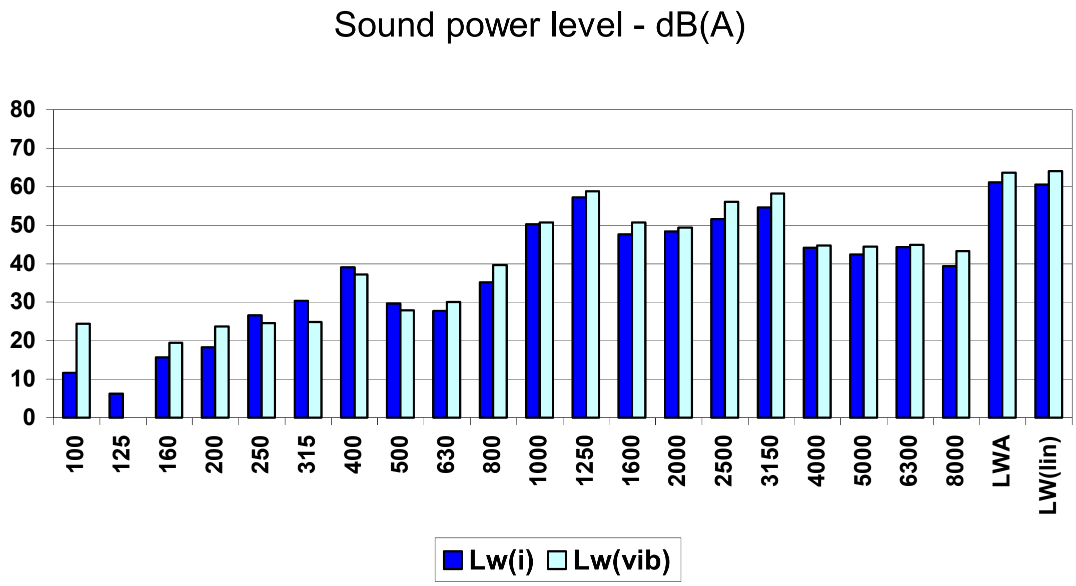

Fig. 4: *Sound power level radiated by a medium size two piston refrigerant compressor; dark columns – results obtained using the intensity measurement; light columns - – results obtained using the vibration measurements and the computation procedure described in the paper.*

CONCLUDING REMARKS

For the moment, the 6 accelerometer method is applied to 15 different small and medium size refrigerant compressors for refrigeration and air-conditioning applications. The obtained results are very encouraging. The developed method gives excellent results for the evaluation of sound power level. The developed concept is tested in the acoustic laboratories of few European compressor manufacturers. The method can be used in very noisy environments – production sites. The developed method may be standardised in the near future. The development of the method is partially financed by the European Commission under the contract NABUCCO / GRD-CT1999-00105.

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