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APPLICATION OF A SCANNING VIBROMETER FOR THE PERIODIC CALIBRATION OF FORCE TRANSDUCERS

Christian Schlegel, G. Kieckenap, and R. Kumme

Physikalisch-Technische Bundesanstalt, D-38116 Braunschweig, Bundesallee 100, Germany,
Christian.Schlegel@ptb.de

Abstract: This paper describes a procedure for the dynamic calibration of force transducers using sinusoidal excitations. Thereby, the main output of such a calibration is the dynamic sensitivity: the ratio between the electrical output signal of the transducer and the acting dynamic force. The force is traced back to the mass time's acceleration. With the aid of a scanning vibrometer, the acceleration distribution can be measured and a more realistic uncertainty for the acceleration can be given. In addition, parameter identification from the resonance behaviour can be applied.

Keywords: dynamic calibration, force transducer, acceleration measurement, parameter identification

1. INTRODUCTION

In the past, very precise procedures for static force measurements were developed and are now routinely used by calibration services and in national metrology institutes [1]. Thereby, relative measurement uncertainties down to $2\cdot10^{-5}$ are obtained using deadweight machines, which are the best standard to realize a traceable force.

Besides the precise realization of a force in a standard machine, there must be selected force transducers available which can be used as a transfer standard to give the primary calibration to the secondary calibration laboratories and industry. The crucial fact now is that often these statically calibrated force transducers are used in dynamic applications. That is the reason why more and more NMIs have established procedures for a dynamic calibration of force transducers and also other sensors.

Currently in the European Metrology Research Programme (EMRP) one promoted research topic is the "Traceable Dynamic Measurement of Mechanical Quantities", which includes, apart from a work package on dynamic force, also work packages on dynamic pressure, dynamic torque, the electrical characterization of measuring amplifiers and mathematical and statistical methods and modelling [2].

Similar to the static calibration philosophy, primary calibrations have to be provided which guarantee traceability to the SI base units and also transfer transducers (reference standards) to transfer these calibrations, e.g. to an industrial application. This transfer turned out to be the most complicated task because of the crucial influence of environmental conditions present in certain applications. Mostly, the transducers are clamped from both sides which leads to sensitivity losses due to the dynamics of these connections, which are more or less not infinitely stiff. On the other hand, the resonant frequency often shifts down to lower frequencies which can also drastically change the sensitivity. The problem can be solved to a certain extent by modelling the whole construction, including all relevant parameters. For that reason it is also important to determine the force transducer parameters, like stiffness and damping, which can also be obtained during a dynamic calibration. This article describes one possibility for a primary dynamic force calibration using sinusoidal excitations. The whole procedure as well as most of the setups were developed over two decades and are extensively described in [3-5]. Other methods as well as analysis procedures for dynamic force calibration are described in [5-7].

2. THE MEASURAND

The main output of a dynamic calibration is the frequency-dependent sensitivity, S_f. Thereby, the sensitivity is the ratio of the electrical output signal of the transducer, U_f, to the acting dynamic force. To initiate a force on the transducer, an additional mass is mounted on top of the transducer. This mass, m_t, in combination with the transducer's own head mass, m_i, contributes to the inertial force generated by the sinusoidal movement. The acceleration, $\ddot{x}_t$, is measured on top of the loading mass using a scanning vibrometer. Finally, the sensitivity is given by

$$S_f = \frac{U_f}{(m_t + m_i)\cdot \ddot{x}_t \cdot K_{corr}} \approx S_{f0}\cdot(1-p\omega^2) \quad (1)$$

The parameter, K_{corr}, takes into account the vertical acceleration gradient over the mass body. Finite element simulations have shown that the individual mass points of the mass body have slightly different accelerations in the vertical direction [3]. This correction factor can be neglected if quite small masses are used (only a few centimetres in height). From the theory it can be shown that the sensitivity can be approximated by a second-order polynomial. This is very useful to average and interpolate sensitivities measured, e.g. with different forces.

Thereby, the factor, S_{f0}, is the static sensitivity obtained for the limiting case ω=0, whereby p is a parameter which is determined by the ratio of the damping to the stiffness of the transducer. As one can see from the equation, the sensitivity is reduced quadratically with increasing frequency ω.

The periodic excitation of the force transducer can be used also to determine the parameter's resonance frequency, stiffness and damping factor of the transducer. This can be obtained by analyzing the resonance line obtained by the ratio of the acceleration measured on the top mass and the acceleration measured on the shaker table or the ratio of the force transducer signal to the acceleration on the shaker table. The resonance line can be fitted by the function of the mechanical resonator, whereby the amplitude, *A*, is given as a function of the frequency, *f*, by:

$$A(f) = \frac{a_0}{\sqrt{(f^2 - a_1^2) + 4a_2^2 f^2}} \quad (2)$$

The parameter a_1 is the resonance frequency f_0 according to:

$$f_0 = a_1 = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \quad (3)$$

From the resonance frequency the stiffness, *k*, of the force transducer can be obtained from equation 3. The parameter a_2 is connected with the damping factor according to:

$$b = 2\pi m a_2 \quad (4)$$

The mass, *m*, is the top mass of the force transducer including the internal head mass of the transducer. Sometimes it is useful also to know the bandwidth, Δf, of the peak which is given by the width of the resonance at $1/\sqrt{2}$ of the amplitude. This can be calculated by the difference of the two frequencies, *f1* and *f2* which correspond to this:

$$f_{1,2} = \sqrt{a_1^2 - 2a_2^2 \mp 2a_2^2\sqrt{a_1^2 - a_2^2}} \quad (5)$$

With the bandwidth, a quality factor, *Q*, can be calculated which is the ratio of the resonance frequency and the bandwidth:

$$Q = \frac{f_0}{|f_2 - f_1|} \quad (6)$$

It should be noted that the resonance peak can also be fitted by other peak functions like the Lorentz function. These functions characterize a peak often with the full width at half maximum, (FWHM). The FWHM can also be used to calculate the Q-Value, however in this case the square of the transfer function must be fitted. From the Q-Value also the damping factor can be calculated according to:

$$b = \frac{\sqrt{km}}{Q} \quad (7)$$

3. TECHNICAL DETAILS

The measurement arrangement can be seen in Figure 1. For the sinusoidal excitation three electromagnetic shaker systems can be used; a small one for forces up to 100 N and 10 Hz until 2 kHz, a medium one up to 800 N for 10 Hz -3 kHz, and a large shaker up to forces of 10 kN and frequencies of 10 Hz to 2 kHz. The shakers consist of two parts, the vibration exciter itself and a power amplifier. The sinusoidal signal is fed from an arbitrary signal generator directly into the power amplifier and modulates the current signal, which drives the coil of the shaker armature.
The beam of the laser vibrometer is guided via a 45° mirror down to the surface of the loading mass, mounted on top of the force transducer. The vibrometer is on a special frame, which is equipped with a passive and an active damping table. The active damping table is mounted directly below the vibrometer. Inside the table six accelerometers continuously measure the vibration. This measurement information is used in a closed loop control to steer 4 servo motors, which keep the position of the table always exactly in the horizontal position. Experience has shown that the table is able to suppress vibrational influences originating from the shaker very well. In addition, a first damping is performed via the passive damping table, which is placed directly on the platform of the mounting frame, see Figure 1.

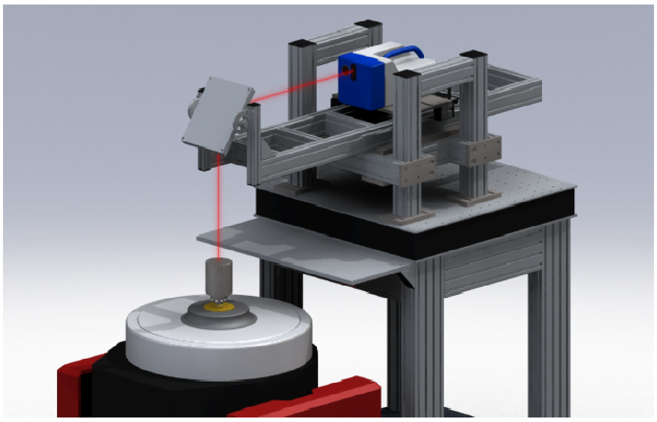

Figure 1. Calibration setup consisting of the electrodynamic shaker with the mounted force transducer and a scanning vibrometer for the acceleration measurement. The transducer is equipped with an additional loading mass.

The scanning vibrometer, PSV-400 M, is an instrument from Polytec. This device consists of the laser head, a controller (OFV-5000), a junction box and a PC system equipped with a 4 channel 5 MHz sampling card from National Instruments. The laser beam can be scanned in an angle region of ±20° in the horizontal and vertical directions. Thereby, the angle resolution is <0.002°. The controller includes several parts, two velocity encoders with analogue and digital outputs, an analogue displacement output and a digital decoder for the output of a down-converted IQ signal. The velocity decoder, referred to as VD-07, for the low frequency range up to 350 kHz, can measure velocities up to 500 mm/s and a second decoder, referred to as VD-09, for frequencies in the range from 10 Hz - 2.5 MHz, can measure velocities up to 10 m/s. In addition to the velocity proportional analogue output signals, the VD-07 decoder has a digital S/P-DIF output with a fixed sampling rate of 42 kHz with 24 bit resolution. The digital IQ converter, referred to as DD-600, provides the baseband Doppler signal in a bandwidth of 0-5 MHz with an IQ phase tolerance of ±2.5°.

4. CALIBRATION EXAMPLE

In the following example a piezoelectric force transducer, Type Kistler 9175 B, with a nominal force of 8 kN for tension and 30 kN for compression, was measured. The charge of this transducer was measured with a B&K Nexus charge amplifier. Figure 2 shows the dynamic sensitivity of the transducer measured with five top masses.

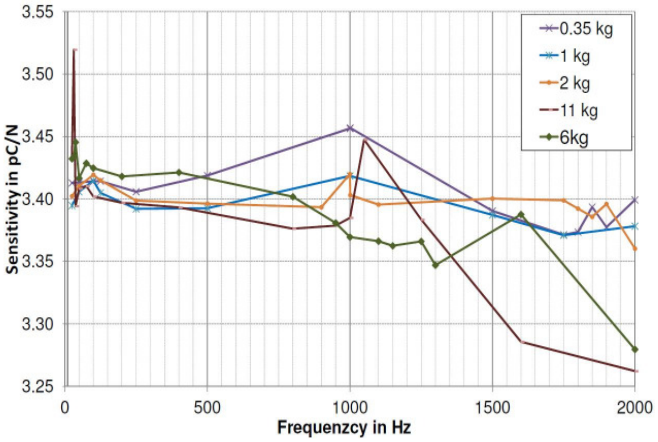

Figure 2. Dynamic sensitivity of a piezoelectric force transducer measured up to 2 kHz using 5 loading masses.

Compared with the strain gauge transducer, the correspondence between data sets of different loading masses is not so good. Especially beyond 1 kHz the curves drift apart quite widely. One reason for that is perhaps the special construction of this kind of force transducer. The transducer is a kind of ring which is clamped from both sides with special screws, so that a certain prestress is created. Both screws can be seen in a mathematical model also as additionally damped spring mass systems, as shown in section 1. The complex vibration between all these coupled spring systems can lead to a more sophisticated behaviour of the force transducer, as in the previously discussed case of the strain gauge transducer. Especially the bump in the sensitivity around 1 kHz might originate from a cross resonance, due to the special construction of the force transducer. The obtained sensitivities of the piezoelectric force transducer can be averaged according to equation 1. The result of this is shown in Figure 3. Thereby, the more straight solid line corresponds to the averaged sensitivity of the masses of 1kg and 2kg, which were measured without a special adapter. The other curve was obtained from the data measured with a 6kg and a 10kg top mass, which were mounted with a special adapter. The corresponding dashed lines are the error bands obtained from the error of the fit parameters. In this case, where measurements were done without and with a special mechanical adapter to mount the top mass, one can see the influence of such a coupling.

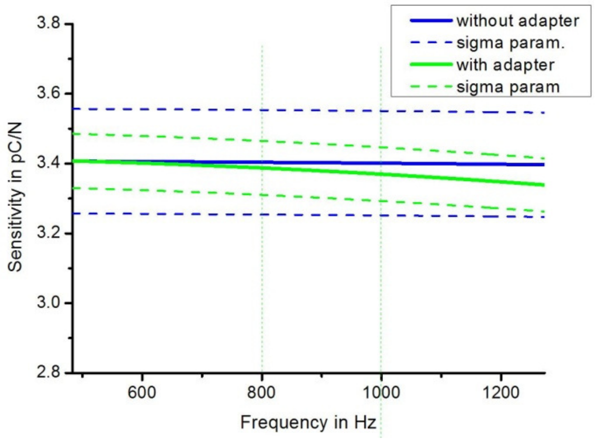

Figure 3. Averaged sensitivity of the data points displayed in Figure 2 according to equation 1.

In one case, where the top mass was screwed directly onto the force transducer, the coupling is very stiff and has no influence on the dynamics; in the other case, the adapter brings an additional softness to the mounting, which results in a sensitivity loss. The coupling can be modelled according to equation 1 with a damped spring. If a dynamic force acts now via the top mass on the transducer, energy also goes into the compression of the adapter spring, which will be missed for the compression of the transducer spring. This behaviour can generally be observed; more springs are mounted in between the source of force and the sensing transducer, as more sensitivity is lost.

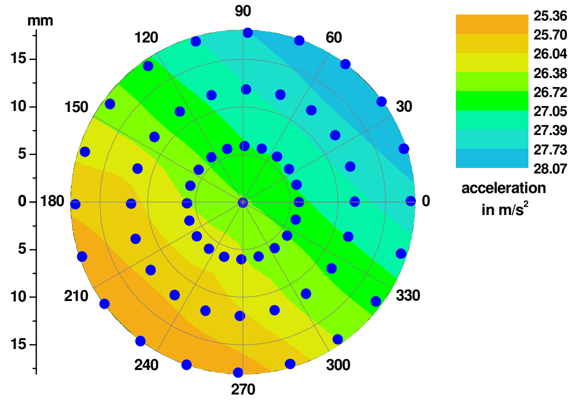

Figure 4. Acceleration distribution measured on the surface of the top mass. The filled circles are the measuring points, the shaded area in between was calculated through interpolation of neighbouring measuring points.

The big advantage in using a scanning vibrometer is the more detailed measurement of the acceleration. In Figure 4 the acceleration distribution measured on top of a loading mass is illustrated. As one can see, in this special case, there is no uniform distribution. Due to rocking modes of the top

mass which might originate from the kind of coupling to the force transducer, the acceleration can vary over the surface. Most of the calibrations which were performed, were normally measured in the middle of the top mass, which do not account for this effect. By averaging all the measured points of the acceleration distribution, one can take into account mechanical influences, like rocking modes in the calibration
result.

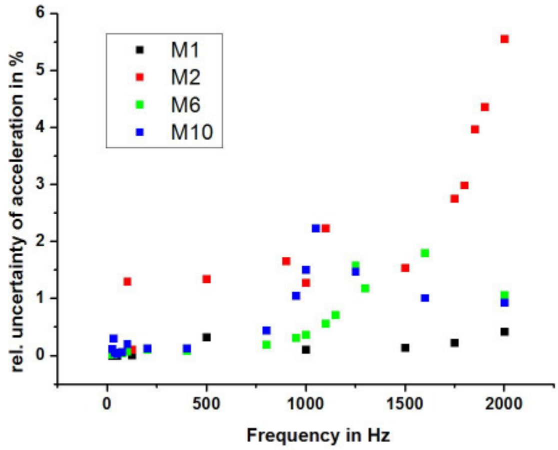

Figure 5. Relative uncertainty of the acceleration obtained from the averaging of the acceleration distribution measured on the surface of the top mass.

Furthermore, the standard deviation resulting from this can be used for a more realistic uncertainty evaluation of the acceleration measurement. To illustrate the actual situation, in Figure 5, the relative uncertainty of the acceleration is shown as a function of the frequency for four different top masses. Thereby, the data points were obtained by the averaging of 50-60 acceleration scan points. According to Figure 5, the acceleration uncertainty is mainly below 1%, apart from the data points obtained with the 2 kg loading mass. On the other hand, the influence of the resonance region leads to higher uncertainties and especially beyond a resonance one will be faced with higher measurement uncertainties.

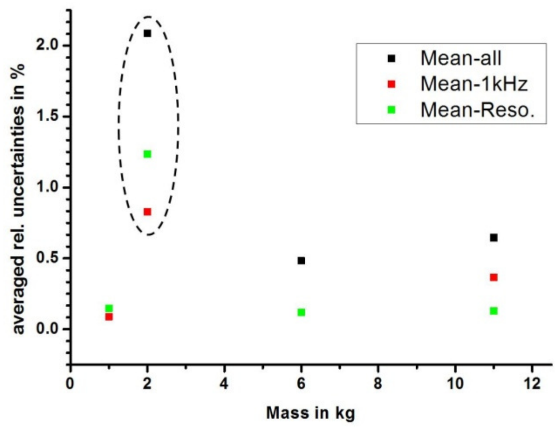

Figure 6. Averaged relative uncertainties for each top mass are shown for three kinds of data sample.

The individual uncertainties according to Figure 5 are averaged for each top mass in Figure 6. Thereby, different sets of data were chosen: first, all data of a certain top mass, second, all data up to a frequency of 1 kHz and, third, all data up to the beginning of the resonance. From this figure, one can conclude that the uncertainty of the acceleration is below 0.25% if the resonance region is avoided. On the other hand, one can see, e.g. on the data obtained with a 2 kg top mass (dashed ellipses), that special influences, which are initiated, e.g. by an imperfect mechanical adaption, can be rapidly detected. In this special case, it might be that the mounting screw thread was not perfectly aligned with the middle axis of the mass cylinder.

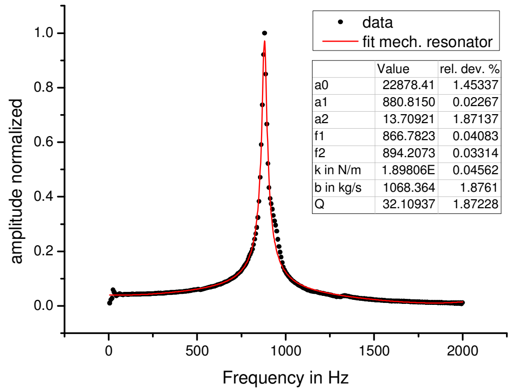

Figure 7. Example of a resonance peak measured with a chirp excitation of the force transducer, together with a fit function (solid line), which was chosen according to equation 2. The inset table shows the fit parameters and their relative deviations. The amplitude was normalized of the peak maximum.

As mentioned in section 1, the dynamic measurement can be used to determine the parameter stiffness and damping of the force transducer. According to equation 2, this parameter can be determined by measuring a resonance peak, as shown in Figure 7. There are several possible ways to obtain such resonance data. In the shown case, the transducer was excited via the shaker system with a chirped excitation. After that, the ratio between the signal of the acceleration measured on the top mass to the acceleration measured on the shaker table was calculated, which can be seen in Figure 7. The data can then be fitted with the function of the mechanical resonator according to equation 2. Each top mass results in a resonance by a different frequency. From the data pairs, resonance frequency and top mass, one can perform a fit according to equation 3, with the stiffness as a fit parameter. With the knowledge of the stiffness, one can calculate the damping factor from the band width of the peak of the mechanical resonator function. As shown in Figure 7, the resonance frequency and the stiffness connected with it, are well determined with quite low uncertainties. In contrast, the damping factor has a slightly higher uncertainty.

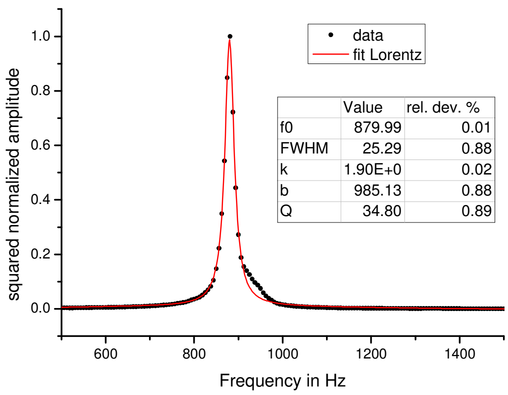

Figure 8. The square of the amplitude (ratio of the measured accelerations) fitted by a Lorentz-Peak. The table shows the fit parameter, resonance frequency, *f0*, and the full width at half maximum, FWHM=*w*, according to equation 8. The parameters, stiffness, *k*, in N/m, the damping, *b*, in kg/s and Q are calculated from the fit parameter.

As mentioned above the resonance can also be fitted with other peak fitting functions. A good candidate is thereby the Lorentz function:

$$y = y_0 + \frac{2A}{\pi} \frac{w}{4(x - xc)^2 + w^2} \tag{8}$$

From a fit of the Lorentz function one can obtain the offset, *y0*, the peak area, *A*, the FWHM, *w* and the resonance frequency, *xc*. The FWHM can be used to calculate the Q-Value and from this one can obtain the damping factor according to equation 7. A comparison between the fit using the model of the mechanical resonator and the Lorentz peak shows only small differences in the resonance frequency and an approximately 6% deviation in the Q-Value.

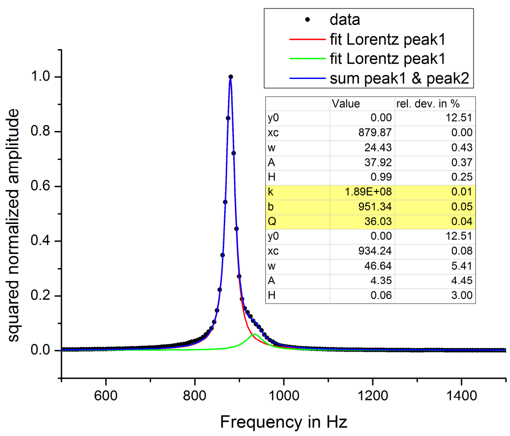

Figure 9. The resonance peak fitted by the combination of two Lorentz peaks. The inset table shows the fit parameters of both peaks according to equation 8. The first parameters correspond to the main resonance.

As seen in Figure 8 the resonance peak has a small tail on the right-hand side. This tail is an indication of a second small underlying peak. To unfold this peak from the main resonance a second Lorentz peak was included in the fit procedure. The result shows a very good agreement between the data points and the combined fit curve. According to the table with the parameters of the fit inside Figure 8, one can see that the width of this second small peak is approximately twice as wide as the main resonance. The origin of this disturbance might come from the mechanical coupler which connects the top mass to the transducer.

Another possibility to obtain the resonance peak is to analyze the ratio of the force signal to the acceleration measured on the shaker table. In this case the frequency response of the conditioning amplifier has to be unfolded first from the data. It should be noted that for the determination of the damping factor, the described procedure might not be the best choice. Other methods using a step or impulse response might be more suited.

5. CONCLUSION

The dynamic calibration of force transducers using sinusoidal excitations initiated by electromagnetic shaker systems was shown. A main part of such a calibration is the acceleration measurement. By using a scanning vibrometer, it is possible to measure the whole acceleration distribution on the surface of the top mass. From this distribution a more precise uncertainty of the acceleration can be given than in the case of a point measurement. In addition, mechanical influences like alignment problems or rocking modes can be seen, e.g. from an asymmetrical distribution. The resonance behaviour of the force transducer can be used to make a parameter identification of the force transducer to obtain the resonance frequency, the stiffness and the damping value. From the physical point of view the model of the mechanical resonator is the best choice. On the other hand also other peak fitting functions like the Lorentz function give reasonable results for the parameters. Thereby only very small differences in the resonance frequency and the stiffness are obtained, the deviation for the damping factor is only at a low percent level.

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6. REFERENCES

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