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DESIGN OF A FACILITY FOR THE PRECISE SIMULTANEOUS GENERATION AND MEASUREMENT OF FORCE AND TORQUE

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Abstract: This article describes the design of a measuring facility which can be used to investigate and calibrate so-called "friction coefficient sensors". These measuring facilities are used to measure the prestressing force and the tightening torque, resp. the friction torque of screws. These measurements are aimed at optimizing screw joints. The measuring facility described here is part of a force standard machine (fsm). In addition to the force which this system can realize with a very small measurement uncertainty of 0.002 % (k = 2) (in the measuring range from 20 kN to 1 MN), it can also generate an extremely precise torque (objective: better than 0.005 % at k = 2) in the range from 20 N·m to 2 kN·m.

Keywords: multi-component measurement, friction coefficient sensor, screws, clamp force, torque

1. INTRODUCTION

Screw joints are an essential constructional element in nearly all fields of economy and everyday life. The screw industry aims to improve screws and to optimize the interaction between the screw and the material to be fastened, in order to make this fastening element safer and better value. In this context, friction coefficient sensors have been used for a few years. These are multi-component transducers which can measure both an axial force and at least one torque. In the case of screws, the forces involved are: the prestressing force and the tightening torque or the friction torque between the screw head and the support as well as inside the thread. Industry entrusted the Physikalisch-Technische Bundesanstalt with the qualification of the various commercially available friction coefficient sensors using a suitable calibration procedure to enable traceable measurements. In order to meet these needs and to fulfil its mandate (dissemination of the units), the Physikalisch-Technische Bundesanstalt set up a novel measuring facility which enables the simultaneous generation of both a precise axial force and of a very accurately known torque into a friction coefficient measuring system. This paper presents the principle and the design of this multi-component measuring facility.

2. DESIGN AND CONCEPTION OF THE AUXILIARY DEVICE

2.1 SCHEMATIC SET-UP

To attain the forces required for the selected measuring range, it was decided to place a two-armed lever with a horizontal lever side plate into the force flow of PTB's 1-MN force standard machine (1-MN-K-NME, [1]). A force couple generating the required torque is to act at the ends of this lever. The force is transmitted to the lever arm by means of thin metallic bands which act tangentially and are placed parallel to each other. Two mass stacks with different load masses are located on either side of the device to generate force. The forces, which are realized as gravitational forces by means of load masses (principle of the direct deadweight effect), ensure that a sufficiently small measurement uncertainty can be achieved. The vertical force is transformed into a horizontal force by means of a converting device. To this end, an air bearing is used to minimize force losses due to friction.

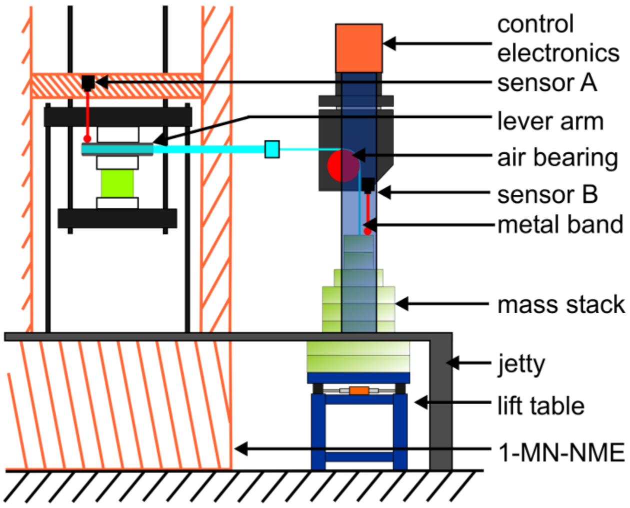

Fig. 1: Schematic representation of the set-up for a loading device, side view with mass stack, air bearing, metal band and lever arm in the set-up of the 1-MN-K-NME

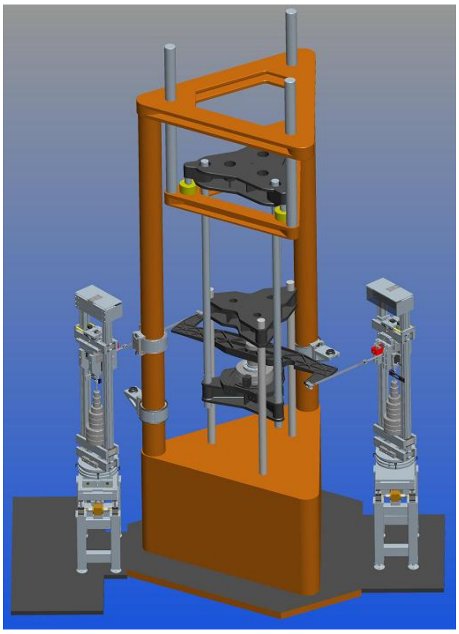

Fig. 2: CAD model of the set-up of the multi-component measuring facility and additional mass stacks

The limited space offers only little clearance for the positioning of the load masses. Lift tables located beneath the crosshead are the optimal solution. The load masses are rested on the lift tables. Two floor panels of the crosshead were removed, in order to allow the loading mechanism to pass through it. The individual weights are applied consecutively by means of a stepper motor. Converting the force effect from a vertical gravitational force into tensile force acting horizontally at the end of the lever posed a problem. Fig. 1 shows a simplified schematic drawing of the planned set-up. The drawing shows one mass stack only – a second one is located on the opposite side. Fig. 2 shows the planned set-up of the auxiliary device in the form of a CAD model. The device in question is a modular principle which can also be applied to other force standard machines.
With the device that is being built, axial forces in the range from 20 kN to 1 MN and torques in the range from 20 N·m to 2 kN·m can be realized.

2.2 TORQUE TRANSMITTING SYSTEM

To generate a pure torque M_z, one needs a force couple with the same force value but of opposite direction. Small deviations from the ideal orientation of the force vectors lead to additional bending moments and to a reduced value of the torque M_z. The force couple needed is realized by means of two identical load masses located at the end of levers facing each other. The exact coordination of the orientation of the mass stacks and of the force bands is particularly challenging.
In order to ensure an optimal positioning of the bands during the loading phase, various sensors are integrated into the system, and the direction can, if necessary, be corrected by means of stepper motors. In the first orientation phase, the spatial position of the mass stacks and of the force bands is determined as a function of the position of the lever. The spatial coordinates are measured using the portable coordinate measuring machine Hexagon Infinite 2.0 in the form of an index arm. First, one of the mass stacks is rotated until the band makes contact with the outer surface of the lever. This position is used as a reference; the coordinate measuring index arm is then mounted onto the opposite side. The coordinate measuring system is maintained, and the relation between the second mass stack and the lever is now determined. The force application point of both bands on the lever must be identical. The second mass stack is shifted on a linear table and rotated by means of a rotary table until this condition is fulfilled.

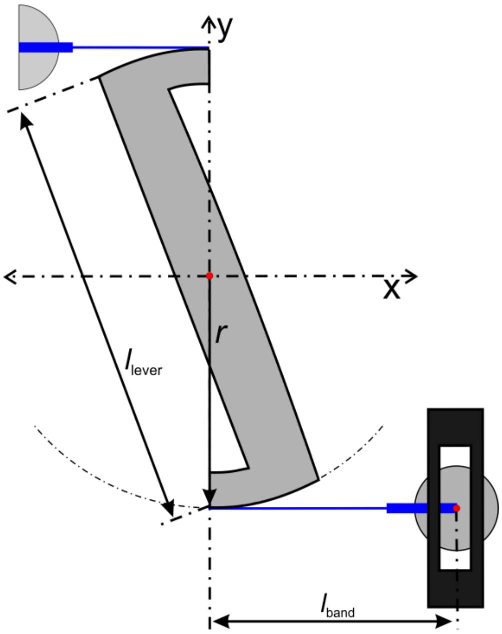

Fig. 3: Schematic top view of the lever system and bands once the orientation of the mass stacks has been completed.

Fig. 3 is a schematic representation of the completed orientation shown from above. The lever plane represents the *x*-*y* plane. The point of origin is defined as the centre of the lever arm.

2.3 VECTORIAL REPRESENTATION

The vectors and positions in space are illustrated in Fig. 4. The points P_{M1}, P_{M2}, P_B and P_0 are measurement points which can be determined by means of the coordinate measuring arm. The points P_{M1} and P_{M2} span the vector $\vec{b}$ which describes the position in space of the axis of the rotor equipped with an air bearing. P_0 lies at the centre of the lever. Point P_B is – geometrically – equidistant from the points P_{M1} and P_{M2}. The point where the band runs off the air bearing shaft is located at the vertical distance *r* (*r* = radius of the air bearing shaft) from the axis.

To simplify the description, all points are considered as lying in the x-y plane. In reality, this assumption can, however, be guaranteed to a certain extent only. When orientating the system, the objective is to come as close as possible to the ideal case $\Delta z = 0$ between all measurement points.

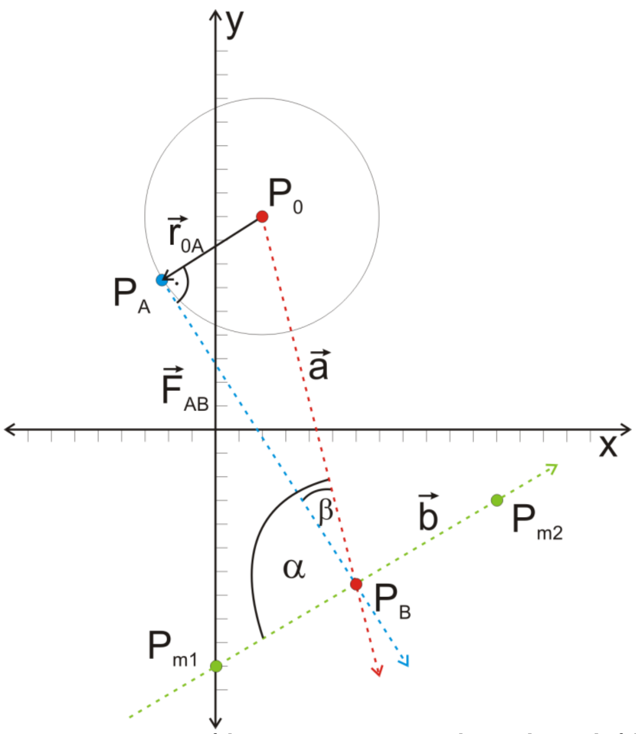

Fig. 4: Representation of the measurement points in the *x-y* plane and of the vectors and angles that can be calculated on this basis and that are necessary to orientate the system.

In order to determine the rotation angle of a mass stack in such a way that the band can be applied tangentially to the lever front surface, the following steps are necessary: determining the vector $\vec{b}$ from P_{M1} and P_{M1}, and the vector $\vec{a}$ on P_0 and P_B. From this, it is possible to calculate the angle α under which the two vectors cross each other. The triangle AB0 must be right-angled. With the lever radius $\vec{R}_{OA}$ and the value of the vector $\vec{a}$ being known, the angle β can be calculated. The rotation angle by which the mass stack must be rotated around P_B is yielded from the difference between the two angles.
The point P_A is the only point that cannot be measured. It represents the force application point and must be determined from the other geometrical quantities.

2.4 SENSORS A-D

During the measurement, displacements of the band, of the air bearing shaft or of the load stack may occur. The individual sensors are there to detect distance changes. With the signals they provide, various stepped motors are controlled which can then compensate for these displacements. The sensors A-D are installed in such a way that it is possible to check the orientation of the metallic bands also during a loading operation of up to 1 MN. In more detail, sensor A is an optical distance sensor (manufacturer: Sensorpart) with a wavelength of 670 nm, type FT-50 RLA-20_S_L4S. The height change between the end of the lever and a fixed reference point is determined with a laser. During the loading, the lever end may drop, the force vector thus changing by an additional *z* component (see Fig. 5). Sensor A controls a stepped motor which varies the height of the air bearing head by the same difference. Sensor B (see Fig. 1) of type FT-50 RLA-40_S_L4S (manufacturer: Sensorpart) has an extended measuring range; its operating principle is identical to that of Sensor A. It is used to determine the distance between the topmost weight and a fixed reference point. Due to a change in the length of the band caused by thermal or mechanical influences, the sequence of the application of the mass has to be adjusted. The weights are lowered and applied onto a central suspension bar. If the motor always drove down exactly the same distance, a change in the length of the band could cause individual weights not to be applied correctly or even not to be applied at all. The data transmitted by sensor B are used to control the lowering effected by the motor.

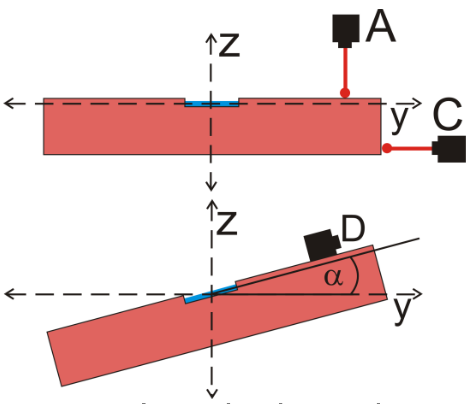

Fig. 5: Sensors A, C and D, used to determine the position and the inclination of the air bearing roll. Displacements are measured optically or capacitatively.

If during the loading operation the force vector $\vec{F}_{ab}$ is no longer perpendicular to the air bearing axis $\vec{b}$, this leads to an additional force effecting on the air bearing shaft. The shaft will then slowly drift laterally inside the air bearing stator. To detect a displacement of the air bearing shaft, sensor C, model BAW M12MG2-IAC208 BP03 (manufacturer: Balluff), is used to measure the distance capacitatively, sensor D to determine the inclination (see Fig. 5). The inclination sensor is custom-made (manufacturer: Seika); it has a measuring range of ± 0.2° and a resolution of $\sim 1 \cdot 10^{-6}$ °. The two sensors control stepped motors which adjust the position of the roll by varying the inclination and by displacing it along the air bearing axis. The optimal position is attained when no more drift displacements are observed.

3. DETAILS

3.1 AXIAL FORCE OF THE 1-MN FORCE STANDARD MACHINE

The axial force is realized by the 1-MN force standard machine – a dead-weight machine. The measurement uncertainty of the 1-MN force standard machine is known sufficiently well and has been investigated on different occasions [2, 3, 4]. The relative measurement uncertainty ($k = 2$) of the 1-MN force standard machine is $2 \cdot 10^{-5}$. The deviation from the ideal force vector development was investigated in [6]. This investigation showed that, at an

axial force F_z = 80 kN, shearing forces $F_x = F_y$ ~ 10 N can be expected.
Furthermore, a 6-component measuring platform was implemented in the upper part of the force standard machine in order to, in future, be able to measure also parasitical influences.

Fig. 6: The figure shows the first version of the aluminium lever mounted on the 1-MN force standard machine. The shown shape has been retained for the INVAR lever. One can also see the measurement points used to determined the position towards which the coordinate measuring machine later moves.

3.2 LEVER ARM

The lever used to generate the torque has a length of 2 m and an uncertainty Δl = 1 µm. Fig. 6 shows a version made of aluminium. The shape has been chosen in such a way that a torque can be generated in both directions (left and right), hereby exploiting the space of installation optimally. In order to reduce changes in torque due to the dilatation of the lever arm, INVAR steel was chosen as a material, since it has a very low linear thermal expansion coefficient of $< 2 \cdot 10^{-6}$ K^{-1} at room temperature.
If the temperature changes by 1 K, the length of the lever changes by $\Delta l = 3.7 \cdot 10^{-3}$ mm. The torque would then change by $\Delta M_z = 3.7 \cdot 10^{-6}$ N·m.

3.3 MASSES

The additional mass stacks located on either side are composed of 10 individual weights. One stack has load masses for the following weight forces: 1 x 10 N, 2 x 20 N, 1 x 50 N, 3 x 100 N and 3 x 200 N. The maximum load mass is approx. 100 kg. Each mass stack thus generates a couple of 1000 N. The material chosen for the weights is X2CrNiMoN 18-14. This is a corrosion-resistant steel which is not easily magnetizable and is mainly used in maritime construction. For the masses, an uncertainty of $\Delta m = 5 \cdot 10^{-6}$ kg can be stated.

3.4 RESULTING FORCE

For a realistic estimation of the resulting force, also ambient influences, such as temperature, pressure, and humidity, have to be taken into account. The air buoyancy contributes to reducing the effective mass. The value of the force can be calculated by means of the following formula:

$$F = m \cdot g \cdot \left(1 - \frac{\rho_{\mathrm{air}}}{\rho_{\mathrm{m}}}\right) \quad \textbf{(1)}$$

$$\rho_{air} = \frac{0{,}34848 \cdot p - 0{,}009024 \cdot h_{\mathrm{r}} \cdot e^{0{,}0612T}}{273{,}15 + T} \quad \textbf{(2)}$$

m - mass	g - 9.81253 m·s^{-1}
p - pressure	h_r - humidity
T - temperature	ρ_m - density of the weights

In order to keep ambient influences, such as changes in temperature, pressure and humidity, as low as possible, the complete measurement set-up is located in a fully air-conditioned hall. The temperature varies in the range of ΔT = 0.2 K, and Δh_r = 5 % for the humidity. The gravitational acceleration inside the measurement hall was determined as $\Delta g = 1 \cdot 10^{-5}$ m·s^{-1}.

3.5 TWO-COMPONENT FORCE-TORQUE SENSOR

For the calibration of the two-component measuring equipment, a transfer standard with a sufficiently small relative measurement uncertainty of $< 1 \cdot 10^{-4}$ is required. The transfer standard must be a two-component transducer covering a measuring range with axial forces F_z ~ 500 kN and torques M_z ~ 500 N·m. The selected force/torque transducer is realized in the form of a build-up system from the company *GTM, Gassmann Testing and Metrology GmbH*. The two transducers are a precision force transducer KA-K-250kN-F-1mV/V [6], and a custom-made torque transducer with the designation Dm-M-500_N·m-2mV/V. The sensor is calibrated separately with regard to the axial force and with regard to torque. The adaption parts allow it to be mounted in the 1-MN force standard machine for an axial load with a relative measurement uncertainty of $2 \cdot 10^{-5}$ as well as in the 1-kN·m torque standard machine (1-kN·m-DmNME) [7, 8] with a relative measurement uncertainty (k = 2) of $2 \cdot 10^{-5}$. Afterwards, the cross-talk of the transducer when loaded simultaneously with F_z and M_z can be investigated in the two-component measuring equipment.

4. VECTORIAL ERROR ANALYSIS

Generating a torque by means of a couple requires a vectorial approach (see Fig. 4). The direction of both the force vector and the distance vector must be determined. Deviations from the orthogonal orientation of the two vectors in the x-y plane make it necessary to consider vectorial error analysis. If only one side of the system is taken into account, the torque obtained then results from the vector product of the distance and the force vectors.

$$\vec{M} = \vec{r}_{0\mathrm{A}} \times \vec{F}_{\mathrm{AB}} = \vec{r}_{0\mathrm{A}} \times \frac{\vec{r}_{AB}}{\left|\vec{r}_{AB}\right|} \cdot F \qquad \textbf{(3)}$$

In order to obtain a statement with regard to the individual moments, one needs the complete data concerning the position of the vectors $\vec{R}_{0\mathrm{A}}$ and $\vec{R}_{\mathrm{AB}}$. The vector product is solved in Equation (3).

$$\vec{M} = \frac{F}{\left|\vec{r}_{AB}\right|} \cdot \begin{bmatrix} r_{0Ay} r_{ABz} - r_{0Az} r_{ABy} \\ r_{0Az} r_{ABx} - r_{0Ax} r_{ABz} \\ r_{0Ax} r_{ABy} - r_{0Ay} r_{ABx} \end{bmatrix} = \begin{pmatrix} M_x \\ M_y \\ M_z \end{pmatrix} \qquad \textbf{(4)}$$

The problem is that the exact position of the force application point P_A is not known. It is therefore not possible to measure directly with the index arm. The point P_A can, however, be calculated from points P_0 and P_B, from the knowledge of the value of the vector $\vec{R}_{0\mathrm{A}}$, but also from the fact that the triangle AB0 must be right-angled. The values for R_{0Ax}, R_{0Ay} and R_{0Az} can be determined with the aid of the software of the coordinate measuring arm – an uncertainty statement is, however, not given. To obtain an uncertainty for the point P_A, the internal calculation must be reconstructed. A detailed calculation of P_{0A}, of the individual components M_x, M_y and M_z, as well as of their uncertainty will be the subject of another publication.
For the torque, a relative measurement uncertainty in the range of $5 \cdot 10^{-5}$ is aimed at. An accurate analysis of the measurement uncertainty budget from sources of errors such as, e.g., the mass, lever length and bending, orientation and length of the metallic band, and ambient influences will also be the subject of a subsequent, detailed analysis.

6. OUTLOOK

The relative measurement uncertainty ($k = 2$) of the 1-MN force standard machine is $2 \cdot 10^{-5}$. According to first estimations, we are expecting a relative measurement uncertainty in a range smaller than $5 \cdot 10^{-5}$ for torque generation. The measurement uncertainty budget will be considered to a greater extent after the facility has been commissioned. The aim is to reduce the measurement uncertainty by optimizing the measurement cycle with regard to the acquisition of the spatial coordinates. It is expected that this novel measurement facility will lay the basis for the calibration of friction coefficient sensors. A first step towards this is calibration by means of a transfer standard. In addition, a comparison of diverse commercial friction coefficient sensors is aimed at.
Furthermore, the development of the facility is to be pursued with regard to a possible introduction of bending moments into multi-component transducers.

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