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SINUSOIDAL TORQUE CALIBRATION: A DESIGN FOR TRACEABILITY IN DYNAMIC TORQUE CALIBRATION

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Abstract – This paper is concerned with the problem of dynamic torque calibration. It presents the concept of a new calibration device designed at PTB, to provide traceability for dynamic calibration with sinusoidal torque.

Furthermore some details of the measurement system, construction and data analysis will be discussed.

Keywords: harmonic torque, dynamic torque, dynamic calibration.

1. INTRODUCTION

For several years now the industrial demand for dynamic calibration of torque transducers has increased.

The main application demanding such calibrations are powered screwdrivers and engine testing stands.

In contrast, the main methods of torque sensor calibration for industrial application are static type lever-arm devices or continuous calibration devices with reference transducers, which are also considered quasi-static.

In analogy to the development in force some years ago, the Physikalisch-Technische Bundesanstalt (PTB) as the national metrology institute of Germany has started to implement dynamic torque calibration with the design of a facility which allows the traceable application of sinusoidal torque to transducers. In the final set-up the range of this new device limited to 100 N·m at a frequency of max. 100 Hz.

Until today there is no calibration facility in existence which allows calibrations with torque changing rapidly with time and with direct traceability to the base units.

2. THE CONCEPT

The basic concept behind this could be described as “rotational harmonic excitation and moment of inertia” which is in direct analogy to the harmonic force calibration [1].

In a first order approximation the dynamic torque $T(t)$ acting on a transducer can be calculated from the reaction of a moment of inertia (θ_0) coupled to the sensor and its corresponding angular acceleration $\ddot{\varphi}$ as:

$$T(t) = \theta_0 \cdot \ddot{\varphi}(t) \qquad (1)$$

where *t* denotes time.

In order to calibrate a torque transducer with sinusoidal load the sensor is coupled to a rotational exciter on the fixation side and a sufficiently large moment of inertia on its measurement side .

The moment of inertia is basically a piece of mass with at least an axisymmetric mass distribution according to the measurement axis of the transducer, e.g. a flywheel. In order to achieve clean traceability the influence of friction on the measurement side of the transducer (i.e. the flywheel) needs to be minimised by use of a rotational air bearing.

During excitation the generated torque is transferred via the transducer onto the moment of inertia and results in the aforementioned angular acceleration $\ddot{\varphi}$ of the moment of inertia.

By measuring the angular acceleration [2] and the moment of inertia, a precise determination of the applied time-dependent torque is possible according to Eq. (1).

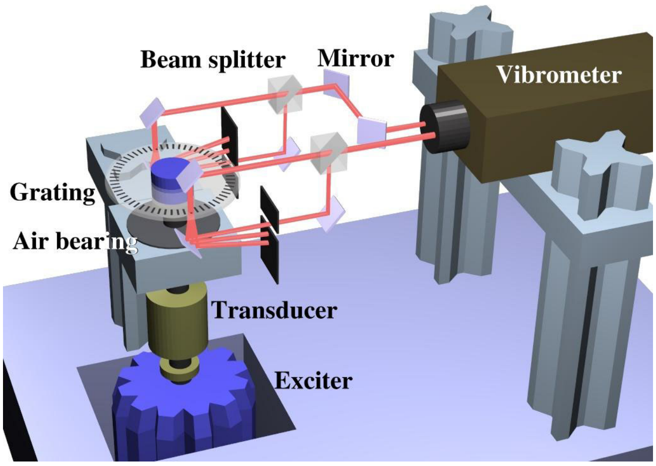

Fig. 1: Basic design of the new dynamic torque calibration device.

2. DESIGN AND METHODOLOGY

The basic design of the new facility is shown in Fig. (1). On the left, the collinear mounting of the exciter (bottom), the transducer (centre), an air bearing and a radial optical grating (top) can be seen.

The moment of inertia in this case is realised by the upper part of the shaft assembly, the rotor of the air bearing and the optical grating attached to it.

On the right of the picture a two-channel vibrometer is shown which is used for the measurement of the angular acceleration in conjunction with the optical grating.

For measurement purposes each beam of the vibrometer is guided via an optical loop through the radial optical grating and back into the vibrometer (cf. Fig. (2)).

On the emission side of the grating the first or a higher order of the diffraction pattern is selected for the re-insertion into the optical loop.

Due to the effect of the diffraction grating the emission has its intensity at well-defined angles which leads to a well-defined relation between the order of diffraction and the Doppler induced phase shift per time upon rotation of the grating. Some easy calculation shows that a rotation of the grating by m grooves (grating constants) per second leads to an optical phase shift of $\alpha = m \cdot n \cdot 2\pi$ per second in the nth order of diffraction.

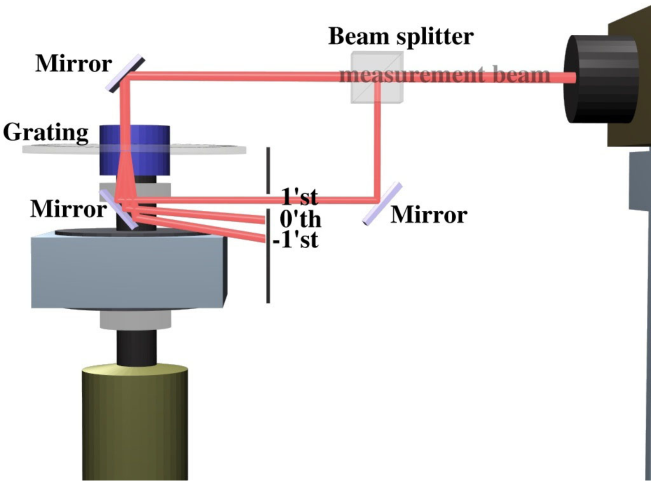

Fig. 2: Diagram of the diffraction measurement in an optical loop.

By measuring this well-defined phase change over time, or the resulting frequency shift, a precise measurement of the angle of rotation or its corresponding angular velocity is possible.

For the phase measurement the vibrometer provides an output of the carrier frequency of its internal Bragg cell at 40 MHz modulated by the phase shift suffered by the measuring beam.

This signal is converted to the 100 kHz band by application of frequency mixing without compromising the phase information. After appropriate low pass filtering for anti-aliasing it is sampled with at least 1 MS/s.

In order to retrieve the included phase information from the rotational shift of the grating a digital quadrature scheme is applied. Therefore the phase-modulated digitised 100 kHz signal is multiplied by a synthetic 50 kHz sine and a 50 kHz cosine series. After low-pass filtering at 75 kHz this procedure yields two signals ($s(t)$ and $c(t)$) at 50 kHz in quadrature with identical phase modulation. The original phase shift can than be extracted from these signals by taking the *arctan* $(\frac{s(t)}{c(t)})$ with subsequent phase unwrapping and subtraction of the linear phase increase generated by the 50 kHz carrier frequency. Fig. (3) is a diagram of this measurement and data analysis chain.

With the retrieval of the phase $\alpha(t)$ the angle of rotation is given by:

$$\varphi(t) = \frac{1}{n \cdot N} \cdot \alpha(t) \tag{3}$$

where N is the total number of lines of the grating and n is the diffraction order used.

For a sinusoidal excitation with a single frequency the angular acceleration can be determined without explicit differentiation of the measured angle of rotation by multiplying the amplitude of the rotational oscillation $\hat{\varphi}$ with the square of the angular frequency ω^2.

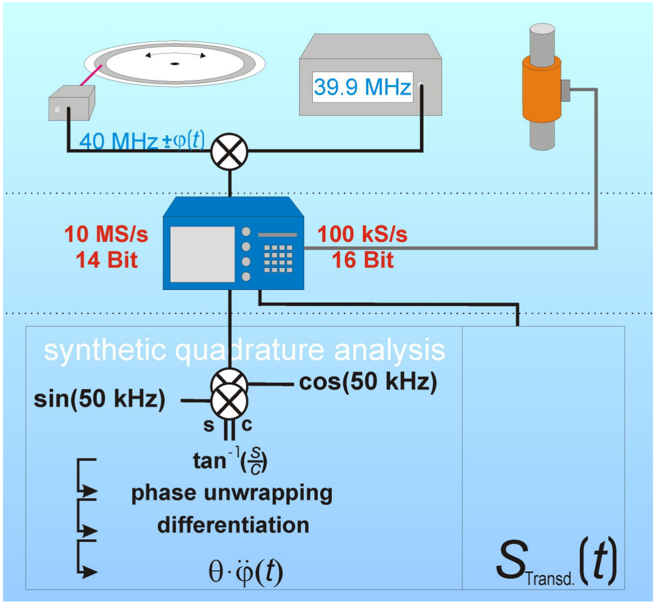

Fig. 3: Diagram of the measurement and data analysis chain used for the phase measurement.

For non-sinusoidal excitations or multi-frequency excitations such as sweeps the angular acceleration has to be determined by digital differentiation of the time series of the angle of rotation $\varphi(t)$.

THE EFFECTIVE MOMENT OF INERTIA

As stated in Eq. (1) the moment of inertia coupled to the sensing element of the transducer is a crucial factor for the determination of the applied torque.

Part of this factor is given by the design of the machine and consists of parts as the

- grating's support disc
- rotor of the air bearing
- primary shaft connected to the rotor
- coupling element for coupling transducer and primary shaft

The total moment of inertia θ_0 of these components does not change during different calibrations and has to be determined with high precision in advance. A measurement of this property is possible by extending the setup of the components with a torsional spring as it is illustrated in Fig. (4). Thus a torsional pendulum is created with period of oscillation

$$\tau = 2\pi\sqrt{\frac{\theta_0}{k}}\,, \qquad (2)$$

where k denotes the stiffness of the torsional spring which is unknown. By adding additional well known moments of inertia (e.g. mass pieces of simple geometric shape) θ_i, a series of measurements of different periods τ_i is possible. For the square of the period follows:

$$\tau_i^2 = \frac{4\pi^2}{k}(\theta_0 + \theta_i) \qquad (3)$$

Thus the value θ_0 can be determined by a linear regression of τ_i^2 vs. θ_i.

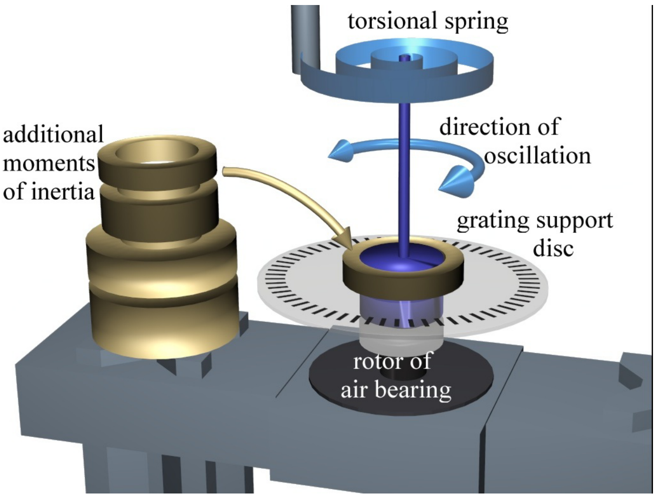

Fig. 4: Illustration of the set-up for the determination of the moment of inertia θ_0 according to eq. (3).

During the described procedure special care must be taken to keep the stiffness of the spring and its coupling conditions constant. Additionally, the torsional spring must carry only a negligible moment of inertia itself. This is possible by using a glass fibre or nylon string for example.

On further inspection some parts of the transducer need to be taken into account additionally, since its mass distribution on the measuring side adds an effective additional component θ_T to the moment of inertia θ_0

Thus equation (1) is better written as:

$$T(t) = (\theta_0 + \theta_\mathrm{T}) \cdot \ddot{\varphi}(t) \qquad (4)$$

At this point the question arises how can one determine this effective contribution of the transducer without detailed knowledge of its internal design.

The solution to this problem is again a measurement with different well known values of $(\theta_0 + \theta_i)$. A linear regression of the ratio of the amplitudes of the transducer's signal and the angular acceleration $T/\ddot{\varphi}$ versus the applied $(\theta_0 + \theta_i)$ yields the value of $-\theta_\mathrm{T}$ as the zero crossing of the line of regression as shown in Fig. (5).

The value of θ_T is a part of the technical specification of the specific torque transducer type which is essential for the dynamic application of the transducer, however, it is rarely documented in the data sheets of torque transducers.

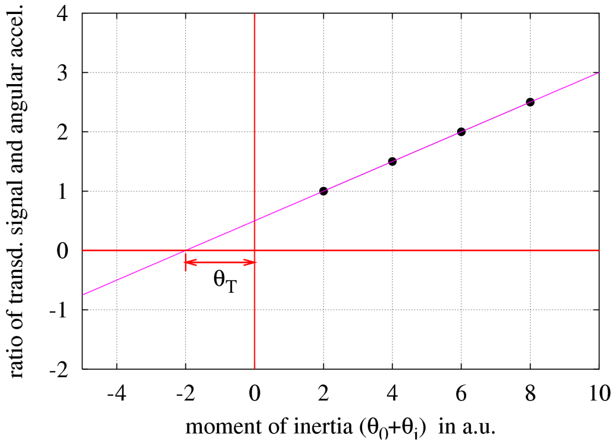

Fig. 5: Determination of the effective moment of inertia θ_T contributed by the transducer from measurements with different θ_0.

TRANSLATORY DISTURBANCE

The two-channel measurement provides the additional option of an analysis and ultimately the elimination of translatory motion artefacts in the vibrometer data.

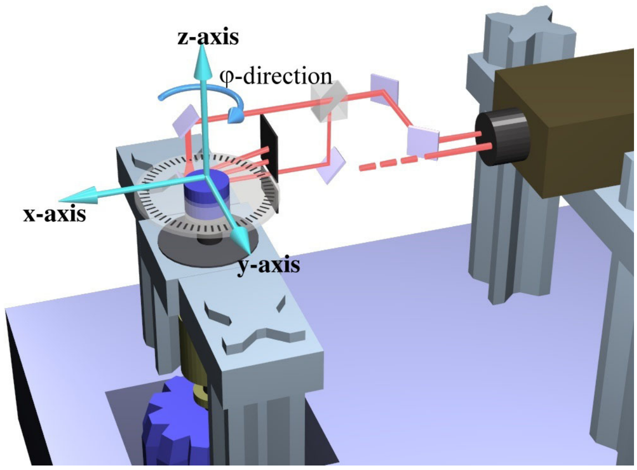

Fig. 6: Co-ordinate system used for the description of movements. (frontal beam is discontinued for better illustration)

Referring to the co-ordinate system introduced in Fig. (6) the influence of the different possible modes of translatory solid body motions of the grating can be discussed.

Vibrations in the z-direction, which may originate from a non-orthogonal misalignment between the axis of rotation and the grating's plane, result in a sinusoidal displacement of the grating disc in the beam direction. Due to the fact that the measurement works in transmission no Doppler shift will result and since the optical path length (corresponding to the thickness of the disc in the beam direction) does not change, despite the movement of the grating, no phase shift generated either. Therefore movements in the z-direction should not influence the measurement even with only a single channel.

There are several options intrinsic to the facility's design which may (unintentionally) generate vibrations in the x- or y-direction, e.g.:

- non axisymmetric mass distribution
- cross forces from the transducer cable
- bending due to misaligned shaft connections

Apart from constructional and operational efforts to minimise these effects, the influence of the generated vibrations on the measurement are already minimised by the design of the machine.

Since vibrations in the y-direction are orthogonal to the beam path of the measurement laser they are not detectable in the phase information. However, such motions have an influence on the effective grating constant perceived by the beam. Thus the direction of the diffraction maximum suffers a small change. This may result in a decrease in intensity. However, since the setup works in transmission the intensity of operation is expected to be some magnitudes higher than those of standard applications of the vibrometer which normally operates in the speckle field reflected from unpolished surfaces.

Vibrations in the x-direction have immediate influence on the measurement of a single channel of the laser vibrometer. A motion of the grating with velocity $V_x(t)$ induces a frequency shift on a measurement laser beam of magnitude

$$\Delta\omega(t) = 2\pi\upsilon_0 \cdot \left(\sqrt{1 - \frac{V_x(t) \cdot \sin\gamma}{c}} - 1 \right)$$

where γ is the angle of diffraction of the diffraction order used and υ_0 the frequency of the vibrometer's laser (ca. $4{,}74 \cdot 10^{14}$ Hz).

However, because of the anti-symmetric choice of diffraction orders re-inserted into the optical loop (n^{th} vs. $-n^{th}$ order) of the respective channel, the frequency shift has an opposite sign for the two vibrometer channels. Thus by simple averaging of the measurements of the two channels it is possible to eliminate the influence of translatory vibrations in the x-direction.

On the other hand it is possible to investigate especially these vibrational components by analysing the difference of the two channels. Since the rotational component has the same sign and magnitude on both channels it cancels out on subtraction and the difference of the two channels includes the frequency shift due to translatory vibration in the x-direction with double magnitude.

CONCLUSION

A new concept for dynamic torque calibration is presented in this paper together with the basic design of a new device currently under development at PTB.
This facility is the first to provide traceable dynamic torque calibration. Its operation is based on the laser interferometric measurement of the angular acceleration via diffraction at a radial optical grating. The data analysis facilitates a synthetic qudrature scheme to retrieve the angle of rotation as a function of time.
The design of the new machine is optimised for the suppression of translatory components and the calibration with sinusoidal torque generated as the reaction of a moment of inertia to angular acceleration.
Finally some special procedures are presented which are necessary for the determination of the various contributions to the effective moment of inertia.

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