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Vibration isolation on weak and finite support - a model and parameter study

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A model of the vibration isolation for machines on weak and finite supports is discussed. Data from measurements on fans and supports are presented. The data are introduced into the model. Finally an example is given to illustrate the use of the model through the use of real input data.

1 Introduction

Practically all machines with moving parts will give rise to disturbances that can cause noise and vibrations. The vibration isolation is dependent on the support and in order to accurately predict its efficiency, a good understanding of the whole system is necessary. Understanding of the system can normally be achieved through analysis or experimental studies. However reliable force input data from the machinery on the support is very scarce. Nevertheless, the noise thus generated must be accounted for when designing solutions for various applications. Normally the requirement is set as a "highest" noise or vibration level. Therefore, in an attempt to fill the gap and enhance the engineering model currently used in our daily work, mobility tests on floors have been performed, along with thorough force measurements on medium sized fans.

2 Model for the vibration isolation

The model for evalutating the vibration isolation is based on the different mobilities inherent to the system. In its most simple form it involves, respectively, the mobility of the source, the isolator and the support.

2.1 The model parameters

For an existing support, finding the mobility is readily done through an impact excitation. Adopting the experimental data in the model, the resonant characteristics of the support are captured.

The vibration isolation is defined as

$$D_{IL} = 20\log\left|\frac{Y_m + Y_i + Y_{ms}}{Y_m + Y_{ms}}\right| \tag{1}$$

where Y_m is the mobility of the moving and excited mass, Y_i is the mobility of the isolators and Y_{ms} is the mobility of the support, most simply represented by a mass spring system in calculations.

3 Floor Measurements

Existing floors have been tested through impact excitation to yield the point mobilities. They have then been inserted into the model for the vibration isolation, see Equation (1), that has been completed with an appropriate choice of parameters.

3.1 A heavy concrete floor

The point mobility of a thick concrete floor is presented in Figure 1. The resultant vibration isolation can be viewed in Figure 2. The chosen parameters are a mass mobility corresponding to a mass of 200 kg, and a spring mobility corresponding to a spring stiffness yielding an eigenfrequency of 4 Hz in connection with the mass. They are compared with the calculated theoretical results for a 200 mm thick concrete floor.

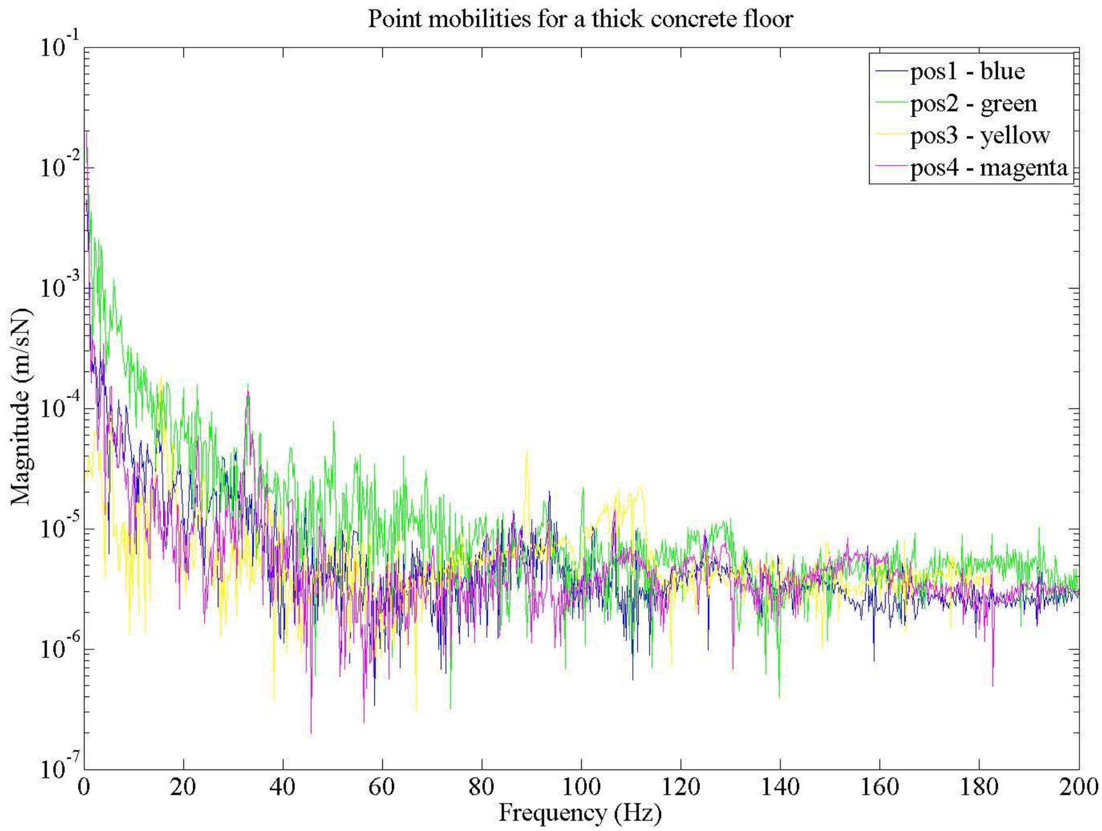

Figure 1: The measured point mobility of a thick concrete floor.

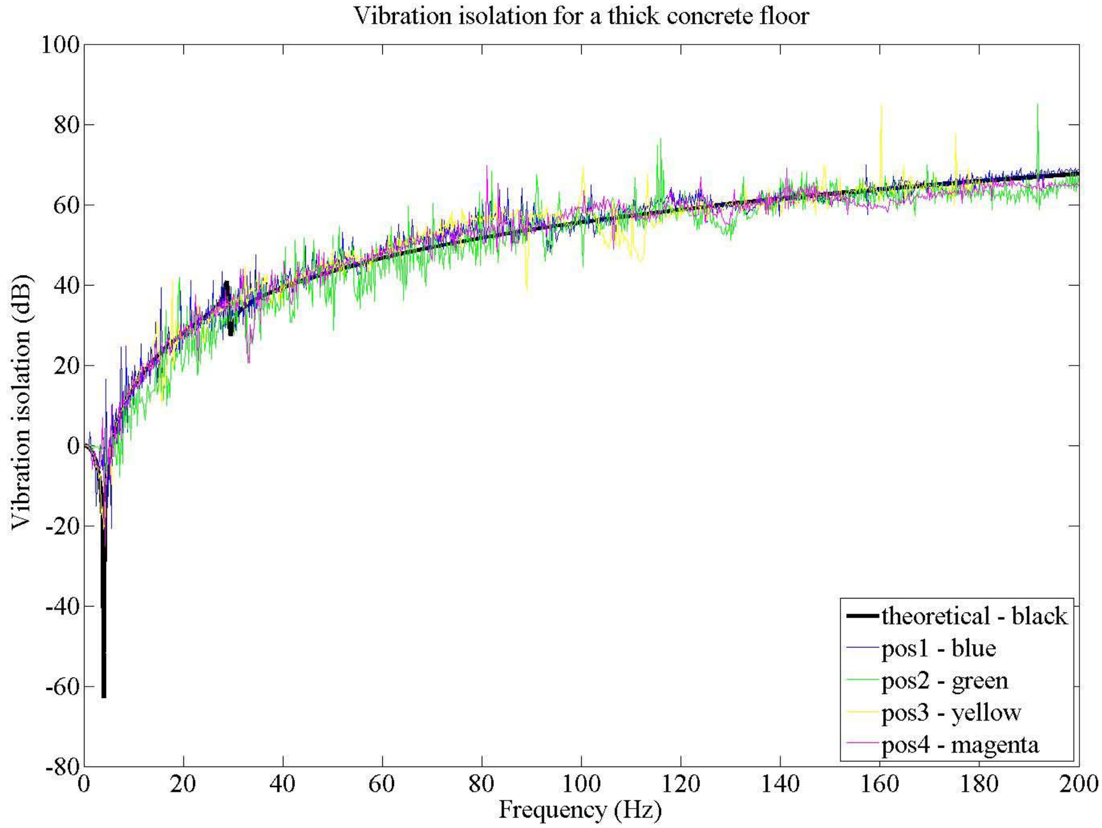

Figure 2: The anticipated vibration isolation for a thick concrete floor. The chosen parameters are the mass mobility corresponding to a mass of 200 kg, and a spring mobility corresponding to a spring stiffness yielding an eigenfrequency of 4 Hz in connection with the mass. It is compared with the calculated theoretical results for a 200 mm thick concrete floor.

3.2 A flexibly mounted lightweight floor

The point mobility of an elastically mounted lightweight floor with a weight of approximately 50 kg/m^2 has been measured and is displayed in Figure 3. The resultant vibration isolation can be viewed in Figure 4. The chosen parameters are a mass mobility corresponding to a mass of 200 kg, and a spring mobility corresponding to a spring stiffness yielding an eigenfrequency of 4 Hz in connection with the mass. The isolation is compared with the theoretical results for a 200 mm thick concrete floor supporting the same rotating mass on springs.

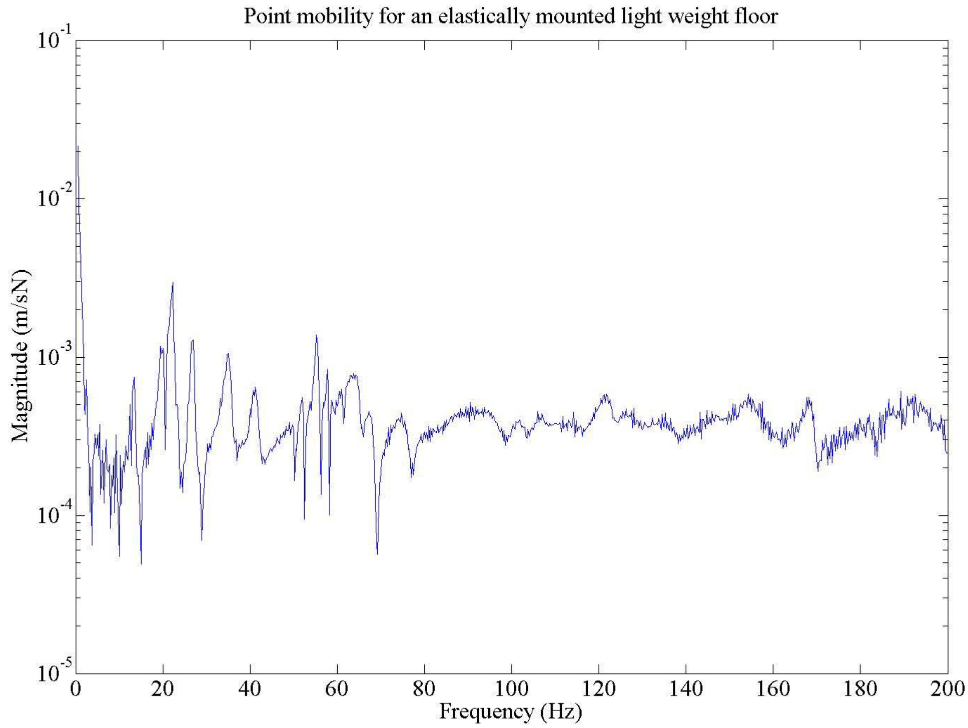

Figure 3: The measured point mobility of an elastically mounted lightweight floor (50 kg/m^2).

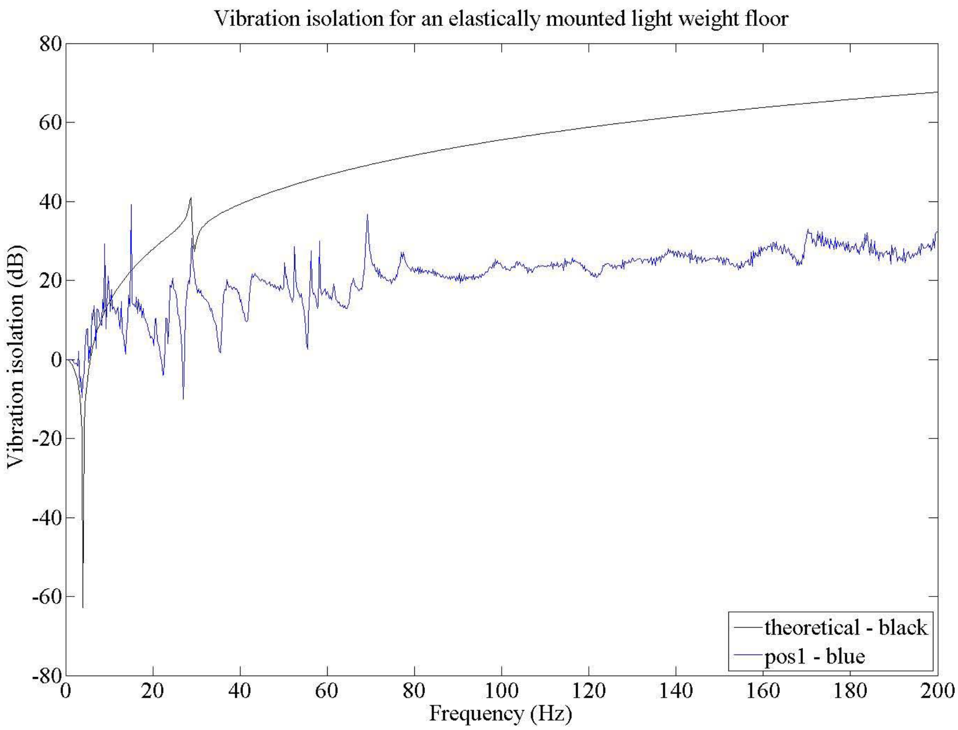

Figure 4: The anticipated vibration isolation for a lightweight floor (50 kg/m^2). The chosen parameters are a mass mobility corresponding to a mass of 200 kg, and a spring mobility corresponding a spring stiffness yielding an eigenfrequency of 4 Hz in connection with the mass. It is compared to the calculated theoretical results for a 200 mm thick concrete floor.

4 Fan Measurements

Undoubtedly the most difficult part in the assessment of structure-borne sound from fans and other machines is obtaining qualitative input data. On the noise side the situation is fairly good, but on the force side, knowledge is poor. Therefore thorough force measurements have been performed as a function of revolution speed, force and flow.

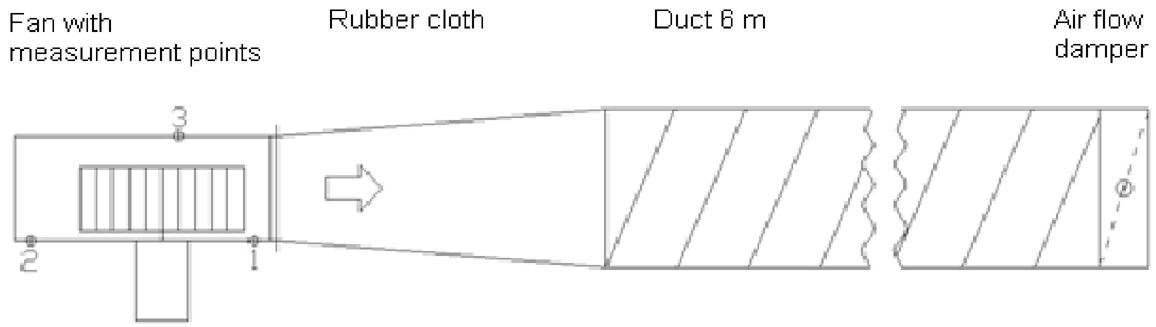

Figure 5: Overview of the fan test set up. The numbers represent the measurement points of the transducers.

Figure 6: Photograph of the test set up.

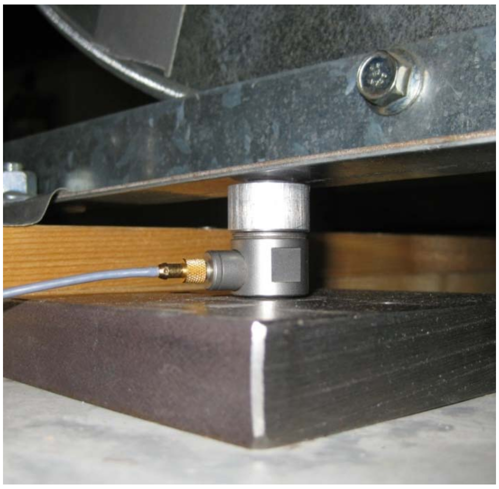

Figure 7: Mounting of the fan on the force transducers.

4.1 Force measurements results on a medium sized radial fan

The results show high disturbances at the revolution speed and at the first harmonic frequency. However, the blade passing frequency, that through experience on noise generation, was expected to be more prominent, is barely seen in measurements.
A strong dependence of the fan's disturbance forces on flow and pressure is noticed; the force increase appears to be approximately linear with increasing flow rate and decreasing pressure.
Force amplitudes, in ordinary running cases, between 10 N to 30 N have been observed for the medium sized fans. The total force to the support can be found by adding the forces on the transducers. In this setup it leads to a total force in the same magnitude as one single force. With larger phase differences the total force will be relatively small while the torque on the support can be large.

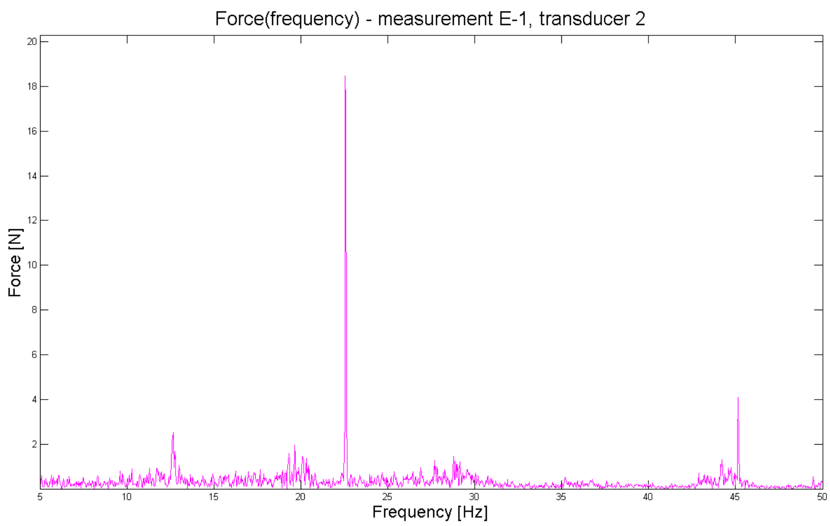

Figure 7: Force as a function of frequency in position 2. Maximum revolution speed and no flow resistance.

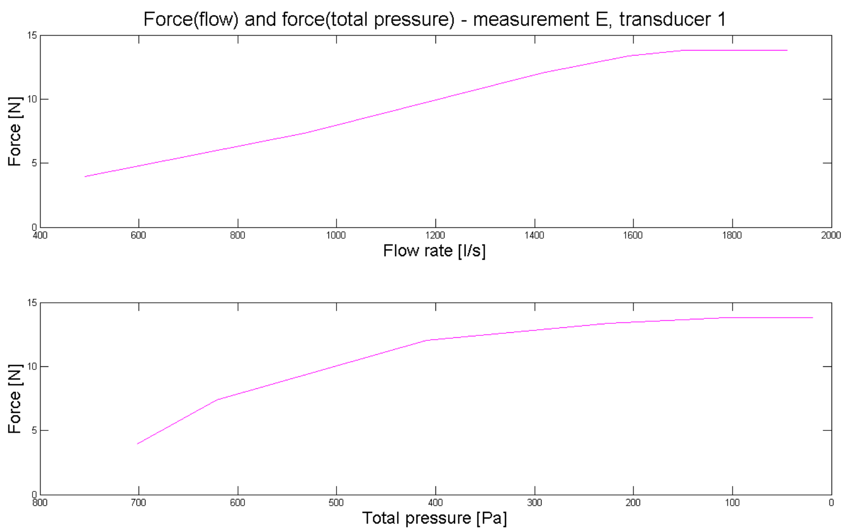

Figure 8: Force in position 1 at the revolution speed as a function of flow rate and pressure.

5 A calculation example

To illustrate how the model and knowledge from the measurements can be used in determining the resultant noise levels when placing a fan on a support, a calculation example is performed. Imagine a fan with a rotating mass of 200 kg being mounted on isolators that yield a mass spring eigenfrequency of 4 Hz. The floor is supposed to be built up by thick concrete yielding a mobility corresponding to the results in Figure 1, and an isolation according to Figure 2 of approximately 20 dB in the vicinity of 20 Hz. Assume further by looking at Figure 8 that it creates a disturbance force of 15 N in that same frequency range for a given flow. For calculation's sake, assume that the floor has a modal mass of 4000 kg and that at the revolution speed frequency, it is mass controlled and moves like a piston. The noise level below would then amount to approximately 40 dB which is an acceptable level. Had instead a light weight floor similar to that presented in Figure 3 been chosen, results could easily become some 30-40 dB higher, likely causing disturbances.

6 Conclusions

An analytical model that predicts the vibration isolation on finite and weak supports is complemented with data from floor measurements and measurements on fans. This has proved a valuable tool in assessing potential disturbances from machines.

References

[1] T. Ahlgren, Vertical force of centrifugal fans at mounting positions, *MSc thesis*, KTH Stockholm, 2009.

[2] T. Ahlgren P. Blom, Fläktars kraft- och vibrationsalstring, *Bygg & Teknik*, 3, 2009, 42-25.