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CHALLENGES OF THE MODEL-BASED DYNAMIC CALIBRATION OF FORCE TRANSDUCERS – A CASE STUDY

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Abstract – Investigations of the model-based dynamic calibration of a force transducer of high bandwidth revealed new challenges for parameter identification. This paper discusses a more generalized mechanical model of the calibration set-up employed. Based on new experimental sine and shock force data, the improved model is studied and its parameters are identified. It is shown that the proposed model is capable of linking the calibration results of both calibration methods to a much better degree.

Keywords: dynamic modelling, parameter identification, sine force calibration, shock force calibration

1. INTRODUCTION

The measurement of dynamic forces is widely used in many areas of industry. Increasing demands on measurement accuracy have to face the fact that dynamic force measurements are still purely based on static calibrations, not to mention the lack of documentary standards or commonly accepted guidelines for dynamic measurements. For this reason, the establishment of traceable dynamic measurements is an important metrological topic, which was recently emphasized by a European Metrology Research Programme (EMRP) joint research project on the traceable dynamic measurement of mechanical quantities [1, 2].

The general approach of a model-based calibration will be followed in which the dynamic behaviour of the force transducer in a given mechanical environment is described by a lumped-mass model using a series arrangement of spring-mass-damper elements. The transducer itself is characterized by model parameters that describe the dynamically relevant distribution of mass, stiffness and damping. Four parameters are usually used: head mass m_{H}, base mass m_{B}, stiffness k, and damping d. The parameters of interest may be identified from dynamic measurements by fitting modelled and measured force data.

The main purpose of the proposed approach is the characterization of the transducer's dynamic behaviour independent of the given experimental set-up, i.e. its mechanical environment or its type of dynamic force excitation, e.g. shock or sine.

Initial experiences with the parameter identification of various strain gauge force transducers demonstrated some discrepancies between results from shock and sine force experiments that need to be explained. The analysis of shock force data showed that consistent results can be obtained for transducers that respond with strong signal ringing [3]. However, the previous parameter identification approach failed when the force transducer responded with smooth shock pulses, which was the case for a transducer of high bandwidth and comparably long shock duration.

2. CHALLENGES

In a given mechanical set-up, e.g. a dynamic measurement application or a dynamic calibration device, the elastically coupled structural mass components of the transducer as well as of its mechanical environment generate inertia forces that influence the dynamic measurement behaviour. The specific models of the dynamic calibration set-ups have to be appropriately developed in order to be able to unambiguously identify the parameters of the transducer under test from the measured data. First experiences obtained with strain gauge force transducers of differing design, size, weight and mechanical coupling revealed several challenges for the model-based calibration.

This case study is focused on dynamic measurements with the HBM U9B / 1 kN strain gauge transducer, which has a mass of about 63 g (see Fig. 1). In particular, this small transducer demonstrated with its large bandwidth as well as its resonance behaviour that the standard sine and shock force calibration set-ups have to be modified to obtain the desired dynamic information. Furthermore, the previous investigations showed that the formerly published model descriptions have to be extended in a more generalized fashion to be able to cover the observed behaviour of the different test conditions.

The following challenges have been identified:

1. Previous shock force measurements showed that the dynamic models have to account for the elastic coupling at the base of the transducer [3, 4]. A weak coupling can result in a low-frequency resonance of the transducer housing which may dominate the dynamic behaviour.
2. A force transducer may also respond with more than one axial resonance depending on its structural mass distribution [5]. This behaviour was experimentally observed for the selected transducer applied to shock excitation. A modal analysis using the finite elements method finally revealed that the transducer exhibits two axial resonances. Figure 2

depicts the calculated resonances for the transducer fixed at its base: a resonance of the housing at about 11 kHz, and a resonance of the upper rod (head mass) at about 27 kHz.

3. The experimental shock force data may not contain sufficient dynamic information to identify the model parameters of a transducer of high bandwidth. This experience was made with the HBM U9B / 1 kN investigated at the 20 kN shock force calibration device. In this case, the shock pulses are too wide to substantially excite the transducer's modal vibrations.
4. The parameter identification from shock force data may be hampered by modal vibrations from the shock-excited mass bodies of the calibration device [3]. The selected transducer exhibits a resonance frequency well above the disturbing components, so that the modal vibrations cannot be removed by low-pass filtering the measurement data for the parameter identification process.
5. For sine force calibrations performed on a vibration exciter system, the transducer is usually equipped with large load masses which drastically lower the system's resonance frequencies. With excitation frequency ranges limited to about a few kilohertz, just one resonance could be experimentally observed so far, although the transducer possesses more than one axial resonance mode.

Fig. 1. Force transducer HBM U9B / 1 kN with a locknut at each bolt end.

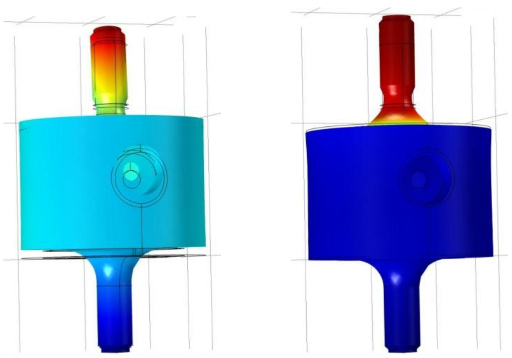

Fig. 2. FE simulation of the first two axial resonances of the force transducer HBM U9B / 1 kN fixed at the very end of the lower threaded bolt: colour-coded elongation of the transducer housing at the resonance frequency of 10.6 kHz (left), vibration of the head mass at 27.2 kHz (right).

3. NEW EXPERIMENTS

To cope with the above-mentioned difficulties, several shock and sine force experiments were performed with the selected transducer using modified testing conditions. The new experiments finally achieved data better suited to the subsequent parameter identification process.

3.1. Shock force measurements with small pendulum mass

The measurements at the 20 kN shock force calibration device show that pulse durations in the order of one millisecond are much too long to substantially excite modal excitations of the small HBM U9B / 1 kN.

To generate shorter pulses, the device's airborne impact mass of 10 kg was replaced by smaller pendulum masses of 89 g and 7 g, respectively [5]. Figure 3 compares the two shock pulses achieved with the impact mass of 10 kg and the pendulum mass of 7 g. The former pulse width of 1.2 ms drops to less than 0.1 ms with the pendulum. It is seen that modal vibrations are excited to a much greater extent for shorter pulses.

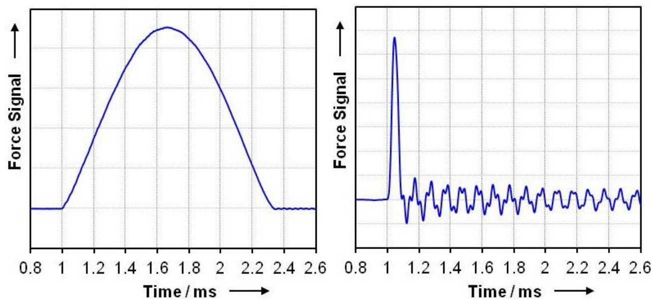

Fig. 3. Shock response of the force transducer HBM U9B / 1 kN using different impact masses, 10 kg (left), 7 g (right).

3.2. Shock force measurements with additional load mass

Another method to strongly excite the modal vibrations of the small force transducer is the application of additional load masses which increase the transducer's head mass and shift the resonances towards lower values [5]. Figure 4 shows the transducer during a shock force experiment at the 20 kN shock force calibration device. An additional load mass made of brass is fixed at the free upper rod end at which the impact force is introduced.

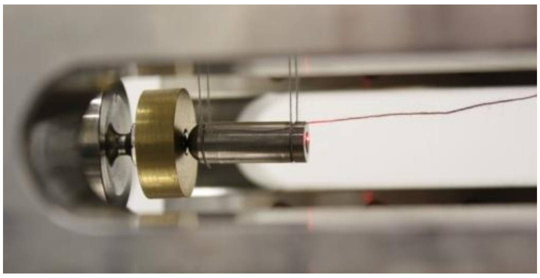

Fig. 4. Shock force tests at the 20 kN shock force calibration device using a pendulum of 7 g and an applied load mass.

Shock force tests with additional head mass obtained data better suited to the parameter identification process, but it was actually proved that this transducer is more complex and its model description has to be refined. These findings were confirmed by the above-mentioned finite element analysis (cf. Fig. 2).

Shock measurements with varied additional load masses are presented in Fig. 5. The diagrams show the discrete Fourier transform (DFT) of the impact-excited signal ringing of the acceleration a_2 of the 10 kg reaction mass body and of the transducer output force F. The excitation was carried out with the pendulum of 89 g. The data shows two load-dependent resonances, one weakly dependent at about 10 kHz, and the other strongly dependent coming down from 28 kHz, as well as several fixed resonances due to vibrational modes of the reaction mass body.

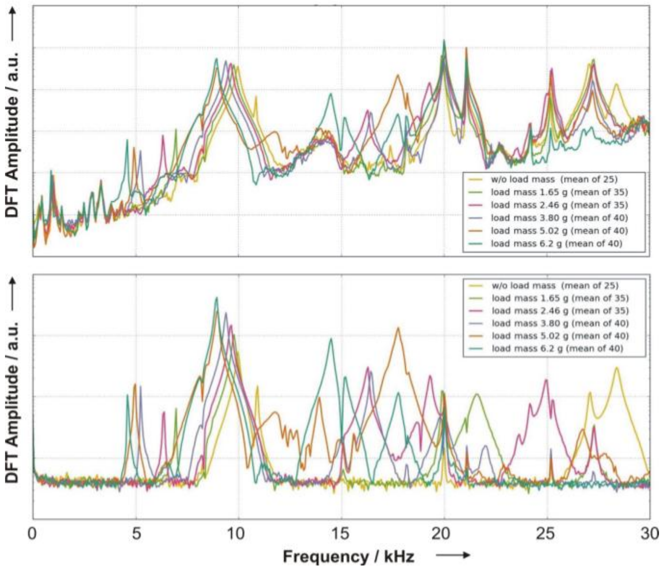

Fig. 5. DFT of the shock-excited ringing of acceleration a_2 (top) and transducer output F (bottom) obtained with different additional load masses measured at the 20 kN shock force calibration device.

The diagrams demonstrate that it is difficult to extract the correct resonance frequencies for the parameter identification process due to the complexity of the resonance behaviour so far not explained by the applied models. To investigate the effect of the increase of the transducer head mass, the variation of the additional load mass had to be kept small so as not to get confused with the disturbing fixed resonances.

For more clarity of this modal structure and also for the purpose of understanding the sine and shock excitation responses, tests with varied load mass were also performed with sine excitation using a high-frequency shaker system.

3.3. Sine force measurements at the HF vibration exciter

Sine force calibrations were carried out at the high-frequency (HF) acceleration standard measuring device of PTB in order to confirm the modal behaviour observed in the shock force experiments. Furthermore, it was necessary to gain experience with small load masses to fully understand the dynamic behaviour and to explain the discrepant values of the parameter identification so far observed in the different experiments.

Figure 6 shows the force transducer HBM U9B / 1 kN with an additional load mass mounted at the platform of the HF shaker. Two laser vibrometers measure the reference acceleration of the base picked up at the upper surface of the adapter used. A set of load masses made of brass with 14 logarithmically graded values from 0.3 g to 216 g was used for the sine force experiments. All screw connections were fastened with defined torque. The transducer was mounted at the shaker only once to guarantee – as far as possible – a constant mounting stiffness during all experiments.

Fig. 6. High-frequency sine force tests with an additional load mass at the HF acceleration standard measuring device.

Figure 7 presents the measured amplitude response, i.e. the amplitude ratio of the transducer output and the base acceleration, of the force transducer with and without applied load masses. The measurements clearly show two resonances, which excellently confirm the behaviour seen in the shock force experiments. The upper diagram demonstrates that a large load mass predominantly affects the lower resonance, whereas a very small mass has the greatest influence on the higher resonance.

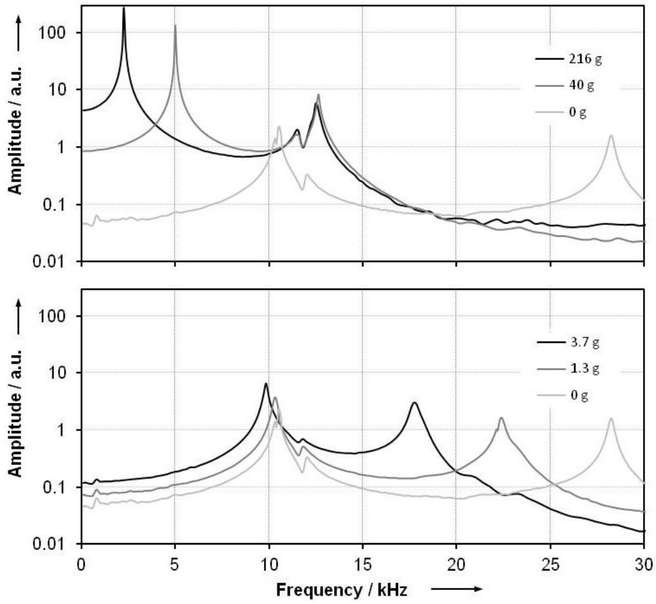

Fig. 7. Amplitude response (ratio of transducer output and base acceleration) of the HBM U9B / 1 kN obtained with different load masses measured at the HF acceleration standard measuring device.

For sine force calibrations, the force transducer is usually loaded with large load masses in the range of several kilograms, which drastically lower the system's resonance frequencies. With a comparably low upper frequency limit of the vibration exciters of about a few kilohertz [6, 7], just one resonance could be observed so far, although the investigated small force transducer exhibits the above-mentioned multi-mode behaviour (see Fig. 12).

4. MODELLING AND PARAMETER IDENTIFICATION

The discrepant values from the parameter identification with sine and shock force data, in particular for the transducer of this case study, point out that the previous model descriptions have to be modified in order to better cover the different test conditions.

In the following, data from shock force experiments as well as sine force excitations will be analysed to determine the model parameters of the transducer under test and to verify the proposed model.

4.1. Analysis of shock force measurements

With regard to the model describing the shock force calibration it is proposed to model the connection between load mass (load button) and the transducer's head mass as elastically coupled. This elastic coupling was originally applied only for the sine force calibration with large load masses [6]. In contrast, sine force models have not yet considered an elastic coupling at the base so far. In order to obtain a sound understanding of the dynamic behaviour independent of the experimental set-up, the model of the force transducer requires elastic couplings at both ends of the transducer.

Figure 8 presents the model of the 20 kN shock force calibration device with the mounted force transducer that will be considered in this paper. The force transducer marked in blue is elastically coupled at both ends to its mechanical environment, which is the base adapter connecting the transducer to the reacting mass body, and the load mass on which the impact force is applied.

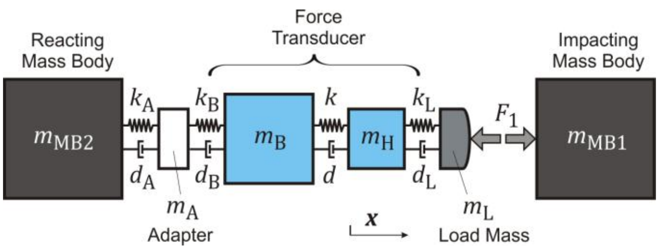

Fig. 8. Model of the 20 kN shock force calibration device with mounted force transducer and additional load mass.

Having identified the respective resonances observed in the DFT analysis of Fig. 5, the stiffness parameters of interest can be determined from the eigenvalues of the characteristic system matrix neglecting damping [3].

In general, the dynamic behaviour of the model components is described by a system of linear ordinary differential equations (ODE) derived from the equilibrium of forces at each mass element as

$$\boldsymbol{M}\ddot{\boldsymbol{x}} + \boldsymbol{D}\dot{\boldsymbol{x}} + \boldsymbol{K}\boldsymbol{x} = \boldsymbol{L}\,, \tag{1}$$

where $\boldsymbol{x}$, $\dot{\boldsymbol{x}}$ and $\ddot{\boldsymbol{x}}$ are the motion vectors (displacement, velocity, acceleration), $\boldsymbol{M}$, $\boldsymbol{D}$, and $\boldsymbol{K}$ denote the matrices for mass, damping and stiffness, and $\boldsymbol{L}$ is the load vector.

The model illustrated in Fig. 8 may be simplified if the adapter is more rigidly coupled to the reacting mass than to the transducer comparing the different mounting conditions (contact area, thread size, mounting torque), see Fig. 9.

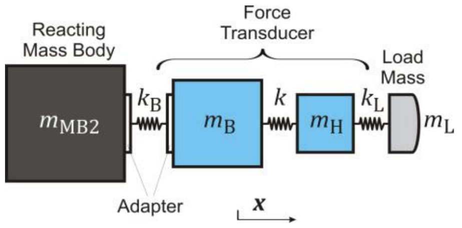

Fig. 9. Series arrangement of four model masses to describe the resonance frequencies of the 20 kN shock force calibration device.

Neglecting damping, the different resonance frequencies are calculated from the eigenvalues of the characteristic system matrix.

$$\boldsymbol{f}_{\mathbf{Res}} = \frac{1}{2\pi}\sqrt{\mathrm{eig}(\boldsymbol{M}^{-1}\boldsymbol{K})}\,. \tag{2}$$

Nevertheless, one has to keep in mind that the chosen uniaxial lumped-mass models can in principle neither model bending modes nor elastic modes of three-dimensional bodies. This means that some of the observed resonances (cf. Fig. 5) may not be described, such as the fixed resonances at higher frequencies which are attributed to elastic modes of the cube-shaped reacting mass body of 10 kg.

4.2. Analysis of sine force measurements

Regarding the mechanical set-up for sine force calibrations, the proposed model takes the form depicted in Fig. 10.

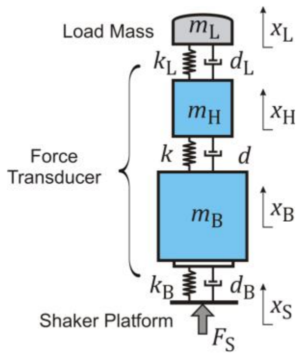

Fig. 10. Model of the sine force calibration device with mounted force transducer and additional load mass.

The vibration exciter generates a sinusoidal acceleration $\ddot{x}_{\mathrm{S}}$ of the shaker platform which introduces a sinusoidal input force F_{S} at the base of the elastically coupled force transducer given by

$$F_{\mathrm{S}} = d_{\mathrm{B}}\,\dot{x}_{\mathrm{S}} + k_{\mathrm{B}}\,x_{\mathrm{S}}\,. \tag{3}$$

The mechanical model of the sine force calibration set-up is described by the ODE system (1), with the system components

$$\boldsymbol{M} = \begin{bmatrix} m_{\mathrm{L}} & 0 & 0 \\ 0 & m_{\mathrm{H}} & 0 \\ 0 & 0 & m_{\mathrm{B}} \end{bmatrix}, \tag{4a}$$

$$\boldsymbol{D} = \begin{bmatrix} d_{\mathrm{L}} & -d_{\mathrm{L}} & 0 \\ -d_{\mathrm{L}} & d_{\mathrm{L}} + d & -d \\ 0 & -d & d + d_{\mathrm{B}} \end{bmatrix}, \tag{4b}$$

$$\boldsymbol{K} = \begin{bmatrix} k_L & -k_L & 0 \\ -k_L & k_L + k & -k \\ 0 & -k & k + k_B \end{bmatrix}, \tag{4c}$$

$$\boldsymbol{x} = [x_L,\ x_H,\ x_B]^T , \tag{4d}$$

and

$$\boldsymbol{L} = \begin{bmatrix} 0 \\ 0 \\ d_B \dot{x}_S + k_B x_S \end{bmatrix} . \tag{4e}$$

The output signal U_F of the force transducer is proportional to the elongation of the transducer's measuring spring k and is given by

$$U_F = \rho(x_H - x_B) , \tag{5}$$

where ρ denotes a scaling factor.

The sought transfer function of force signal U_F and input acceleration $\ddot{x}_S$ is expressed in the Laplace domain as

$$H_{FS}(s) = \frac{U_F(s)}{\ddot{x}_S(s)} = \frac{\rho(x_B - x_H)}{s^2\, x_S} \quad . \tag{6}$$

Neglecting damping, this simplifies to

$$H_{FS\ \mathrm{undamped}}(s) = \frac{A(s)}{B(s)}$$

with (7a)

$$A(s) = \rho\, k_B(m_H + m_L)\left[k_L + \frac{m_H m_L}{m_H + m_L} s^2\right] , \tag{7b}$$

$$B(s) = -k^2(k_L + m_L s^2) + [k + k_B + m_B s^2] \cdot [-k_L^2 + (k + k_L + m_H s^2)(k_L + m_L s^2)] . \tag{7c}$$

Figure 11 presents the comparison of the amplitude responses of the measured and fitted models for greatly differing load masses (216 g, 26 g, 0.3 g). The fitting process applied different weighting procedures – no weighting and weighting with emphasis on an interval of +/-500 Hz centred around the respective resonance frequency. It is seen that the fitting result is sensitive to the chosen weighting scheme, with larger load masses being less sensitive and modelled frequency responses showing better agreement.

Future research will focus on the optimization of the presented parameter identification and the employed fitting methods. In addition, the causes of the deviations between the measured and modelled frequency responses will be investigated. This will include the explanation of parasitic resonances (cf. Fig. 7), which may affect the fitting process, as well as the investigation of some observed dependency on the fastening torque of the load masses.

The simultaneous fitting of measurements obtained with different load masses is further expected to improve the parameter estimation. That is, the results shown in Fig. 11 were obtained by fitting the model to the individual frequency response amplitudes. On the one hand, this allows a cross-validation of the estimation result by comparing the estimates with those from other measurements. On the other hand, not all measurements show the same sensitivity to changes in the parameter values. Thus, a fit to an ensemble of measurements may allow a more robust parameter estimate with improved consistency for different load masses.

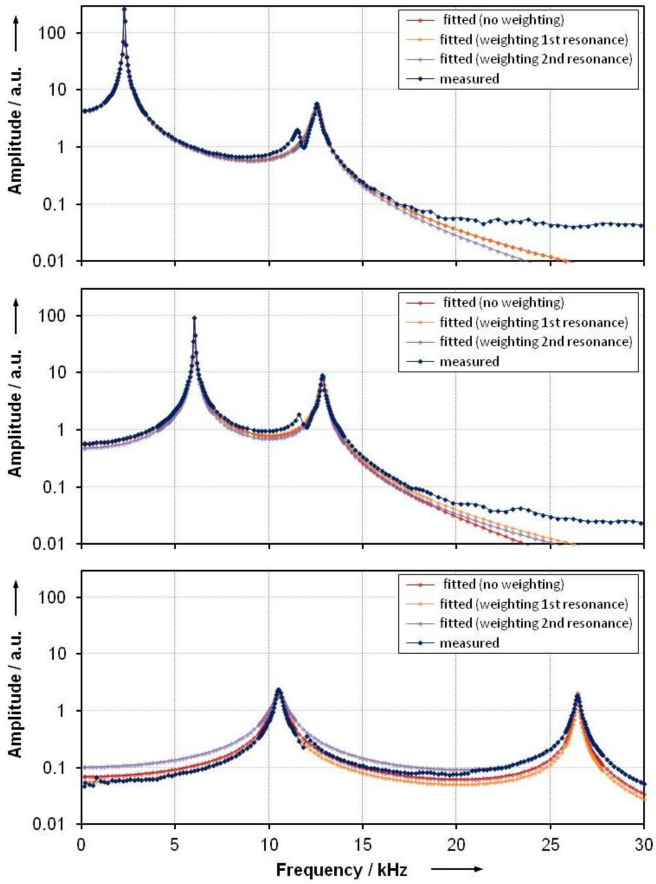

Fig. 11. Comparison of fitting procedures for the parameter identification from HF sine force measurements investigating the force transducer HBM U9B / 1 kN applied with different load masses of 216 g (top), 26 g (centre), and 0.3 g (bottom).

4.3. Comparison and discussion

Figure 12 compares measured and modelled resonance frequencies of the HBM U9B / 1 kN applied with different load masses. The diagram visualizes the three modelled resonances in dependency on the load mass. It is seen that the highest resonance stays well above 50 kHz and thus cannot be observed experimentally. The lowest resonance – the vibration of the transducer housing – starts at about 10 kHz and significantly decreases with increasing mass for loads greater than about 10 g. The second resonance starts at 28 kHz and drops to about 11 kHz for increasing loads.

The figure further displays the resonance frequency calculated with a transducer model of only one degree of freedom (dotted line) given by

$$f_{\mathrm{Res}} = \frac{1}{2\pi}\sqrt{k_{\mathrm{eff}}/m_{\mathrm{L,eff}}} , \tag{8}$$

where k_{eff} denotes the effective stiffness of the series connection of the mounted force transducer with applied load mass, and $m_{\mathrm{L,eff}}$ denotes its effective mass value. For large load masses, a one-mass model is typically applied, because only one resonance is visible in the measurable frequency range. However, this model significantly deviates for smaller load masses and is insufficient to model the full range of load masses or to agree with parameter identifications with small or without load mass, e.g. from shock force experiments.

The resonance frequencies observed in the HF sine force experiments (red circles) and in the previous shock force tests [5] (black crosses) are plotted in the diagram. In addition, the red square represents a sine force calibration with a large load mass of 1 kg that was performed at the Spanish national metrology institute CEM [8].

The measured values from the different shock and sine force experiments agree very well with the simulated values given by the proposed model with three degrees of freedom. For larger load masses, the second resonance presents some deviations that might be explained by an overestimated value of mass m_B due to the comparably thick cable connection that severely affected the determination of the transducer's total mass by weighing. Regarding the shock force results, one should consider that the differing coupling conditions possibly resulted in a weaker coupling stiffness k_B between the base mass and the adapter.

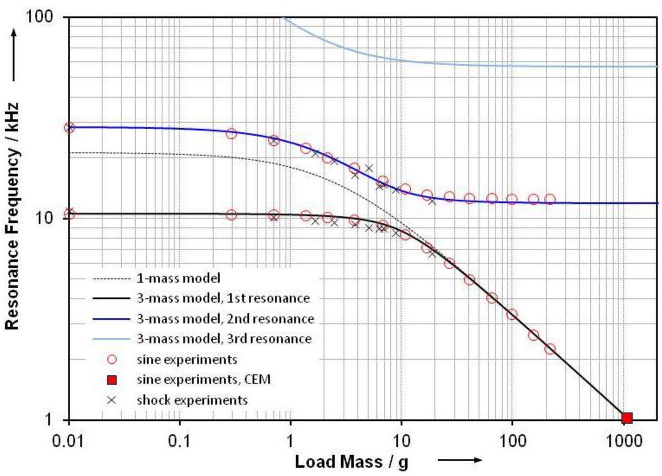

Fig. 12. Comparison of measured and modelled resonance frequencies of the HBM U9B / 1 kN with different load masses.

5. CONCLUSIONS

The approach of a model-based dynamic calibration of force transducers is employed to describe the dynamic behaviour of force transducers in a given application irrespective of the mechanical calibration set-up. In order to link results from shock and sine force calibrations, new experimental investigations are presented focusing on measurements with a small strain gauge transducer of high bandwidth. It was shown that several challenges have to be solved to identify the model parameters with consistent results independent of the calibration methods used. This paper presents a more generalized mechanical model which is capable of linking the parameter identification results obtained with different experimental methods to a much better degree than before.

The chosen model approach correctly describes the observed resonance behaviour of a force transducer with small and large load masses in shock and sine force calibration measurements. In particular, for the first time the proposed model consistently relates the sine force measurements with large load masses to those with small or zero load mass, and those from shock force experiments.

However, future research is required to investigate the additional resonances observed in the sine and shock force experiments and to minimize their influence on the fitting results. In addition, the numerous influences on the parameter identification process and the evaluation of the measurement uncertainties for parameter identification will be the focus of future investigations.

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