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Acoustic analysis of an induction motor with viscoelastic bearing supports

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Abstract

The demand for silent bearing applications has resulted in the development of an effective damping layer between the outer ring of a rolling bearing and the surrounding structure. By means of numerical modeling using both FEM and BEM techniques an induction motor for household appliances is analyzed. A hybrid modeling approach combining measured structural velocities with a BEM formulation is used to validate the acoustic model. The numerical results are compared with results obtained from sound intensity measurements estimating the radiated sound power level for a running electric motor. It is found that a relatively simple boundary element model is capable of predicting the radiated sound power in a wide frequency range. By using BEM in combination with the radiation modes formulation it is found that a properly designed viscoelastic layer in the vicinity of the bearing is theoretically capable of reducing a fair amount of sound emitted by the motor.

1. Introduction

In the automotive and household appliance industry, an increasingly better performance is required concerning the vibrational and acoustical behavior of rolling bearings. In rotor dynamic applications, such as electric motors, the bearings act mainly as vibration transmitters. The main vibration source comes from the (radial) electromagnetic forces due to a variable air gap between the stator and rotor. These vibrations are transmitted to the housing, which is the main radiator of noise. The vibration transfer path can be interrupted by means of a soft viscoelastic bearing support. In this way, vibrations of the rotor can be isolated and damped resulting in a reduced noise radiation of the housing.

In these investigations a structural analysis tool is used to model a complete bearing application with viscoelastic supports. This tool is dedicated to rolling bearings and is propriety of SKF. Within the models, the rolling bearing properties, the flexibility of elastic components (such as the shaft or the housing) and viscoelastic material behavior are taken into account (ref. Wensing [1], Tillema [2]). The use of FEM enables to model both elastic and viscoelastic components of arbitrary geometry. Moreover, a CMS reduction method is applied to these FEM models allowing for efficient computation in both the frequency and the time domain. In this study, an induction motor for household appliances is analyzed numerically and experimentally. In Figure 1 the numerical model and a photo of the application are shown.

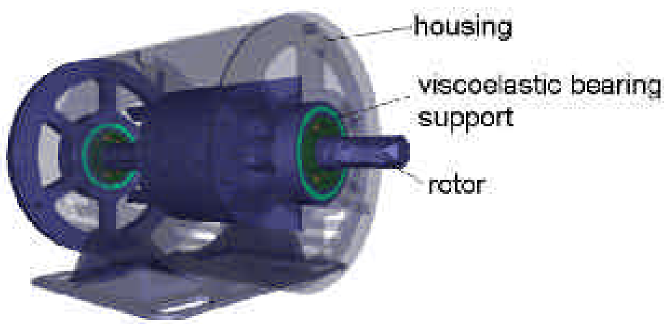

Figure 1: Numerical model (left) and photograph (right) of an electric motor with viscoelastic bearing supports.

Besides the structural dynamic behavior, also the acoustic behavior of the motor is studied. A boundary element method (BEM) model is used to predict the acoustic behavior of the motor in terms of the radiated sound power level. The validity of this model is tested by comparing the results of sound intensity measurements with numerically obtained results of a running electric motor. To ensure accurate normal velocity values of the radiating surface, measured data are used as input for the acoustic BEM model. This hybrid structural-acoustic approach is discussed in section 2. In section 3 the experimental set up for the sound intensity measurements is addressed, whereas in section 4 the numerical and experimental results are compared.
For efficient acoustic computations on the electric motor with variable viscoelastic support designs, BEM is applied in combination with the radiation modes technique (ref. Cunefare [4], Kuijpers [3]). This technique enables to rapidly determine the total radiated sound power of a vibrating structure (section 5).

2. Hybrid structural-acoustic modeling

In order to validate the acoustic model it is of great importance to determine the structural velocities of the vibrating object accurately. Therefore, in these investigations, the measured structural response of the electric motor is used as input for the computational analysis. A running mode analysis was carried out on the motor and the vibration spectra are determined for each point of the measurement grid. These spectra contain 16384 spectral lines, resulting in a frequency resolution of 0.5 Hz in the frequency domain. To be able to perform an acoustic analysis in a reasonable timeframe the frequency range from 0 to 3 kHz is divided into 120 bands of 25 Hz, as the excitation frequencies occur at harmonics of 25 Hz. In these frequency bands the spectral power is summated. The measured velocity data is projected and linearly interpolated on a cylindrical BEM model, representing the radiating surface of the motor (Figure 2).

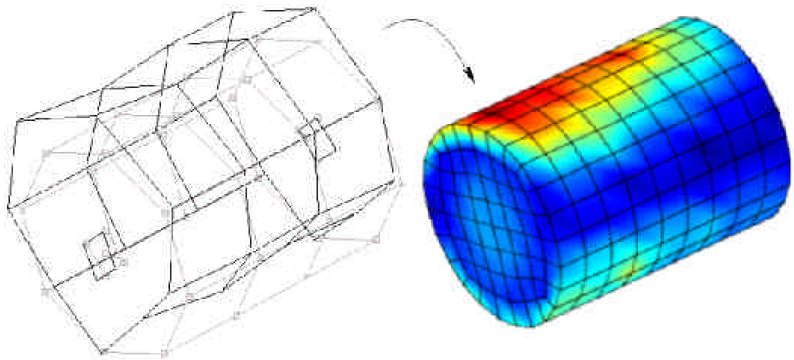

Figure 2: Measured normal velocity projected on a BEM model for a specific frequency.

Next, the acoustic radiated power is calculated as a function of frequency in accordance with the direct Boundary Element formulation. It is noted that in the analyses a half-space formulation is applied as the electric motor is fixed to a large smooth surface.

3. Experimental setup

Sound intensity measurements are performed to estimate the radiated acoustic power of the electric motor. In general, the measurement accuracy is in the order of 1.5 dB (p-I index).
Before the actual measurements on the electric motor can take place, a number of boundary conditions need to be ensured. Normally, under operating conditions an electric motor is fixed. Therefore, in this study the motor is mounted to a table with such a high mass that vibrations of the table are negligible. Moreover, to further avoid any contribution of the table radiation a plate with smooth surface was placed around the motor (see Figure 3). The plate is structurally decoupled from the table using thick pieces of foam and a small air gap with the table. This setup allows us to consider the system as a vibrating structure in a half-space.

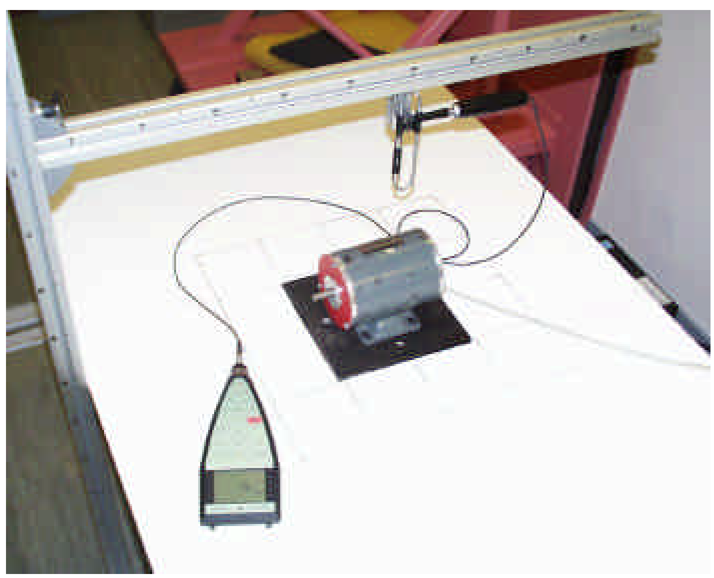

Figure 3: Setup for acoustic intensity measurements of the electric motor.

The half-space boundary (plate) is considered acoustically hard, i.e. all incoming sound is reflected into space. As the dimensions of the plate are 100 by 150 cm, it enables us to perform accurate analyses from about 500 Hz. For lower frequencies the acoustic wavelengths are too large with respect to the table dimensions, meaning that the half-space assumption is violated.

4. Results

The acoustic radiated power is determined in 1/3-octave bands, both experimentally and numerically, as depicted in Figure 4. Only the frequency bands from 500 to 2500 Hz are considered as the acoustic power decreased considerably for higher frequencies.

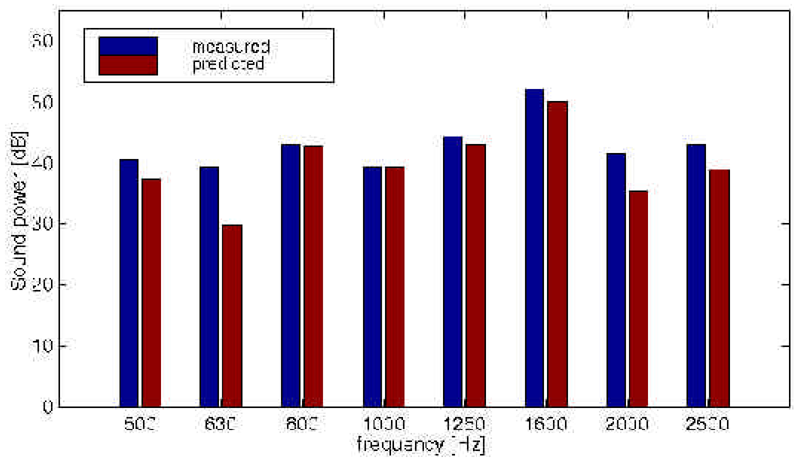

Figure 4: Measured and predicted radiated sound power for the running induction motor.

From Figure 4 a fairly good agreement can be observed between predicted and measured acoustic power. Apart from a few 1/3-octave bands, the accuracy is within 3 dB. It is seen that the lower sound power values, e.g. for 630 and 2000 Hz, are underestimated. An important reason for this discrepancy is the effect of spectral leakage as a result of windowing and filtering techniques. It is clear, however, that the trend of the acoustic radiated power is predicted well for this system, using just a very simple BEM model. What is essential to note is that the effect of model geometry, mesh refinement, element type or interpolation scheme are of minor importance in comparison with the measurement accuracy and the system variability. It was investigated that these features showed no noticable difference with respect to the final dB level of the system.
It is, however, important to include precise velocity data as input for the analyses. It is therefore essential to perform accurate vibration measurements or to have a very accurate structural model of the application for the complete frequency range.

5. Numerical modeling

As the acoustic BEM model is successfully validated with experiments it can be used as a design tool for acoustic optimization. In this section a pure numerical approach is presented in which a viscoelastic bearing support is applied as noise reducing measure. The numerical model as shown in Figure 1 is used for the structural dynamic analysis of the electric motor. The elastic components are modeled with FEM and, subsequently, reduced with a Component Mode Synthesis (CMS) technique. The viscoelastic bearing support is modeled similarly with the exception that viscoelastic material behavior, featuring a frequency dependent shear modulus and material loss factor, is included (ref. Tillema [2]). The rolling bearing model consists of a rigid

inner and outer ring, which are connected by Hertzian springs and viscous dampers simulating the dynamics of each rolling contact.
The components are joined at their interfaces resulting in a complete electric motor assembly as depicted in Figure 1. Finally, the foot of the motor is fixed to the ground, whereas an axial preload is applied to the bearings in order to ensure full contact of the rolling elements with the raceway. This model was validated with experimental modal analysis and showed a good agreement up to 1.1 kHz.
In operating conditions the electric motor is excited mainly by electromagnetic forces acting both on the rotor and the stator. The vibration amplitudes of the nodal points on the motor are determined as a function of frequency by performing a harmonic response analysis.
For acoustic radiation it is assumed that the outer surface of the application, i.e. the housing and the two end shields, contribute most to the total radiated sound power. Therefore, only these components are acoustically modeled with BEM. The normal velocity data of the structural model are projected on the BEM model with which the total radiated sound power is calculated.
For the sound power calculation the radiation modes formulation is used (Cunefare [4], Kuijpers [3]). The radiation modes form an orthogonal modal basis with respect to the radiated acoustic power of a vibrating structure. These are dependent on the geometry and frequency only, not on the boundary conditions. Therefore, the radiation modes can be reused as long as the geometry of the structure remains unchanged. This is of interest in these investigations as in general the radiating surface of a bearing application does not change when a viscoelastic layer is added in the vicinity of the bearing.
In Figure 5 the radiated acoustic power is shown in 1/3-octave bands for the electric motor with and without viscoelastic supports. As a result of the applied reduction methods such an acoustic analysis with 193 DOF for 300 frequency steps is performed in the order of minutes.

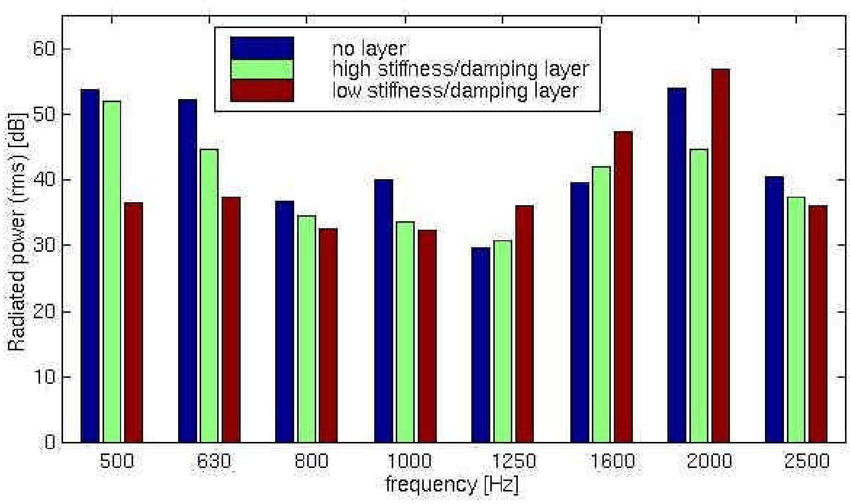

Figure 5: Radiated acoustic power level of a motor with and without viscoelastic supports.

It is observed that by applying a high damping viscoelastic support the predicted sound power level of the motor is reduced significantly, specifically in the high frequency range. For a layer with low stiffness and damping characteristics the sound level is considerably lower in the low frequency range, but the level increases for high frequencies. Apparently, in this case vibrations are effectively isolated at low frequencies but marginally damped at higher frequencies. As the stator of the electric motor directly excites the housing it is essential to create damping in the system.

For several reasons it appeared very difficult to accurately predict the acoustic behavior of the running motor based on a pure numerical approach. The structural model, for example, is not accurate enough in the mid frequency range from 1 to 2 kHz. More importantly, the excitation forces in this study are modeled poorly, i.e. by sinusoidal forces, which are constant with frequency. Clearly, this must be improved to accurately determine the radiated power of a running induction motor. The acoustic model does, however, serve as a valuable design tool when viscoelastic supports are concerned. A harmonic response analysis in combination with the radiation modes technique enables to efficiently determine the acoustic radiated power of the application. This is of interest for design optimization problems, in particular.

6. Conclusions

The acoustic BEM model appeared very efficient for the prediction of the radiated sound power of a vibrating electric motor. This was experimentally validated with sound intensity measurements. The simple cylindrical mesh is capable of predicting the acoustic behavior in a fairly wide frequency range. The mesh geometry, mesh refinement, the use of higher order boundary elements or a more sophisticated interpolation scheme hardly affects the computed radiated sound power in comparison with the measurement accuracy and the system variability.

For a good prediction of the radiated sound power in broadband problems it is essential to include accurate structural velocity data in the acoustic analysis. Therefore, either a hybrid modeling approach using measured velocity data or an accurate numerical model needs to be used.

It was investigated numerically that a high damping viscoelastic bearing support is capable of reducing the sound power level of an electric motor. A soft and low damping viscoelastic support can isolate vibrations in the lower frequency range, but the vibration level may increase for higher frequencies.

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