IMEKO 23rd TC3, 13th TC5 and 4th TC22 International Conference
30 May to 1 June 2017, Helsinki, Finland

MECHANICAL INFLUENCES OF THE SHAKER ARMATURE ON PERIODIC FORCE MEASUREMENT

Saher R. Hassan, Ch. Schlegel, and R. Kumme

Physikalisch-Technische Bundesanstalt, Bundesallee 100, 38116 Braunschweig, Germany,
saher.r.hasan.ext@ptb.de

Abstract: This paper investigates the mechanical influences of the shaker armature's mechanical structure on periodic force calibration, and hence on measurement uncertainty. A finite element model (FEM) was developed in order to perform modal and harmonic analyses of the measurement setup. The verification and validation of the FE model were continuously conducted through a measurement comparison, whereby a scanning vibrometer was used. The results show that the dynamic behaviour of the shaker armature should be taken into consideration before carrying out a periodic force measurement.

Keywords: dynamic force, periodic force calibration.

1. INTRODUCTION

The measurement of dynamic force is widely used in many industrial applications, such as material testing, production processes, and vehicle dynamics. Therefore, a number of national metrology institutes have developed periodic force calibration facilities [1]–[3]. The dynamic behaviour of a periodic calibration is affected by three main factors: the internal structure of the force transducer; the mechanical coupling between the transducer and the shaker armature on one side and between the transducer and the top mass on the other side; and the dynamic behaviour of the shaker armature. The former factor introduces a periodic input excitation force to the base of the force transducer. This force is directly connected to the shaker armature's dynamic parameters. There are three types of effects, which have influences during calibration. The main goal of the calibration process is to estimate the dynamic parameters of the force transducers, such as stiffness and damping, in addition to their dynamic sensitivity. The stiffness k_f and damping b_f of the force transducer can be estimated according to equation 1 [4], assuming a system with a single degree of-freedom.

$$\frac{\ddot{y}_f}{\ddot{y}_s} = \sqrt{\frac{1+\left(\frac{b_f}{k_f}\cdot\omega\right)^2}{\left(1-\frac{m_t}{k_f}\cdot\omega^2\right)^2+\left(\frac{b_f}{k_f}\cdot\omega\right)^2}} \qquad (1)$$

These parameters can be estimated only if the acceleration is measured at two points; on the top of the load mass and at the base of the force transducer. One of the main limitations of this approach is that the measurement of the acceleration at the base of the force transducer is actually performed on the shaker armature and not directly under the force transducer. This actual acceleration measurement is affected by the acceleration distribution over the shaker armature.

So far, research has tended to focus either on the modal analysis of the measurement setup with the assumption of a lumped mass shaker armature [5] or on the force transducer only without the shaker and the load mass [6]. To investigate the dynamic behaviour of the complete mechanical structure of a periodic force measurement setup, an FE model has been developed and verified to carry out the modal and harmonic analyses. The model takes the complete mechanical structure of the setup into consideration, which includes the force transducer, the load mass, and the shaker armature as in a real application.

2. METHODOLOGY

A photograph of the mechanical portion of the experimental arrangement is provided on the left-hand side of Figure 1, while a 3-D model of the mechanical structure is shown on the right of the same figure. More details about the measurement setup can be found in [7]. A 100 mm high hollow aluminium bar was used as a force transducer. The bar's outer diameter is 26 mm with a wall thickness of 2 mm. The bar was mounted onto the shaker armature using a mechanical thread adapter. An 80 mm diameter and 97 mm high brass cylinder with a mass value of about 4 kg was used as a load mass. This load mass has a threaded hole in the centre at one end, which is attached to the bar using a

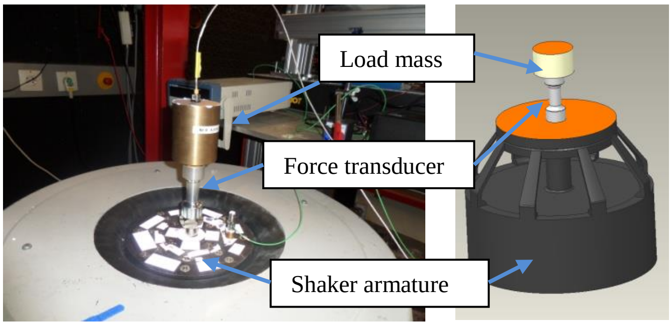


Figure 1: Photograph of a periodic measurement setup (left) and the 3D model (right).

mechanical thread adapter. The vertical acceleration was measured at 69 points; these points are distributed over the entire top surface of the load mass as shown in Figure 2. Pseudorandom excitation was applied to the shaker with a predetermined frequency range and amplitude in order to obtain the frequency response of the whole mechanical structure. As the acceleration distribution is only measured on top of the load mass through measuring the vertical acceleration, only the modes that have longitudinal displacement/acceleration components can be detected.

		13	22	31	40	49		
	6	14	23	32	41	50	58	
1	7	15	24	33	42	51	59	65
2	8	16	25	34	43	52	60	66
3	9	17	26	35	44	53	61	67
4	10	18	27	36	45	54	62	68
5	11	19	28	37	46	55	63	69
	12	20	29	38	47	56	64	
		21	30	39	48	57		

Figure 2: Distribution of the scan points on the load mass. 69 points are uniformly distributed over the load mass surface. The scanning vibrometer measures the vertical acceleration of each point.

An FE model has been developed to perform the modal and harmonic analyses of the complete mechanical structure of the periodic force measurement setup. The mechanical structure includes the force transducer, the load mass, and the shaker armature. The mechanical properties of the three parts were taken from tabulated data. All threaded connections were assumed to be rigid. The model assumes the absence of damping; therefore, it is predicted that a steep frequency response curve and relatively high resonance amplitudes will occur. To perform modal analyses of all possible vibration modes, fixed boundary conditions were provided at the base of the shaker armature, without any other boundary conditions on the rest of the structure. The harmonic analysis was performed by adding harmonic excitation signals to the base of the shaker armature. The frequency response of the averaged acceleration on the top surface of the load mass was visualized in order to simulate the real measurement setup as realistically as possible, and obtain comparable results. A comparison between the measured and simulated longitudinal resonances of the shaker armature has been used to verify the FE model proposed. ANSYS software was used to perform the FE calculations.

3. RESULTS AND DISCUSSION

The results are introduced in this section in the following order: verification of the FE model, modal analysis, and harmonic analysis.

Verification of FE model: Figure 3 (top) represents the longitudinal mode of the shaker armature using the FE model. The simulated resonance frequency of the shaker armature is 2481.3 Hz and it deviates about 1.3% from the measured resonance frequency. Figure 3 (bottom) shows the measured frequency response using pseudorandom excitation in the range of 0-10 kHz. As can be seen in Figure 3(bottom), the measured resonance frequency is 2450 Hz.

Hence, a deviation of only about 1.3% between the measured and simulated results has been achieved which represents a good agreement between the FE model and the experimental setup.

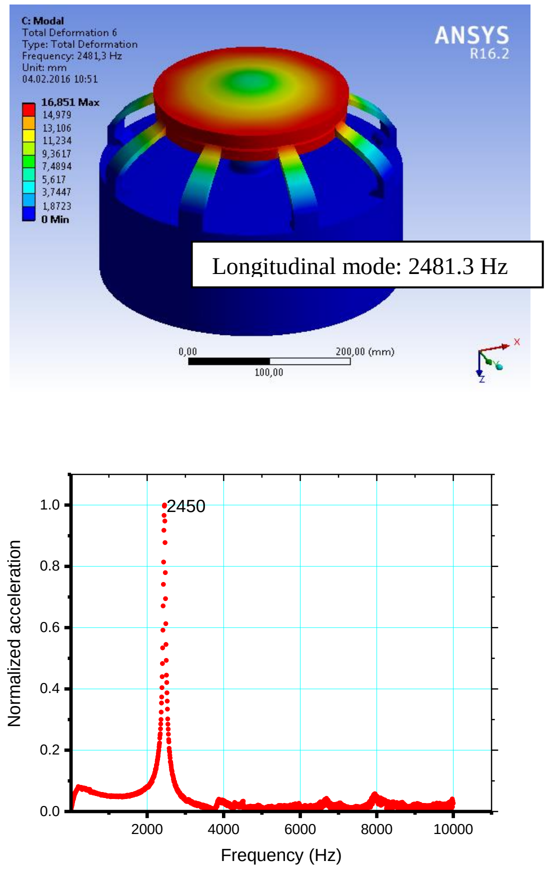
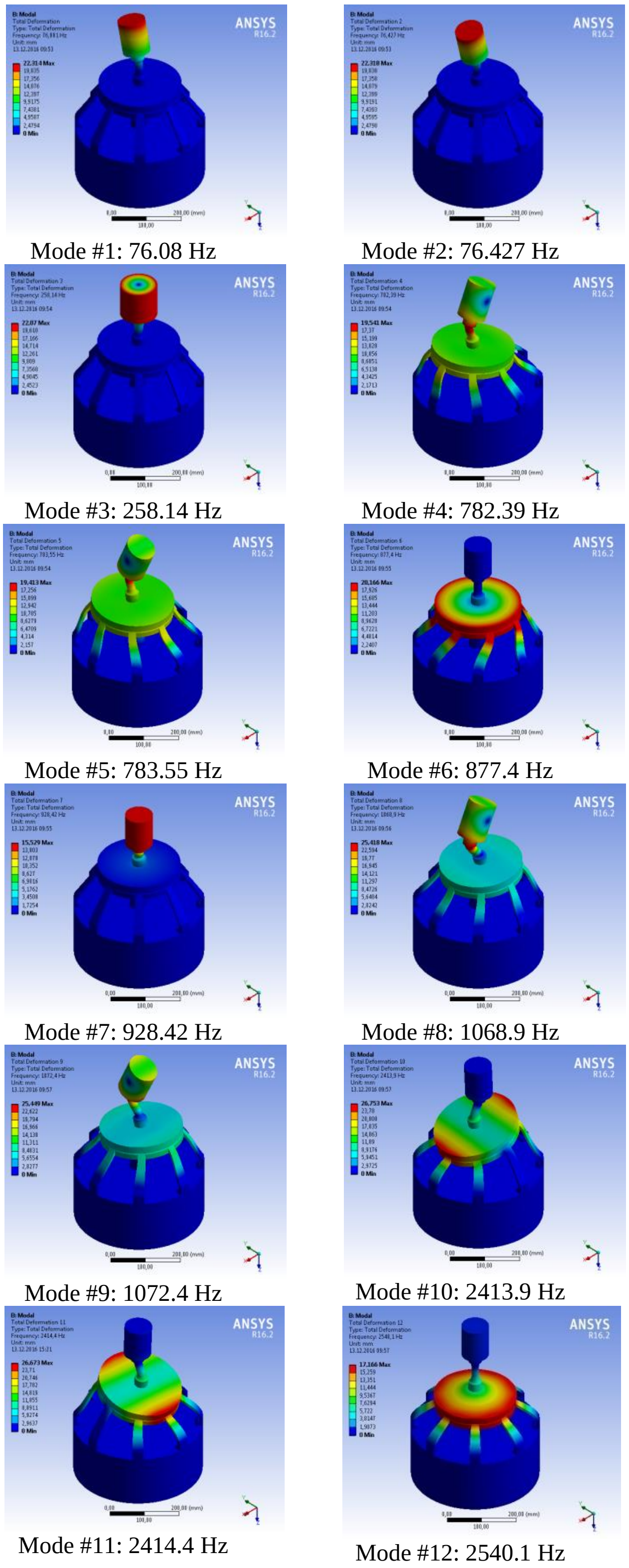


Figure 3: The FE modal analysis of the shaker armature with the longitudinal resonance (top) and the measured frequency response (bottom).

Modal analysis of the measurement setup: Figure 4 shows the spatial displacement for different dynamic modes of the simulated setup in the range of 0-2600 Hz. These modes can be categorized into tilting, torsion, and longitudinal modes.

In titling modes, one element of the structure (e.g. load mass) tilts relative to the other two elements. For the first two modes, the load mass tilts relative to the force transducer. The 4th, 5th, 8th, and 9th modes represent the tilting of both the load mass and the force transducer relative to each other. The tilting of the shaker armature is represented in the 10th and 11th modes.

In torsion modes, there is a difference of the spatial displacement over the radial direction of either the load mass (as shown in the 3rd mode) or the shaker armature (as shown in the 6th mode).



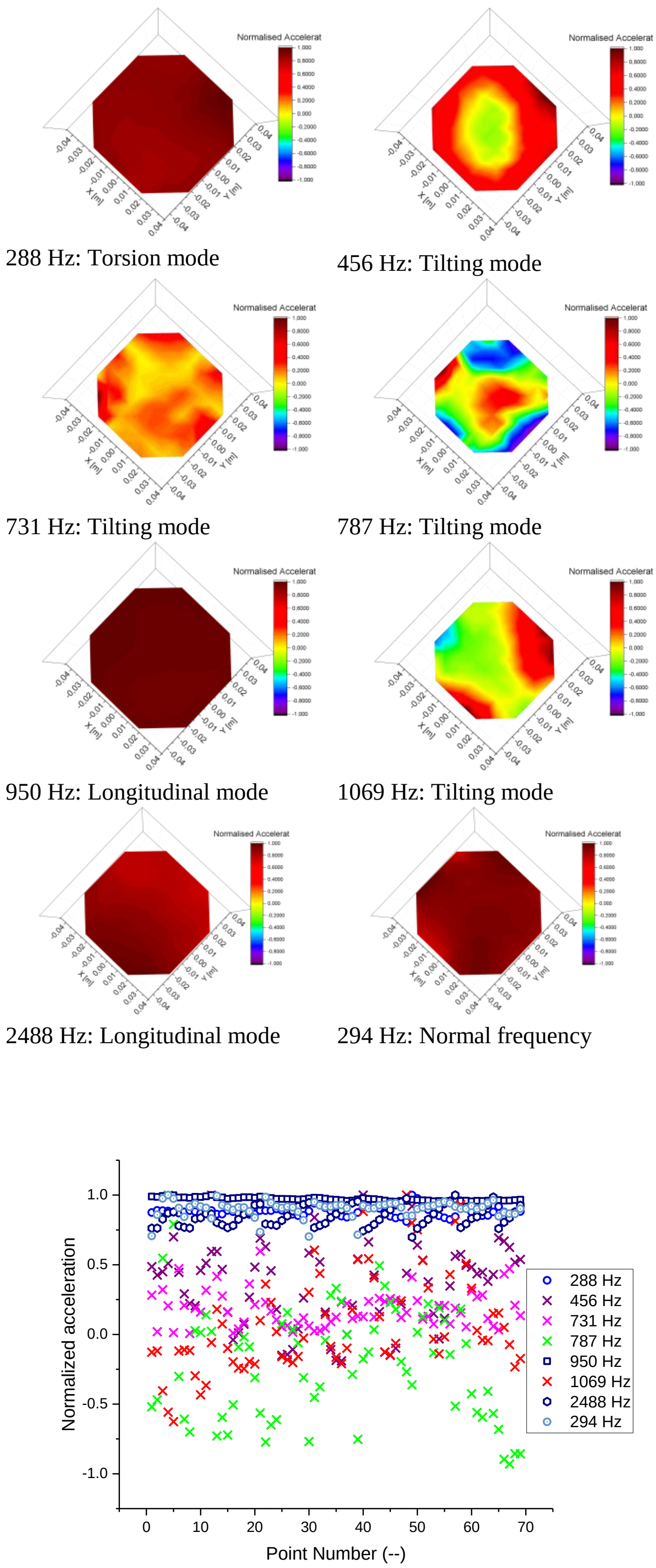


Figure 4: FE modal analysis of the complete measurement setup including force transducer and load mass mounted on the shaker armature. The analysis contains 12 modes in a frequency range of 0-2600 Hz. The colour code gives the displacement values whereby red indicates high displacement and blue indicates zero displacement. Note the exaggeration of the displacement values.

Figure 5: Normalized measured vertical acceleration distribution on the load mass of different frequencies (top) showing high variations for tilting modes (bottom). Crosses represent tilting modes while circles and squares represent the other modes.

Vertical displacement/acceleration components are assumed to be close to zero in the torsion modes.

Two longitudinal modes (main system resonances) have been noted; the first resonance, which corresponds to the force transducer, appears in the 7th mode, and the second one, which corresponds to the shaker armature, appears in the 12th mode.

Figure 5 shows the vertical acceleration distribution over the load mass at certain frequencies (written below each figure). These frequencies were selected where certain modes can obviously be observed. There is a significant dispersion of the vertical acceleration distribution of the tilting modes rather than longitudinal modes and normal operating frequencies; thereby, a good agreement has been achieved between the measured and the simulated results.

Uncertainty analysis: It is obviously predicted that the calibration of force transducers within the tilting modes will lead to higher contributions of type A uncertainty, and hence the expanded uncertainty will significantly increase. It is well known that the main uncertainty contribution in the periodic force calibration of force transducers comes from the acceleration measurement of the load mass [8]. Figure 6 shows the relative standard uncertainty of the measured acceleration on the top surface of the load mass at the same selected frequencies as in the last section.

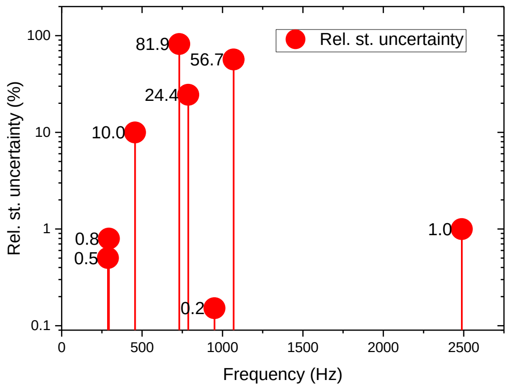


Figure 6: The relative standard uncertainty of the measured acceleration on the top of the load mass. The uncertainty was obtained by averaging 69 acceleration measuring points. In addition, note the log scale y-axis.

It should be noted that the uncertainty given above does not include the contribution influenced by the measuring equipment.

Harmonic analysis: Figure 7 shows the measured and simulated frequency resonance of the complete structure. The results show a good agreement between the simulated and measured resonance frequencies of the system with a deviation of about 2.3%.

Figure 7 (bottom) reveals that there has been a sharp drop of the FE model amplitudes far from the resonances caused by the assumption of an absence of damping.

Compared with the second resonance, the first resonance, which represents the fundamental frequency of the force transducer, exerts higher amplitudes stemming from the measurement/visualization of the frequency response over the load mass. It is almost certain that the second resonance, which represents the shaker's resonance, amplitude would be higher than that of the first resonance if the frequency response were measured/visualized on the top surface of the shaker armature.

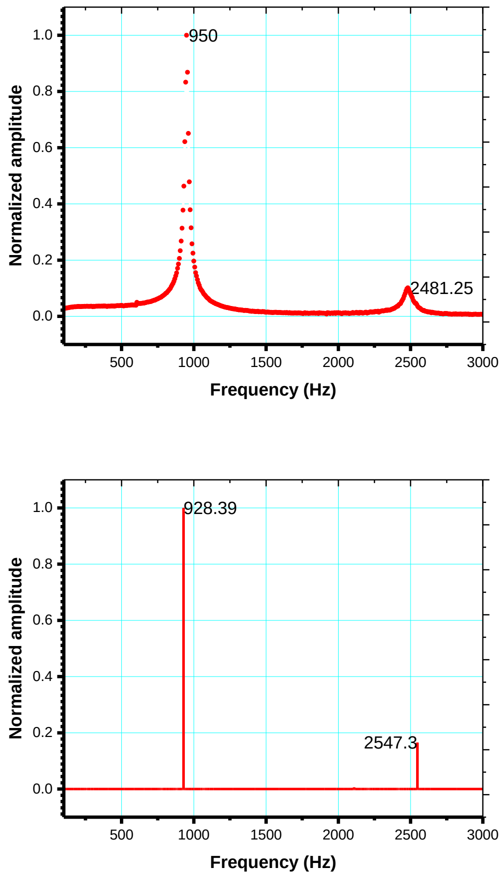


Figure 7: Harmonic analysis in the range of 0-3000 Hz of the measurement setup using FE model due to harmonic excitation in the longitudinal direction (bottom) and the measured acceleration resonance over the load mass using pseudorandom excitation (top).

4. CONCLUSION

In this paper, the mechanical influences of the shaker armature on the periodic force measurement have been thoroughly investigated. Because the investigations have been performed through the establishment of a measurement setup and a new FE model proposal, the following conclusions can be drawn:

- A new numerical model of the complete mechanical structure was developed.

- A good agreement between the simulated and measured results has been achieved.
- The reasons behind the small deviation of the simulated results may be due to using tabulated rather than measured material properties, the assumptions of boundary conditions, or the assumption of rigid connections.
- Based on the given results, the tilting modes of the complete mechanical structure introduce a significant impact on the acceleration distribution over the load mass, and hence on the expanded uncertainty.
- The accuracy of the FE model can be enhanced through the inclusion of the damping properties.

ACKNOWLEDGEMENT

The authors gratefully acknowledge support by the Braunschweig International Graduate School of Metrology B-IGSM. The authors would also like to thank their colleague Holger Kahmann from PTB for his support in FEM using ANSYS software.

5. REFERENCES

[1] C. Schlegel, G. Kieckenap, B. Glöckner, A. Buß, and R. Kumme, “Traceable periodic force calibration,” *Metrologia*, vol. 49, no. 3, pp. 224–235, Jun. 2012.

[2] N. Medina and J. de Vicente, “Force sensor characterization under sinusoidal excitations,” *Sensors (Switzerland)*, vol. 14, no. 10, pp. 18454–18473, 2014.

[3] N. Vlajic and A. Chijioke, “Traceable dynamic calibration of force transducers by primary means,” *Metrologia*, vol. 53, no. 4, pp. S136–S148, 2016.

[4] R. Kumme, “Investigation of a primary method for a dynamic calibration of force measuring instruments: a contribution to reduce the measuring uncertainty,” Doctoral Thesis, in German, PTB, 1996.

[5] C. Schlegel, G. Kiekenap, H. Kahmann, and R. Kumme, “Mechanical influences in sinusoidal force measurement,” *ACTA IMEKO*, vol. 4, no. 2, pp. 57–61, 2015.

[6] M. Kobusch and S. Eichstädt, “Challenges of the model-based dynamic calibration of force transducers – A case study,” in *Proc. 21st IMEKO World Congress*, 2015.

[7] C. Schlegel, G. Kieckenap, and R. Kumme, “Application of a scanning vibrometer for the periodic calibration of force transducers,” pp. 3–6, 2012.

[8] C. Schlegel, G. Kiekenap, and R. Kumme, “Uncertainty evaluation of sinusoidal force measurement,” in *Proc. 21st IMEKO World Congress*, 2015.