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EXPERIMENTAL METHOD FOR THE NON-CONTACT MEASUREMENT OF ROTATIONAL DAMPING

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Abstract: For a model-based description of the dynamic behaviour of a dynamic torque calibration device, it is necessary to determine the model parameters. This paper describes a method for the determination of damping properties with rotational excitation. To minimise the influence of the measurement, the oscillations were investigated by non-contact measurement.

Keywords: damping measurement, rotational vibration, rotational damping.

1. INTRODUCTION

A dynamic torque calibration device is described by a physical model [1]. This model assumes linear and time invariant (LTI) behaviour for all components. In order to be able to identify the unknown model parameters of a mounted transducer for future calibrations, it is necessary to identify the model parameters of the calibration device first. To this end, auxiliary measurement set-ups were developed. The experimental method for the determination of the damping will be presented in this paper.

The calibration device is realised as a drive shaft set-up for rotational oscillations. The damping properties should, therefore, be determined for torsional vibrations. A calculation of the damping properties is not feasible due to the complex shape of the devices under test (DUTs) and the unknown material parameters, which – if available – would have unknown measurement uncertainties.

2. DAMPING MEASUREMENT

The linear model of the measuring device assumes a viscous damping behaviour for all considerations. For the components to be investigated, low damping values are expected. Two different methods of measurement are commonly used in this case [2]:

1) Half intensity width (see Figure 1a): From the width of the resonance peak at the level $\hat{q}_{\max}/\sqrt{2}$ of a forced oscillation transfer function $\hat{q}$ in the frequency domain, the damping ratio ζ can be calculated from

$$\zeta = \frac{f_2 - f_1}{2\, f_0} \tag{1}$$

with the natural frequency f_0 and the frequencies f_1, f_2 at half intensity of the resonance peak.

2) Decay analysis (see Fig. 1b): Analysing the decay of an oscillation $q(t)$ after a (step or pulse) excitation in the time domain leads to the decay rate δ through the equation

$$q(t) = q_0 \cdot \mathrm{e}^{-\delta t} \cdot \sin(\omega_\mathrm{d}\, t + \varphi)\ , \tag{2}$$

with the magnitude q_0, the damped angular natural frequency ω_d and the phase φ. The corresponding damping ratio is given with the natural frequency ω_0 by

$$\delta = \zeta \cdot \omega_0\ . \tag{3}$$

An alternative analysis in the time domain is the logarithmic decrement Λ which describes the decay from an oscillation k to the next n oscillations by

$$\Lambda = \frac{1}{n}\ln\left(\frac{q_k}{q_{k+n}}\right). \tag{4}$$

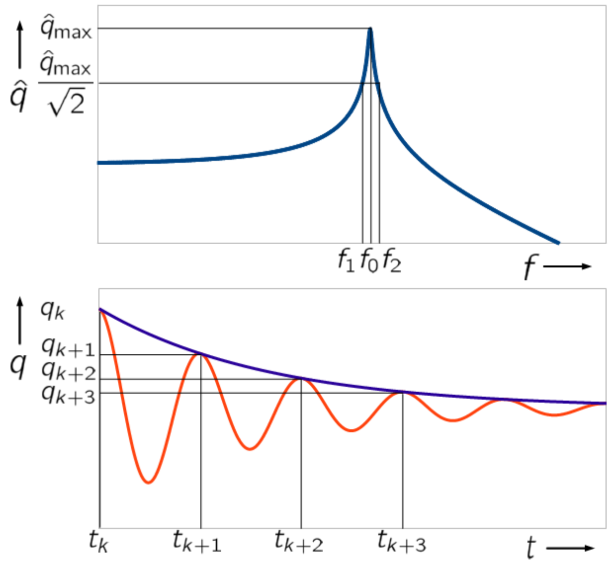

Figure 1a (top): Resonance width damping measurement.
Figure 1b (bottom): Decay analysis.

For underdamped systems with $\zeta \ll 1$, the relation of Λ and ζ is given by

$$\zeta = \frac{\Lambda}{2\,\pi}\;. \tag{5}$$

As mentioned before, the damping d is one parameter of the linear model. The product of this parameter d and of the time dependent angular velocity $\dot{\varphi}(t)$ equals the torque $M(t)$ as

$$M(t) = -\,d \cdot \dot{\varphi}(t)\quad . \tag{6}$$

It is possible to calculate the damping coefficient based on the results for the damping ratio (see Eq. (3)) with the torsional stiffness c or the mass moment of inertia J:

$$d = \frac{2 \cdot \zeta \cdot c}{\omega_0} = 2 \cdot \zeta \cdot J \tag{7}$$

For the set-up described, the decay analysis was chosen to determine the damping parameter. Advantages of this method are the easier generation of the step excitation of the oscillation and the only little modifications of the DUT for measurement.

3. MEASURING SET-UP

The main focus for the development of the measuring set-up (see Fig. 2) was to analyse the damping properties of the coupling elements, which are assumed to have the lowest torsional stiffness of all of the components of the dynamic torque calibration device, and therefore, dominantly influence the dynamic behaviour. These coupling elements are designed to be connected to rod ends with collet chucks. The design is rotationally symmetrical to minimise imbalance. For the damping measurement, the oscillating components of the DUT should neither need to be modified, nor should additional components need to be mounted for the excitation or the measurement to minimise unwanted effects due to any modification.

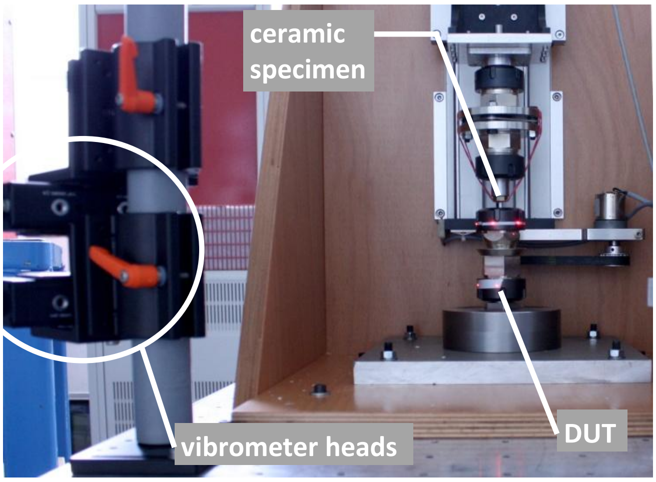

Figure 2: Measuring set-up with specimen and DUT.

To meet these demands, a non-contact measurement of the oscillations is carried out by means of rotational laser Doppler vibrometers, which have the additional advantage of being robust regarding parasitic translational in-plane and out-of-plane oscillations. To prevent measuring the damping of the base plate of the measuring set-up, two rotational vibrometers measure the oscillations at the top and at the bottom of the device under test (see Fig. 3).

The oscillations to be analysed for the damping measurement are generated by a negative torque step applied to the DUT. The torque step is generated by a specimen, which is supposed to break under a certain torque load. The specimens are made of technical ceramics, which exhibits brittle fracture generating the negative torque step.

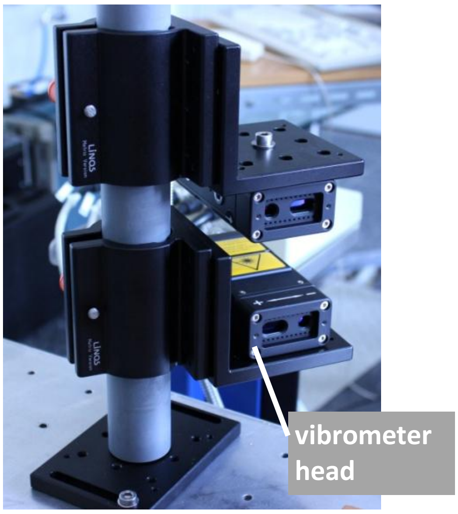

Figure 3: Set-up of the two rotational vibrometer heads.

The torque load on the specimen and the device under test is generated by means of an indicating torque wrench and transferred by a spanner socket to a hexagonal tool attached to the top of the specimen (see Fig. 4 at the top). The torque generation mechanism is supported by bearings to avoid axial loading and bending moments. After the breaking of the brittle specimen, a lifting tool pulls the components back from the DUT to minimise mechanical interference.

The device under test is mounted on a heavy cylindrical steel structure to minimise oscillations of other components than the DUT. The whole measuring set-up is mounted on a solid concrete foundation for the same reason.

4. MEASUREMENTS

For the measurements, specimens of three different materials were tested to investigate their suitability for the negative step excitation. Specimens made from the machinable ceramic Macor manufactured by Corning Inc., zirconium dioxide (ZrO_2) and aluminium oxide (Al_2O_3) were tested. The different specimens are depicted in Figure 4. The advantage of Macor is the flexibility in machining the specimen; however, it is expensive compared to the other materials. Both zirconium dioxide and aluminium oxide have a higher breaking torque to diameter ratio, which is about twice as high as that of Macor. It was, therefore, possible to use specimens with smaller diameters.

All three materials turned out to be suitable for the measurements. No difference in the brittleness of the fractures was discovered.

The data acquisition was carried out by means of a high-speed transient recorder. The data from the two channels of the rotational vibrometers were acquired simultaneously with a sampling rate of 200 kSamples/s.

To avoid influences due to the varying mounting torque, the collet chucks were fastened to a torque level similar to that of the application in the dynamic torque calibration device by means of a torque wrench.

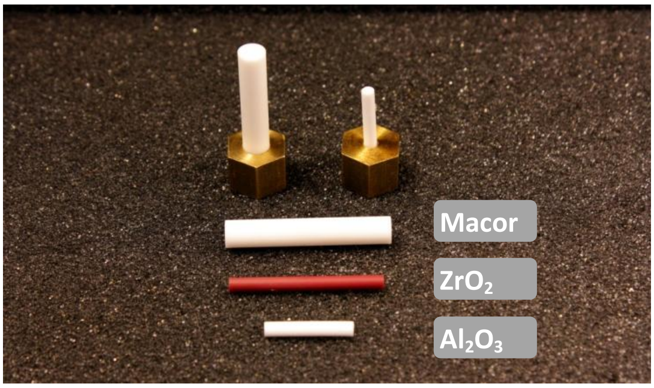

Figure 4: Different specimens with hexagon tool for the torque drive (top) made of Macor, aluminium oxide (Al_2O_3) and zirconium dioxide (ZrO_2).

5. DATA ANALYSIS

For the determination of the torsional damping, the decay of the oscillations generated by the negative step function is analysed. The acquired data consists of the measured angular velocities $\omega_1(t)$ at the top and $\omega_2(t)$ at the bottom. For the data analysis, these angular velocities were cumulatively numerically integrated using the trapezoidal rule. The derived angle values $\varphi_1(t), \varphi_2(t)$ give the torsion angles of the DUT φ_{DUT} following

$$\varphi_{\mathrm{DUT}}(t) = \varphi_1(t) - \varphi_2(t)\,. \qquad (8)$$

The derivative of the time dependent torsion angle of the DUT then gives its angular velocity, which is the basis for the damping analysis. It is calculated numerically as the vector of differences and low-pass filtered afterwards.

The determination of the damping is based on the analysis of the decay of the generated monofrequent sinusoidal oscillations, as described in Eq. (2). For this purpose, a nonlinear least squares approximation is calculated. As outcome of this approximation, the parameters decay rate δ and natural frequency ω_0 are needed for the determination of the damping ratio ζ as given in Eq. (3). Due to the damping, the approximated angular frequency ω_{d} differs slightly from the natural frequency ω_0 for the undamped case, the relation is given by

$$\omega_0 = \sqrt{\omega_{\mathrm{d}}^2 + \delta^2}\,. \qquad (9)$$

The damping coefficient, which is the value needed for the model parameter identification of the dynamically calibrated torque transducer under test, can be calculated from Eq. (7) using additional parameters which have been determined as well [1].

6. MEASUREMENT RESULTS

First measurements are promising (see Fig. 5). The oscillations generate a sufficiently high output signal of the rotational vibrometers, and the decay behaviour is as expected. However, the generated signal is not a pure monofrequent sine even after low-pass filtering (see Fig. 6). The determination of the damping ratio ζ by using Eqs. (2) and (3) requires a monofrequent signal. The frequency selected for the analysis algorithm was always the dominant frequency in the discrete Fourier transform.

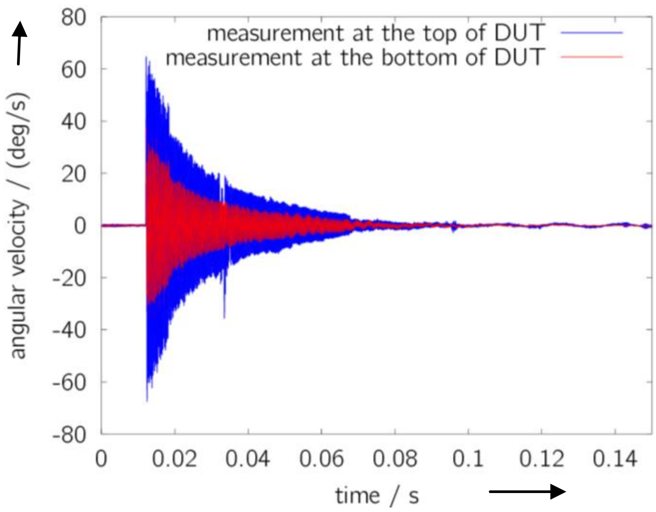

Figure 5: Vibrometer output at the top and at the bottom of the DUT from a typical measurement.

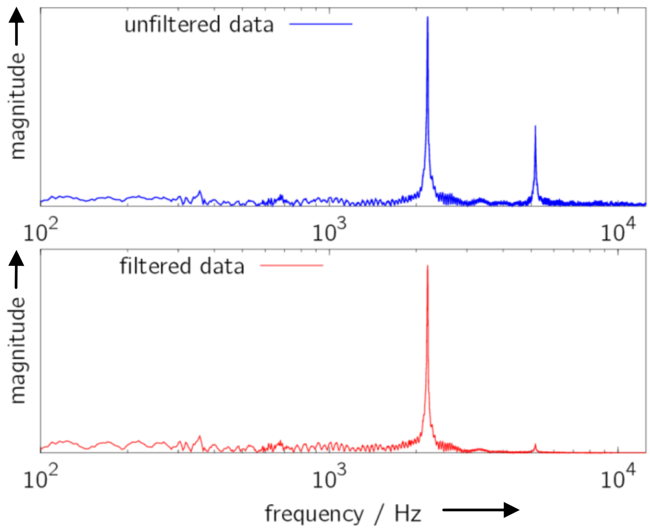

Figure 6: Discrete Fourier transform of unfiltered (blue, top) and filtered (red, bottom) data of vibrometer output.

From the approximation of a monofrequent sine (see Fig. 7), relatively high residuals result. This is not a surprise, as the fit function does not represent the real multifrequent signal well.

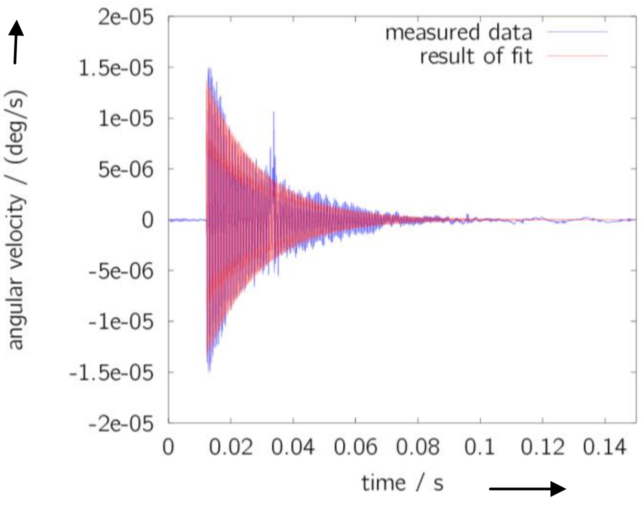

Figure 7: Measured data (blue) and fit (red).

The first measurement results show a relatively high relative standard deviation (in the range of 5 % for 5 measurements) from repeated measurements of the same DUT. Due to the brittle failure of the specimen for the torque step generation, a slight modification of the set-up is carried out for each measurement by replacing the broken specimen and fitting a new one. The discovered standard deviations are significantly higher than the given measurement errors of the rotational vibrometer, which are in the range of $\pm$ 1.5 %.

This is not fully satisfying and will be further investigated, both in terms of checking the set-up and data acquisition, as well as in terms of the data analysis. However, in comparison to other parameters, the damping is supposed to be less critical for the parameter identification and higher measurement uncertainties might be acceptable for this not so easy to measure quantity. Anyway, the deviations will be included in the future evaluation of the measurement uncertainty.

7. CONCLUSIONS

The method presented enables the determination of the damping of rotational oscillations. The non-contact measurement of the oscillations and the avoidance of mechanical modifications of the DUT minimise unwanted influences on the measuring result. Oscillations are generated by a negative step excitation and measured at the top and at the bottom of the measuring device. The acquired data is used for a calculation of the time-dependent torsion angle of the DUT for the analysis of the decay of the generated excitation. The first measurement results are promising. Reasons for the relatively high standard deviation of the measurement results need to be further investigated.

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