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Investigations regarding the influence of static load and airflow resistance on the measurement of dynamic stiffness

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ABSTRACT

Within the scope of a research project regarding the physical properties of insulation materials made from renewables, one of PTB’s tasks is to determine the dynamic stiffness of such materials used as elastic layers. The common measurement procedure is described in DIN EN 29052-1. As a closer examination of this standard bears some open questions, several investigations have been conducted recently. This presentation focuses on two aspects: The first one is the influence of the static preload. This might be relevant for insulation materials used for thermal insulation composite systems, where, compared to floating floors, the static preload is significantly smaller. As the standardized measurement procedure is not suited for small static preloads, an alternative measurement set-up was designed to investigate this topic. The second aspect is the effect of airflow resistance on the resulting dynamic stiffness. The measurement setup as devised in the standard does not necessarily reflect the boundary conditions the material is exposed to when it is mounted in situ. The standard describes a procedure to handle the influence of airflow resistance, but this is partially incomplete and not applicable for some materials. Some theoretical considerations and measurement results regarding this topic will also be presented.

Keywords: Dynamic stiffness, insulation materials, airflow resistance, sound insulation

1. INTRODUCTION

Insulation materials are often used as elastic layers in floating floors or thermal insulation composite systems. One of the relevant parameters to predict the structure-borne or airborne sound insulation of such constructions is the dynamic stiffness of the elastic layer. For materials used in floating floor constructions, the measurement procedure to determine the dynamic stiffness is described in the international standard ISO 9052-1 (1) and the corresponding German standard DIN EN 29052-1 (2). It is one of PTB’s tasks within the framework of a research project regarding the properties of insulation materials made from renewables to measure the dynamic stiffness of a wide range of samples. The standardised measurement set-up was realised, and some preliminary tests were performed. During these tests, the standard procedure soon revealed to bear some open questions, which were already partially investigated in the past (3,4). Chapter 2 presents a brief description of the standardised procedure and some of its shortcomings. A pivotal aspect is the influence of the static preload on the dynamic stiffness. This is especially important if the insulation material is used in a thermal insulation composite system and not in a floating floor, as in this case it is exposed to significantly smaller static loads. As the standardised set-up is not suited to handle small static loads, an alternative measurement set-up was designed and tested. The alternative set-up and the obtained results are discussed in chapter 3.

If an insulation material is employed in a floating floor construction to create an elastic layer, it is contained in an approximately airtight enclosure. This is not reflected by the standardised measurement set-up, where the edges of the sample are uncovered, allowing the surrounding air to travel freely in and out. Thus, depending on the physical characteristics of the sample, the stiffness of the enclosed air may not be considered correctly. The standard contains some directions on how to approach this problem, but these are in some ways inconsistent. A detailed discussion of this problem and a suggestion for a possible workaround are given in chapter 4. Finally, chapter 5 presents some investigations regarding the influence of the excitation force.

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2. STANDARDISED MEASUREMENT PROCEDURE

For the standard procedure, a sample with the size of 200x200 mm² is prepared from the insulation material. The sample is placed on the floor or on a heavy base plate, and a steel plate with a thickness of 25 mm is put on top of the sample as mechanical load. Together with a layer of gypsum, which acts as a levelling course, the overall surface mass load adds up to 200 kg/m², which is a good representation for a typical concrete screed. Figure 1 shows a sketch of the arrangement.

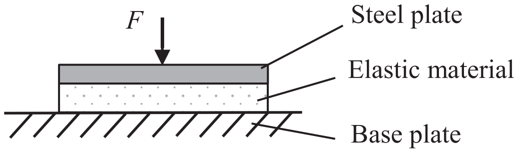


Figure 1 – Sketch of standard arrangement

As the base plate is considered stationary, the whole arrangement forms a spring-mass system, with the sample acting as a spring with a stiffness s, while the steel plate and the gypsum layer act as a mass m. By determining the resonance frequency f_r, the dynamic stiffness s of the sample, which is usually expressed per unit area, can be calculated using

$$s'_t = 4\pi^2 \cdot m'_t \cdot (f_r)^2 \qquad (1).$$

The index "$_t$" is used to indicate that the influence of the enclosed air is perhaps not considered properly, and that a correction term must be added. This will be comprehensively discussed in chapter 4.

The resonance frequency is measured by excitation of the system with a periodical force F. This must be applied in a way that the system oscillates only in vertical direction, which is usually best achieved by applying the excitation force at the middle of the steel plate. Stationary excitation (sinusoidal or noise) using a shaker or impulse excitation with a hammer are possible. When using sinusoidal excitation, the resonance frequency is determined by varying the excitation frequency while keeping the excitation force unchanged. A special procedure must be followed if the stiffness of the material is dependent on the excitation force. Chapter 5 takes a look at this topic.

3. ALTERNATIVE SET-UP FOR SMALL STATIC LOADS

3.1 Measurement Set-up

As already mentioned, an insulation material is exposed to a much smaller static load when used in a thermal insulation composite system. The measurement standard states that in this case, a variation of not more than 20 % for the dynamic stiffness, with respect to a load of 200 kg/m³, can be expected, which is regarded as negligible for practical applications. During the ongoing research project, it was nevertheless desired to verify this effect by measurements. As an excitation of a thin and lightweight load is likely to be difficult when using point excitation, an alternative measurement set-up was designed. Here, a base plate with a mass m_1 of 13 kg is mounted on a robust vibration exciter (shaker), which is capable to bear high mass loads. On this base plate, the sample and the mass load are mounted as in the standard set-up, but with an aluminium composite panel with a total mass m_2 (including the still required gypsum layer) of 0.3 kg, instead of the steel plate. The whole system is now excited by the shaker using a swept sine signal, and the measurement delivers the acceleration spectra $a_1(\omega)$ and $a_2(\omega)$. Eventually, the complex transfer function $a_2(\omega)/a_1(\omega)$ is calculated. Figure 2 shows a schematic diagram and the realisation of the alternative measurement set-up.

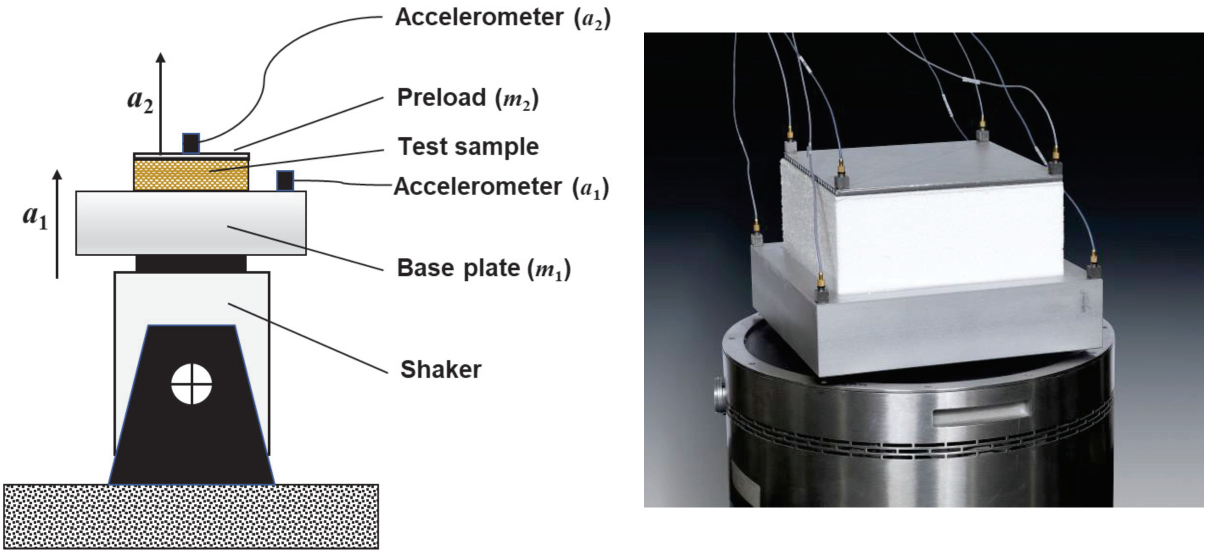


Figure 2 – Schematic diagram (left) and realisation (right) of the alternative test set-up

As wobbling and tilting movements of both the base plate and the preload cannot be completely excluded, it is generally desirable to measure the acceleration at the centre spot of the plates. As this is not possible for the base plate, the measurement is performed on all four corners of the plate, and subsequently a complex average of the obtained acceleration spectra is calculated. This approach was validated by measuring the acceleration response of the base plate at the centre and on the corners (without a test sample). The result is shown in Figure 3. Up to a frequency of 300 Hz, the averaged measurements at the corners produces the same results as the measurement at the centre. The deviations at higher frequencies are caused by the fact that the base plate does not act as a rigid body anymore.

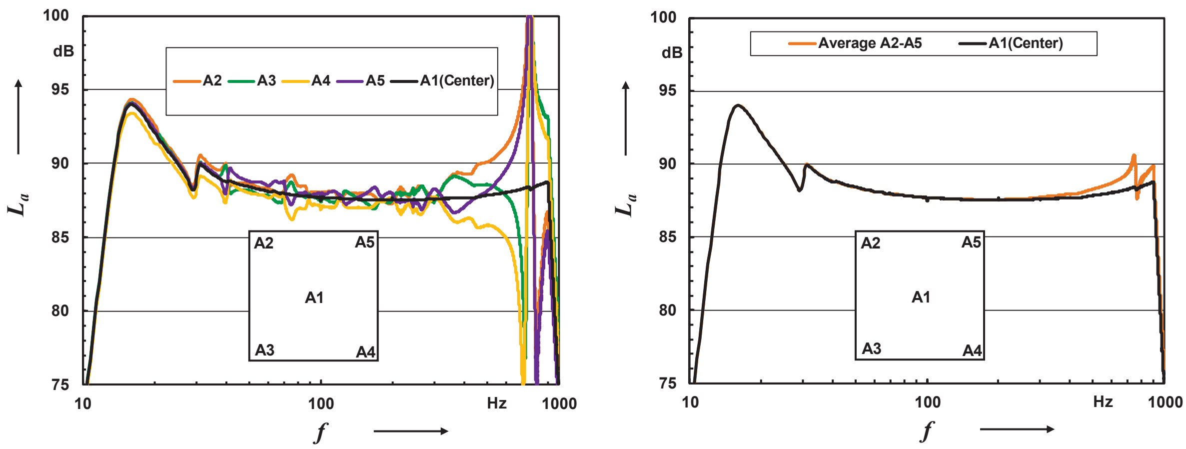


Figure 3 – Acceleration responses on the corners measured individually (left graph) and averaged (right graph) vs. response at centre of the base plate

While the procedure according to the measurement standard assumes a simple spring-mass system, we now have a slightly more complex mass-spring-mass system, which is the reason why it is not sufficient to measure the acceleration on the load plate ($a_2(\omega)$) only, but also on the base plate and to determine the stiffness from the transfer function $a_2(\omega)$ / $a_1(\omega)$. As Figure 4 illustrates, the transfer function shows a different resonance frequency, which is the correct value for this set-up.

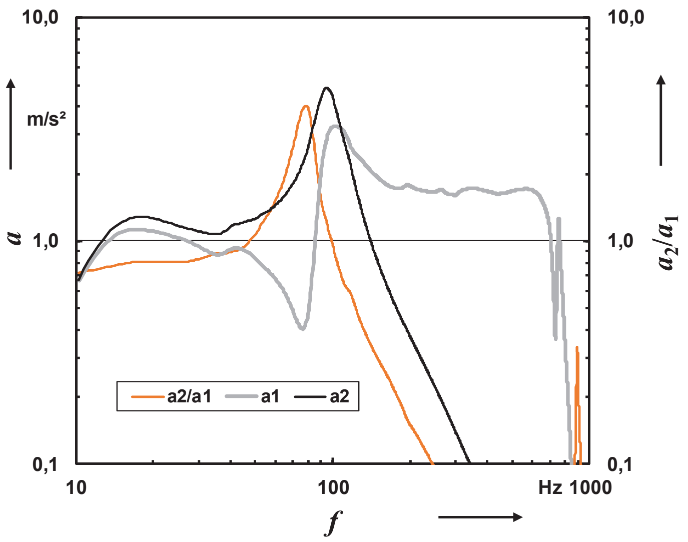


Figure 4 – Acceleration response of the base plate ($a_1(\omega)$) and the load plate ($a_2(\omega)$), and corresponding transfer function $a_2(\omega)\ /\ a_1(\omega)$

3.2 Evaluation Procedure

The dynamic stiffness may be determined the same way as in the standard procedure by detecting the resonance frequency from the transfer function. With this setup, however, it is also possible now to calculate the stiffness for every frequency ω, at least in the frequency range where the measurement set-up works properly. The simplest approach is a mass-spring-mass system with a lossless spring:

$$s'_{\mathrm{t}} = m'_2\,\omega^2\,\frac{1}{1 - a_1/a_2} \tag{2}$$

At the resonance frequency ω_{r}, the fraction on the right-hand side approaches 1, and Eq. (2) equals Eq. (1), which is used in the standard procedure. A more realistic approach is to model the test sample as a spring with losses. The resulting equation is

$$s'_{\mathrm{t}} = m'_2\,\omega^2\,\frac{1 - |a_1/a_2|\cos\varphi}{(1 - |a_1/a_2|\cos\varphi)^2 + (|a_1/a_2|\sin\varphi)^2} \tag{3}$$

Both approaches assume lumped elements (mere masses and springs). As the mass of the test sample is not negligible anymore when using small static loads, a modelling of the test sample as an expansion bar, with or without losses, can also be considered. The resulting equations can probably only be solved with a numerical approach, which was also used but will not be described in detail.

3.3 Measurement Results

At the time of writing, only one test sample (80 mm EPS foam) was tested. This is, however, a very common material used for thermal insulation composite systems. Apart from the low preload of 0.3 kg, an added mass of 2.5 kg and the steel plate as used in the standard procedure were also tested. For all measurements, the excitation force was varied in a range of 30 dB. A selection of the results for a mass load of 0.3 kg and 8 kg is shown in Figure 5 and Figure 6.

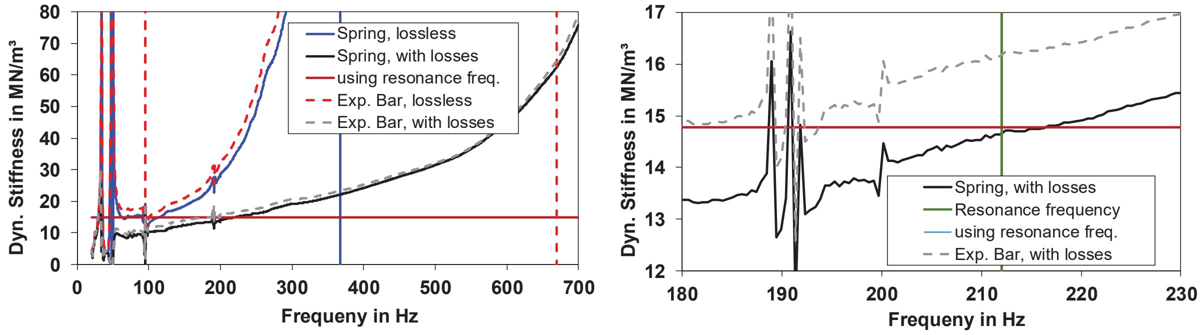


Figure 5 – Left: Measurement with m_2 = 0.3 kg mass load. Right: An excerpt of the left-hand graph without the lossless solutions

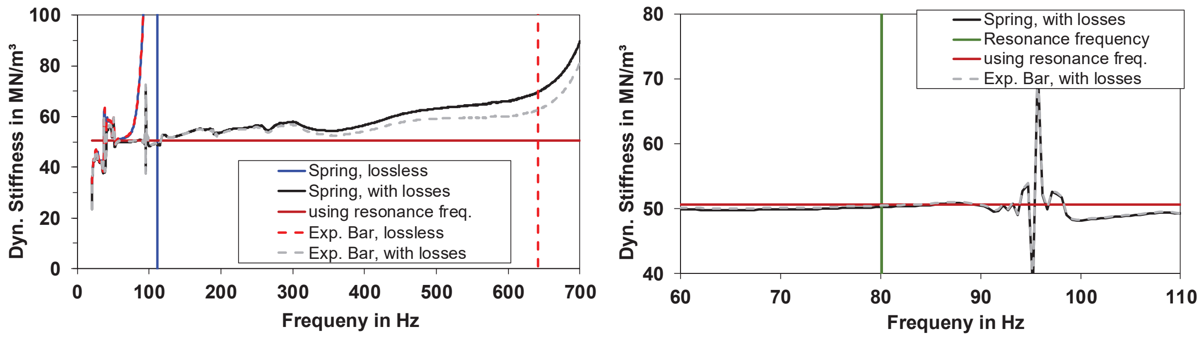
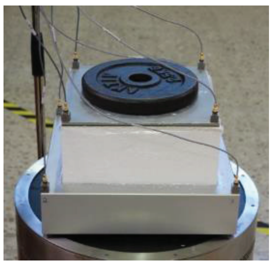
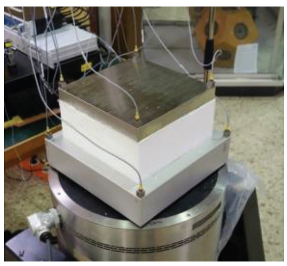


Figure 6 – Left: Measurement with m_2 = 8 kg mass load. Right: An excerpt of the left-hand graph without the lossless solutions

In most cases, the lossless solutions do not deliver reasonable results. However, if losses are considered, the “spring” and the “expansion bar” approach both do not only show very similar results over a wide frequency range, but also a good match with the dynamic stiffness obtained by using the resonance frequency according to Eq. (1). Especially with the small mass load, one could expect an even better match when the expansion bar model is used. On the other hand, the difference between the spring and expansion bar solution is quite small, and the determination of the resonance frequency also has a considerable uncertainty.

The major topic to be investigated with the alternative set-up is the relation between dynamic stiffness and mass load. Table 1 shows the results for the sample under test. The displayed stiffnesses were calculated using the resonance frequency. The measurements were performed keeping the current fed into the shaker on a constant level.

Table 1 – Relation between dynamic stiffness and mass load, EPS 80 mm

			
m_2 in kg	0,3	2,8	8,0
f_r in Hz	212,0	95,7	80,1
s_t' in MN/m³	14,8	25,6	50,7

For this test, the dynamic stiffness decreases by a factor of about 3 in relation to the standard procedure when a small mass load is employed. This is initially in contradiction to the statement in the measurement standard, where a deviation in the maximum range of 20 % is proposed. Here, of course, more research is needed to verify the observed effect.

4. INFLUENCE OF THE ENCLOSED AIR VOLUME

The current version of ISO 9052-1 (1) describes how the stiffness due to the air enclosed in the porous elastic layer should be considered if the measurement is performed according to Figure 1. This is based on work by Kraak (5) which is briefly sketched in the following.

Starting point are the equations of motion for the porous media

$$-\mathrm{grad}\, p \;=\; \chi\, \rho_0\, \frac{\partial\, \boldsymbol{v}}{\partial\, t} + \varepsilon\, \sigma\, \boldsymbol{v}\;;\;\; -\mathrm{div}\, \boldsymbol{v} \;= \frac{1}{\kappa p_0}\frac{\partial\, p}{\partial\, t} - \frac{v_p}{d\, \varepsilon} \tag{4}$$

with the fluid pressure p, fluid velocity $\boldsymbol{v}$, mean density ρ_0, static pressure p_0, porosity ε, airflow resistivity σ, structural factor χ, ratio of specific heats κ, thickness of the porous layer d and the velocity of the cover plate v_p. Using harmonic excitation with angular frequency ω gives

$$-\mathrm{grad}\, p \;=\; j\, \omega\, \chi\, \rho_0\, \boldsymbol{v} + \varepsilon\, \sigma\, \boldsymbol{v}\;;\;\; -\mathrm{div}\, \boldsymbol{v} \;=\; j\, \omega \frac{1}{\kappa p_0} p - \frac{v_p}{d\, \varepsilon} \tag{5}$$

which leads to the wave equation

$$\Delta\, p \;-\; j\, \omega \frac{\varepsilon\, \sigma}{\kappa p_0} p = -\frac{\sigma}{d} v_p \tag*{(6).}$$

Using polar coordinates, i.e. a circular cover plate, normalising the radial coordinate by

$$\alpha = \sqrt{-j}\, \sqrt{\frac{\omega\, \varepsilon\, \sigma}{\kappa p_0}} \tag{7}$$

and averaging the pressure over the cover plate finally gives

$$s'_\sigma = s'_a\; Re\left[1 - \frac{2}{\alpha\, R}\frac{J_1(\alpha\, R)}{J_0(\alpha\, R)}\right]\;;\;\; s'_a = \frac{\kappa p_0}{\varepsilon\, d} \tag{8}$$

with the cover plate radius R and the dynamic stiffness of an air layer s_a'. The same calculation was also performed for a quadratic plate in (5). The ratio between the stiffness of an air layer s_a' and the stiffness due to the airflow resistivity s_σ' is plotted as a function of the generalised coordinate $\alpha\, x$ where x is the radius for the circular cover plate and the half of the edge length for the quadratic cover plate (Figure 7, left graph). For $\alpha\, x < 1.5$, the effect of air stiffness does not change the result of the stiffness measurement according to Figure 1. Therefore, the stiffness of the final assembly (complete floor) can be calculated by adding the air stiffness to the measured stiffness in this range. With the estimates $x \approx 0.1$ m; $p_0 \approx 10^5$ Pa; $\varepsilon,\kappa \approx 1$ and $\omega \approx 200$, the condition $\alpha\, x < 1.5$ turns out to be equivalent to σ<100 kPa s/m². This is the value mentioned in (1). For larger airflow resistivities, the measurement automatically includes the effect of the air stiffness and thus, no further correction is required. ISO 9052-1 (1) states that this is the case for σ > 100 kPa s / m². Nevertheless, a national foreword in Germany (2) changes this to σ > 1000 kPa s / m² without mentioning how to handle the range between 100 and 1000 kPa s / m². Kraak (5) himself mentions in his paper the criterion $\alpha\, x > 15$, which would correspond to an airflow resistivity of 10000 kPa s / m² with the above-mentioned estimates.

To perform test measurements on this effect, a special test assembly was used at PTB. It consists of an acrylic glass frame which is mounted around the standardised test set-up with an airgap of about 1 mm (Figure 7). Measurements of the dynamic stiffness were performed for two different materials with a lateral airflow resistivity of about 40 kPa s / m² and 4 kPa s / m² once with the frame and once without the frame. The presence of the frame generally increases the measured stiffness. The increase was about 30 % for the first material and about 60 % for the second. This follows the general theory that the low resistivity material is more affected. Nevertheless, further detailed investigations are necessary.

Another problem is the measurement of the airflow resistivity, which in this case must be determined in lateral direction. Unfortunately, most existing measurement devices are only suited for measurements in orthogonal direction, which may deliver significantly different results.

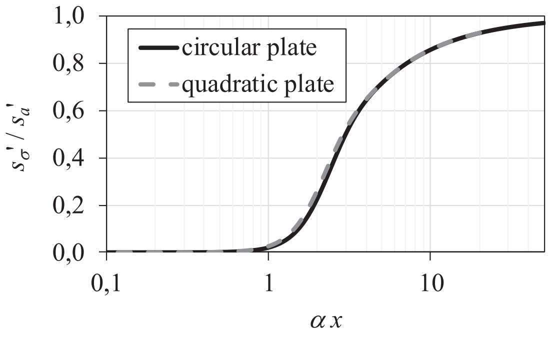


Figure 7 – Left: Calculated ratio of air stiffness and stiffness due to resistivity. Right: Test assembly with adjustable frame

5. INFLUENCE OF THE EXCITATION FORCE

The dynamic stiffness of most of the common insulation materials varies with the excitation force that acts on it. For sinusoidal excitation, the measurement standard requires the measurement to be performed in a range of 0,2 N $\leq F \leq$ 0,8 N for s' > 50 MN/m³ or 0,1 N $\leq F \leq$ 0,4 N for s' $\leq$ 50 MN/m³. No hint is given if the required forces relate to peak or rms values. The measurement must be performed at three different excitation forces in the respective range, and the final value for the resonance frequency is then found by extrapolation to 0 N. This approach has several drawbacks. As shown in Figure 8, the sensitivity of the resonance frequency to variations of the excitation force may be quite pronounced when approaching 0 N. Furthermore, an excitation force of 0 N is obviously unrealistic for built-in situations. A previous publication of the authors (3) offers a pragmatic suggestion to perform the measurement with a peak force of 1 N. As the extrapolation to 0 N represents some sort of “safe side” approach, a question raises about the effect on the stiffness if 1 N is used. This was investigated for different insulation materials; the results are shown in Table 2.

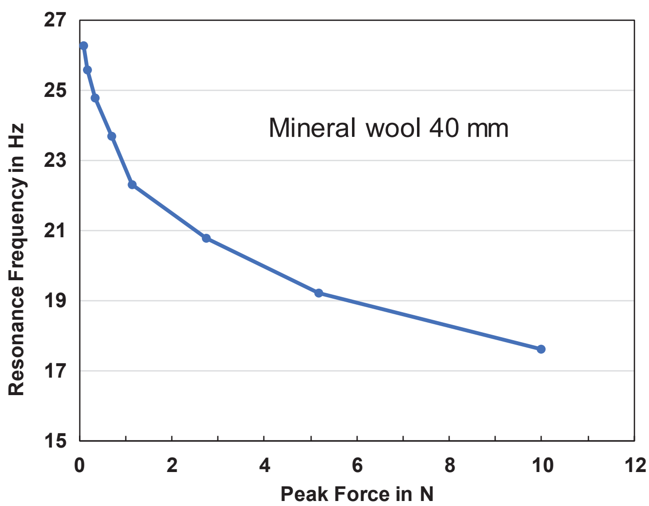


Figure 8 – Dependency of the resonance frequency to the excitation force

Table 2 – Comparison of extrapolation to 0 N vs. measurement at 1 N

Material	d in mm	$s'_{\mathrm{t,0N}}$ in MN/m³	$s'_{\mathrm{t,1N}}$ in MN/m³
XPS	30	295	290
Minera wool 1	40	3,4	2,7
Mineral wool 2	40	34,8	29,1
Mineral wool 3	40	5,7	4,5
Coconut fibre	35	3,5	2,7

As expected, the dynamic stiffness decreases up to 30 % with an excitation force of 1 N. As this force is still at the very low end of a force range that can be expected in built-in situations (3), this can still be considered as a conservative estimation. Regarding the given problems that come with the extrapolation to 0 N, a measurement with 1 N still appears to be a viable option.

It is still unclear, however, how to determine the excitation force at the resonance frequency when using noise or even impulse excitation signals, even though this is permitted in the standard.

6. CONCLUSIONS

An alternative measurement set-up for low static loads was successfully designed and tested. The first results indicate that the dependency of the dynamic stiffness with respect to static load is probably larger than expected, but more measurements are necessary to verify this observation.

The theoretical background regarding the effect of the enclosed air volume in relation to the airflow resistance was discussed. The current regulations in the measurement standard regarding this effect are in some points confusing and inconsistent. A possible solution would be the simulation of the in-situ situation by covering the edges of the sample with an adjustable frame. Measurement results using this approach were presented.

The handling of the interdependence between dynamic stiffness and excitation force according to the standard is also problematic. It has been shown that the use of a fixed excitation force of 1 N could be a realistic solution.

The results of this contribution could be considered in the next revision of the measurement standard.

ACKNOWLEDGEMENT

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