INVESTIGATIONS FOR THE MODEL-BASED DYNAMIC CALIBRATION OF FORCE TRANSDUCERS BY USING SHOCK FORCES

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Abstract: To investigate and validate the approach of a model-based dynamic calibration of force transducers by using shock excitations, numerous measurements were performed at PTB's 20 kN primary shock force calibration device. The tests included several strain gauge force transducers of greatly differing structural design, size, weight and mechanical coupling. Previous studies proved modal oscillations of the measurement set-up of various origins that limit the usable bandwidth. Their effect on the modal-based calibration method with its data analysis procedures, which are to be developed, is further investigated and discussed in this paper.

Keywords: dynamic calibration, impact shock force, dynamic modelling, shock pulse analysis.

1. INTRODUCTION

Dynamic force measurements are widely used in many industrial areas, and the increasing demands on measurement accuracy have set new metrological challenges. But still, traceability for dynamic measurements is solely based on static calibrations, and documentary standards or commonly accepted guidelines for dynamic measurements do not exist. For this reason, the establishment of traceable measurements under dynamic conditions is a highly relevant topic. This importance is emphasized by a current EMRP (European Metrology Research Programme) joint research project dedicated to the traceable dynamic measurement of mechanical quantities [1].

In this context, the general approach of a model-based calibration methodology will be followed in which the dynamic behaviour of the force transducer in a given mechanical calibration set-up is described by an appropriate model consisting of a series arrangement of spring-mass-damper elements. The characteristic dynamic model parameters of the force transducer – i.e. values describing its distribution of mass, stiffness and damping - have to be determined. By fitting modelled and measured shock force data, the parameters of interest may be identified from the dynamic measurements. Considering the fact that the measurement data may exhibit modal oscillations of the mechanical set-up of various origins, which was found in previous experimental studies supported by finite element simulations [2], the development and selection of adequate

methods and procedures to analyze the shock force data is of great importance.

2. EXPERIMENTAL TESTS

2.1 Devices under test

Within the scope of the above-mentioned EMRP project, several strain gauge force transducers of greatly differing size, weight, mechanical design and adaptation were investigated at PTB's 20 kN primary shock force calibration device. All transducers feature a threaded bolt with spherical end face, and can measure compression as well as tension forces. This choice also allows calibrations with sinusoidal forces for comparing the different dynamic results.

Traceability of shock force was realized by the determination of mass (from weighing) and acceleration (by means of laser vibrometers). Further information about this device is given in [3]. A recent modification of its measurement geometry now enables on-axis vibrometer measurements similar to those obtained at the larger 250 kN calibration device [4]. As this geometry is not susceptible to parasitic rotational vibrations possibly excited by the impact, the quality of the interferometric acceleration signals has been improved.

Two selected force transducers and their adaptation to the cube-shaped 10 kg mass body of the calibration device are shown in Fig. 1.

Figure 1: Mounted strain gauge force transducers:
HBM U9B / 1 kN (left), Interface 1610 / 2.2 kN (right)

The small HBM U9B / 1 kN has a mass of about 63 g and uses a measuring body with a flexing diaphragm applied with strain gauges that ends in the threaded bolt. The mass of this upper part (head mass) is less than 3 g. The manufacturer specifies a fundamental resonance of 24 kHz.

In contrast, the large Interface 1610 / 2.2 kN is a shear beam strain gauge transducer of more than 1.5 kg (including the connector) and a diameter of 105 mm. This transducer hardly fits into the squared air bearing of 108 mm clearance and clearly marks the maximum size for the 20 kN shock force calibration device.

Former tests with a larger Interface transducer model of similar design showed that the shock response might be predominately influenced by the comparably low coupling resonance [2]. Regarding the small HBM U9B with its thin threaded bolt that connects to the base adaptor, a similar behaviour might be expected. In addition, a third transducer (HBM U2B / 10 kN) of considerably rigid base coupling was chosen for comparison purposes. Earlier dynamic tests made with this transducer are described in [3].

The shock forces were generated without applying any pulse-shaping material at the impact surfaces, i.e. a hard metallic shock contact was achieved. For all mechanical couplings, the various threaded connections were fastened with defined torque.

To obtain supplementary data that might be beneficial for the parameter identification process, the mechanical impact configuration was modified by mounting additional load buttons with a spherical end face (see Fig. 2). This modification alters the transducer's dynamic response as the increased mass at the force introduction bolt reduces the transducer's fundamental resonant frequency, which basically is a function of the elastic coupling of the transducer's head mass.
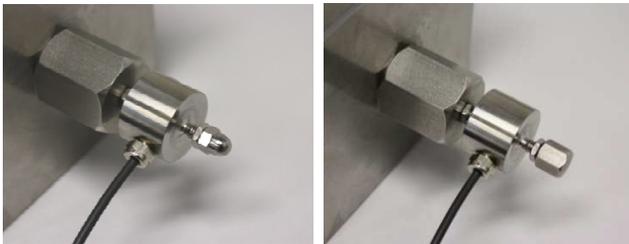

Figure 2: HBM U9B / 1 kN with mounted load buttons, increased head mass of 3.1 g (left) and 7.0 g (right)

2.2 Shock force measurements

Shock pulses generated by the impact of the 10 kg mass body have durations of about one millisecond in case of hard metallic contact. Typical examples of measured shock force signals are shown in Fig. 3. The plots demonstrate the variety of responses in the time domain. Different types of transducers with their specific mechanical adaptations respond quite differently.

The HBM U9B / 1 kN measured a very smooth pulse without post-impact signal ringing. In contrast, the much heavier Interface 1610 responded with shorter pulse duration, superposed oscillations and a prominent signal ringing. In the above-mentioned investigations with the larger Interface model, the lowest resonance was identified as the vibration of the transducer against its fixation on the reacting mass body [2, 5]. The same behaviour can now be expected here.

Typical measurement results obtained with different transducers and optional load buttons are exemplarily given

in Figs. 4 to 6. The plots show the three acquired signals, which are the accelerations a_1 and a_2 of both mass bodies derived by differentiation of the vibrometer signals, as well as the transducer output signal force F.
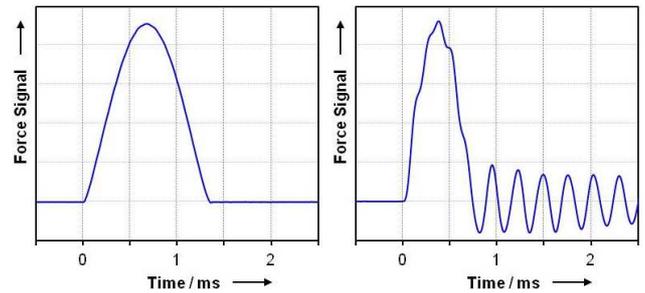

Figure 3: Measured shock force pulses (with 20 kHz low-pass): force transducer HBM U9B / 1 kN (left), Interface 1610 (right)

Figure 4 demonstrates that the acceleration signals can experience strong shock-excited noise. Using a low-pass filter of 20 kHz, which is below the expected fundamental resonance of the HBM U9B / 1 kN, these (parasitic?) signal components are still very strong and might have to be filtered for the parameter identification process.
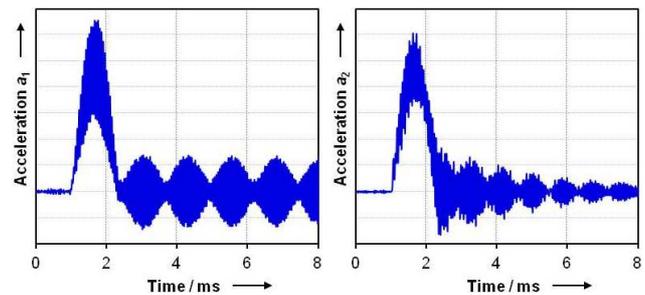

Figure 4: Shock-excited ringing of the acceleration signals: HBM U9B / 1 kN with load button 7.0 g, low-pass filtered at 20 kHz

The acceleration a_1 of the impacting mass body exhibits a nearly undamped vibration with a superposed beat signal. A similar behaviour shows the signal a_2 of the reacting mass body with the mounted transducer, but the damping is much stronger. In general, the repeatability of the high-frequent oscillation pattern is excellent. Repeated shocks of similar amplitude are nearly indistinguishable in the time domain.

The effect of an additional load button was tested with the small HBM U9B / 1 kN. Figure 5 illustrates the spectral content of the post-impact signal ringing for the various mechanical configurations. Some strong acceleration signal components above 20 kHz might result from modal oscillations of the mass bodies. Without additional head mass, the transducer shows an oscillation below 30 kHz that probably is its fundamental resonance. This peak apparently shifts towards lower frequencies with increasing mass of a load button. In addition, a strong resonance appears at about 9 kHz which might be explained by a bending mode.

The last example in Fig. 6 illustrates measurements in the time and frequency domain performed with the Interface 1610 / 2.2 kN. Both force F and acceleration a_2 signals show a strong vibration at 3.8 kHz, and the spectral analysis

reveals a weaker component at 7 kHz. With similar amplitudes of the dominant component in both signals, this behaviour differs from the previous experience [2], which may indicate to a different origin.
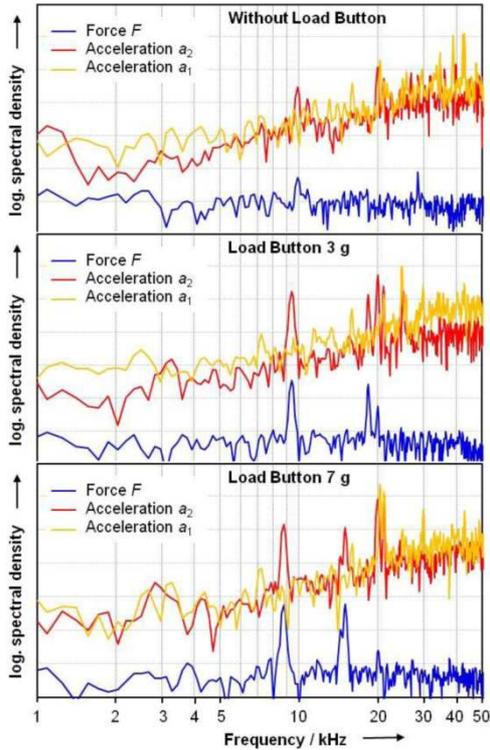

Figure 5: Spectral analysis of the signal ringing of HBM U9B / 1 kN with different load buttons
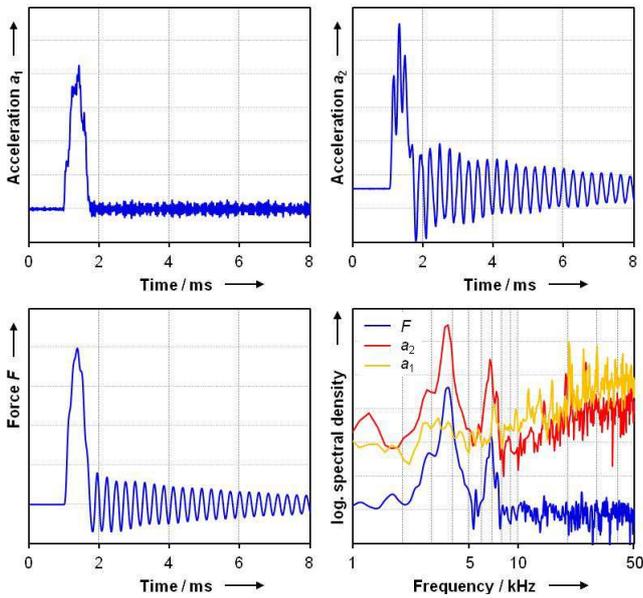

Figure 6: Shock signals in the time domain and frequency domain (signal ringing only) measured with Interface 1610 / 2.2 kN: accelerations a_1, a_2, force F, low-pass filtered at 20 kHz

Looking at the different shock responses of the various mechanical impact configurations, the identification of the transducer's dynamic parameters from shock measurements may require various analysis tools to provide satisfying results. To elaborate these mathematical methods and analysis procedures for the parameter determination from the experimental data, hundreds of shock force measurements were performed. The great number of tests will further provide statistical significance for the estimations of the associated measurement uncertainty.

3. MODELLING AND PROCEDURES FOR PARAMETER IDENTIFICATION

3.1 Model of the calibration device

The mechanical system of the calibration device with mounted force transducer is mathematically modelled by a one-dimensional multibody system. The model consists of a linear series arrangement of rigid masses coupled by visco-elastic springs. The force transducer is described by two masses (m_B, m_H) and a linear spring element of stiffness k and damping d. The dynamic behaviour of the model components is expressed by a system of linear differential equations. Some of the model parameters, such as the mass values of the two mass bodies (MB) and the adaptation parts, can be measured before the actual calibration. The remaining parameters, in particular those of the force transducer, have to be inferred from the dynamic calibration data or from CAD calculations.

Three models with 3, 4 and 5 model masses were investigated (see Fig. 7), which basically differ in their description of the mechanical adaptation of the transducer to the reacting mass body. The 3-mass model assumes a rigid fixation, whereas the models with 4 and 5 masses describe an elastic coupling of the force transducer.
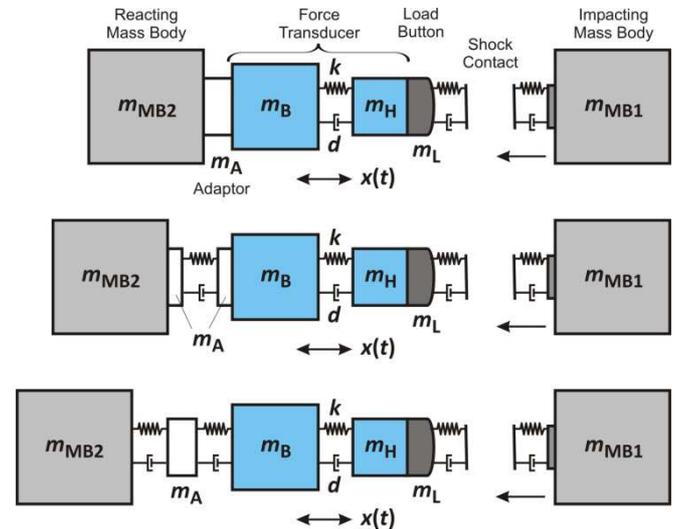

Figure 7: Models of the shock force calibration device: 3-mass model (top), 4-mass model (middle), 5-mass model (bottom)

3.2 Parameter identification

Assuming normally distributed measurement noise, we carry out a maximum-likelihood estimation of all parameters by means of non-linear least-squares. Therefore, a Runge-Kutta method for integration of the differential equations is employed together with a Levenberg-Marquardt method for

optimization [6]. The optimization merit function was chosen as

$$G(\boldsymbol{\theta}) = \|\boldsymbol{y} - S(\boldsymbol{x}(t_0), \boldsymbol{\theta}, \boldsymbol{\gamma}, F_1(t))\|^2 , \qquad (1)$$

where $\boldsymbol{\theta}$ is the vector of sought parameters, $\boldsymbol{y}$ the measured data, $\boldsymbol{x}(t_0)$ the ODE initial values, and $\boldsymbol{\gamma}$ the mass vector. Data $\boldsymbol{y}$ contains the transducer output signal $F(t)$ and the acceleration $a_2(t)$ of the reacting mass calculated by numerical differentiation of its measured displacement. Thus, the function $S(\cdot)$ denotes the numerical integration of the ODE corresponding to the considered system model and the calculation of the resulting system output data:

$$S(\boldsymbol{x}(t_0), \boldsymbol{\theta}, \boldsymbol{\gamma}, F_1(t); t_n) =$$
$$\left(\theta_1 \big(x_H(t_n) - x_B(t_n) \big), \ddot{x}_{MB2}(t_n) \right)^T , \qquad (2)$$

with ODE trajectories $x_H(t)$ and $x_B(t)$ denoting the displacements of the transducer's head mass and base mass, respectively, and $\ddot{x}_{MB2}(t)$ the acceleration of the reacting mass body. The trajectories are calculated from solving the ODE corresponding to the chosen model. The parameter θ_1 denotes the estimated stiffness of the force transducer.

The measured displacement of the impacting mass body was utilized to calculate the force input signal to the measurement system. In order to account for the system change during the measurement due to the non-constant contact of the considered masses, we employed a rectangular window to the measured displacement of the impact mass. The attenuation of noise in the calculation of acceleration from displacement was carried out by a 4th order Butterworth low-pass filter of 18 kHz.

The result of our parameter estimation for the Interface transducer is shown in Fig. 8. It shows the result of the two elements of function $S(\cdot)$ for the estimated parameters.
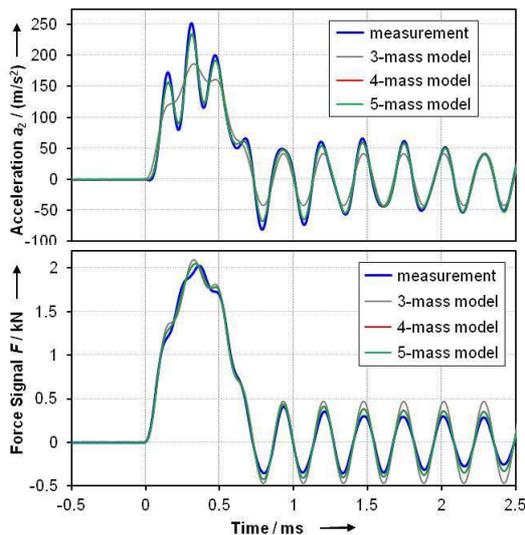

Figure 8: Comparison of modelled and measured shock force signals for Interface 1610 / 2.2 kN: acceleration a_2 of the reacting mass body (top), force signal F (bottom)

It can be seen that the 3-mass model is hardly capable of modelling the behaviour of the system. The 4-mass and 5-mass model, on the other hand, both show similar estimation quality. This demonstrates that for such experiments the coupling of the transducer to the reacting mass body cannot be assumed to be completely rigid. However, the result obtained for the 5-mass model still shows deviations from the observed signals. Moreover, this model does not include the additional peaks in the spectra shown in Figs. 5 and 6.

The analysis of various measurements with an identical force transducer under modified measurement conditions (e.g. pulse amplitudes, mechanical adaptations) will allow a verification of the suitability of the mechanical model and the data analysis methods applied. In particular, it will demonstrate the influence of various disturbances, e.g. from high-frequent modal oscillations [2] not explained by the chosen model, on the estimation of the model parameters of interest. The result of these investigations will then be incorporated in an extended system model.

4. CONCLUSIONS

This paper presented new research on the model-based dynamic calibration of force transducers by using shock excitations. By analyzing numerous experimental shock measurements with several force transducers of differing structural design, size, weight and mechanical coupling, the suitability of the mathematical model and the applied methods for the estimation of the transducer's model parameters were described and discussed.

Previous studies have proved modal oscillations of the measurement set-up of various origins that limit the usable bandwidth. Their effect on the modal-based calibration method with its data analysis procedures to be developed was further investigated and discussed in this paper.

ACKNOWLEDGEMENTS

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