DOI 10.5162/sensor2013/A1.2

Force/Torque measuring facility for friction coefficient and multi-component sensors

S. Baumgarten[1], D. Röske[1], H. Kahmann[1], D. Mauersberger[1] and R. Kumme[1]
[1]Physikalisch-Technische Bundesanstalt (PTB), FB 1.2, Germany, Bundesallee 100, 38116 Braunschweig
E-Mail: sebastian.baumgarten@ptb.de

Abstract

This article describes the design of a measuring facility which can be used to investigate and calibrate friction coefficient sensors. These measuring devices are used to measure the prestressing force and the tightening torque and/or the friction torques of bolted joints. The measuring facility described here enhances a 1-MN deadweight-type force standard machine (fsm). In addition to the force which this system can realize with a very small expanded relative uncertainty of 0.002 % (k = 2) (in the measuring range from 20 kN to 1 MN), it is the aim to generate an extremely precise torque (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, prestressing force, torque

1 Introduction

Bolted joints are an essential constructional element in nearly all fields of the economy and of 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 and/or the friction torque between the screw head and the support as well as inside the thread. With these measurement data, the friction coefficient between the screw and the material can be determined, especially as a function of different lubricants and types of surface finish. Industry entrusted the Physikalisch-Technische Bundesanstalt (PTB) with the task of qualifying 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 (i.e. disseminating the unit), the PTB 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 Conception of the auxiliary device

The calibration of friction coefficient sensors requires axial forces in a force range of up to 1 MN as well as torques of the order of 1kN·m. Setting up an auxiliary device for PTB's 1 MN fsm (1-MN-K-NME) seemed a suitable solution. To this end, a two-armed lever is integrated into the force flow of this facility; a metallic band is fastened at both ends of this lever for force application purposes. The basic idea is to generate a force couple which acts on the lever and consists of two parallel forces of equal value which, although parallel to each other but act in the opposite direction to each other. Each of these two forces acts on one end of the lever. Considered as shearing forces, these two forces cancel each other out due to their opposite signs. Therefore, they do not generate a bending moment around a horizontal axis. Only the desired torque is yielded from the product of the force value and of the total length of the lever. In combination with the axial forces in the range from 20 kN to 1 MN, torques in the range from 20 N·m to 2 kN·m can be realized with the aid of the completed auxiliary device (see Fig. 1 and Fig. 2).

3 Setup of the auxiliary construction

The axial force is realized by the "1-MN-K-NME" deadweight fsm. The measurement uncertainty of the 1-MN force standard machine is known sufficiently well and has been investigated on different occasions [2, 3, 4]. Their expanded relative uncertainty (k = 2) is approx. $2 \cdot 10^{-5}$. The deviation from the ideal force vector development was investigated in [4]: it turned out that at an axial force of

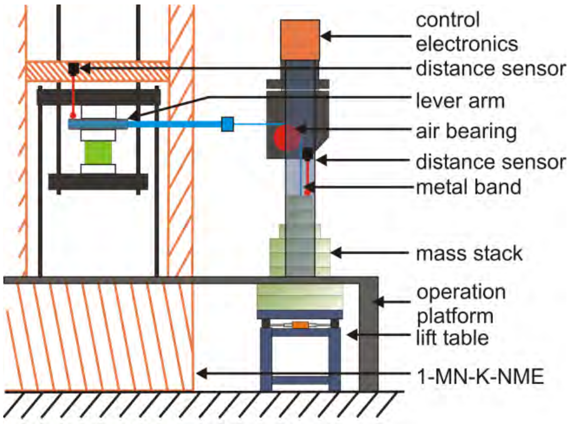

Fig. 1: Schematic representation of the setup for a mass stack; the second one is located symmetrically on the opposite side.

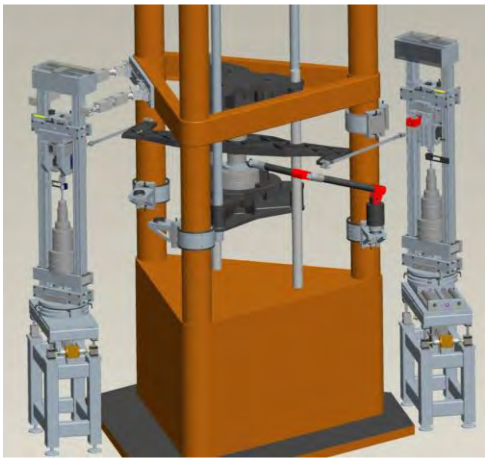

Fig. 2: The figure shows the complete setup of the auxiliary construction as a CAD-model.

F_z = 80 kN, shearing forces of $F_x = F_y \sim 10$ N can be expected. Normal calibration operation for tension and pressure transducers has to be ensured in the 1-MN force standard machine also after the modification. The additional construction was designed in such a way that a simple as well as complete assembly of all its modules is possible at all times. 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. In the following, the individual modules and their utilization will be explained in more detail.

3.1 Lift tables, rotary tables, and linear tables

A lift table is located below the mass stack attachment and the operation platform (see Fig. 3). In order to minimize interactions, this lift table is rested on a base which is decoupled from that of the 1-MN force standard machine. The height of the lift table is adjusted in such a way that the linear table mounted onto it

Fig. 3: Lift table set up below the operation platform of the 1-MN force standard machine.

Fig. 4: Rotary table mounted onto a linear table; the latter is connected to the lift table. These are used to shift and adjust the complete mass stack.

(see Fig. 4) is in perfect agreement with the level of the operation platform. By means of the lineartable, the complete mass stack can be positioned in 5 µm steps over a total length of 620 mm with a repeatability of ±0.66 µm. The rotary table mounted on top rotates the complete mass stack in such a way that the metallic bands can be positioned parallel to each other. An angle sensor determines the rotation with a resolution of 5.5×10^{-4} ° and a relative measurement uncertainty ($k = 2$) of 1.4×10^{-5} °.

3.2 Frame

The outer frame, which is made of aluminium sections (see Fig. 5), is bolted to the rotating disc. The full air-bearing head is mounted in the central area; it consists of an air bearing, a rotoras well as sensors and stepped motors in order to ensure reproducible positioning. The load mass stack is integrated into the lower part. The control unit and the electronics are fitted at the top end. In addition, the frame is equipped with two arms conceived as bracket

Fig. 5: Frame with two bracket arms. The load masses will be located in the lower part; the air-bearing head would be in the centre, the electronic control system is mounted in the upper part.

elements; these are firmly assembled with the outer frame of the 1-MN fsm. These arms support the set-up against the shearing forces occurring under load conditions. To lift the whole set-up onto the operation platform, an additional loop for sling gears is attached at the top end of the frame.

3.3 Mass stacks

The additional mass stacks located on either side of the device consist of ten individual deadweights (see Fig. 6). 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. Both mass stacks together thus generate a couple of two forces with 1000 N each. The material chosen for the weights is X2CrNiMoN 18-14. This is a type of stainless steel which is corrosion-resistant and not easilymagnetizable. For the masses, a relative uncertainty of 5×10^{-6} can be stated.

The load mass stacks are conceived as chain attachment, i.e. the lift table is progressively lowered, which causes each load mass to move downwards, one after the other, and to be introduced in the order of their arrangement on the mass stack. Hereby, the direct deadweight effect principle is applied. By introducing additional mass, however, also the metallic band is extended and the transducer, together with the lever, is twisted. This change in length – or deformation and distortion – can lead to an additional lowering of the masses. Therefore, an optical distance sensor measures the distance between the upper mass slice and a fixed point by means of a laser with a wavelength of 670 nm. Height deviations are

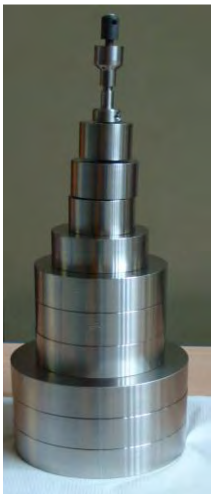

Fig. 6: Load mass stack with ten individual mass slices of X2CrNiMoN 18-14 steel.

Fig. 7: The figure shows the air-bearing head with a place in the middle for the air bearing and rotor. On the right side you can see the rail for the height adjustment.

corrected by means of an optical feedback to the stepped motor.

3.4 Air-bearing head

Converting the force effect from a vertical gravitational force into a tensile force acting horizontally at the end of the lever was a challenge. The converting device used is an air bearing whose rotor is used to roll off the metallic band; the advantage hereby is that only a small part of the force is lost due to friction effects (see Fig. 7). The rotor is a high-precision polished shaft with a guidance rail for the metallic band. The air bearing is made of a porous material, which guarantees the formation of a high-capacity air film of a similar shape. Position changes at this point cause additional shearing forces and bending moments and must, thus, be kept as small aspossible. To this end, a special tilt sensor and a capacitive sensor are mounted into the air-bearing head. The inclination sensor is custom-made for the measuring range of ±0.2° with a

DOI 10.5162/sensor2013/A1.2

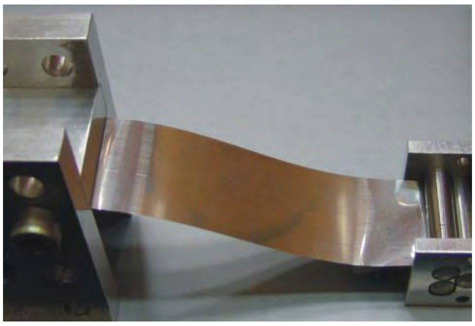

Fig. 8: Metallic band between the coupling element of the mass attachment and the element for connection with from lever coming band.

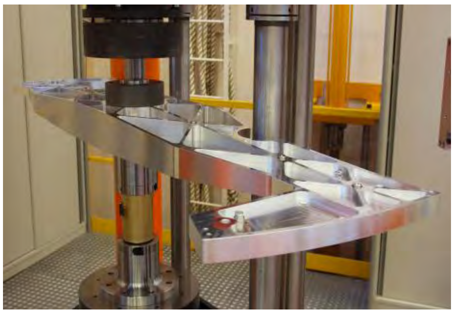

Fig. 9: The photo shows the first variant of the aluminium lever mounted on the 1-MN fsm. The form depicted will be used for the second variant made of INVAR.

resolution of ~ 1 × 10^{-6} °. Combined with each other, these sensors measure both the tilt and a potential drift of the rotor in the air bearing. The feedback ensures, via stepped motors, the displacement and rotation of the complete air-bearing head in order to stabilize its position.

3.5 Metallic bands

A metallic band (see Fig. 8) consists of two individual parts. Both parts are made of steel, are 30 mm in height and 0.3 mm thick. One band is connected to the load masses, whereas the other one is firmly clamped to the lever side plate. Whereas the part of the band which is attached to the lever end has a vertical orientation, the second part has a horizontal orientation once it has been unwound by the air-bearing rotor, i.e. the two parts run parallel but orthogonally to each other and have to be firmly connected by means of coupling elements. Due to thermal expansion, the metallic bands are subjected to additional expansion, which, under laboratory conditions, is, however, small compared to the other effects.

3.6 Lever arm

The lever used to generate the torque has a length of 2 m and an uncertainty Δl = 1 µm. Fig. 9 shows a variant made of aluminium. The

Fig. 10: The photo shows the portable coordinate measuring arm which is mounted onto one of the supports of the 1-MN force standard machine by means of a mounting ring.

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 thermal expansionof the lever arm, INVAR steel was chosen as a material for the actual lever, since it has a very low linear thermal expansion coefficient of < 2 × 10^{-6} K^{-1} at room temperature. When the temperature changes by 1 K, the lever undergoes a change in length of Δl = 3.7 × 10^{-3} mm. The torque would be subjected to a change by ΔM_z = 3.7 × 10^{-6} N·m.

Under the influence of the dead weight and of the load of the 1-MN fsm, the lever will be deformed. This leads to an disturbance value of the metallic bands to each other. An optical distance sensor with a wavelength of 670 nm is mounted onto the frame of the 1-MN fsm; this sensor measures continuously the change inheight of the lever side plate by means of a laser. If the position shifts, an feedback with a stepped motor in the air-bearing head is triggered off, so that the height of the air bearing is re-adjusted by the differential amount measured.

4 Realisation and adjustment

4.1 Coordinate measuring index

In order to adjust the whole set-up as accurately as possible, the position of each module has to be determined individually. This is done by means of a portable coordinate measuring index (see Fig. 10). This index can be mounted onto the frame pillars of the 1-MN fsm. Individual points can be measured with a

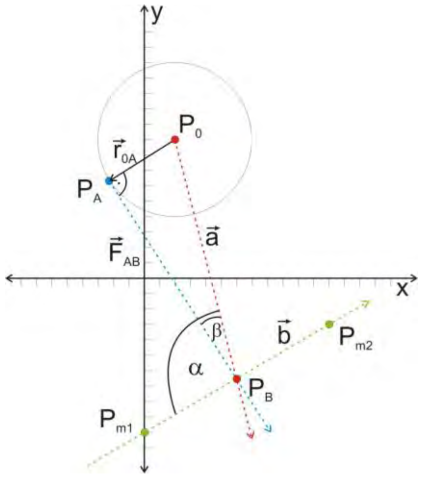

Fig. 11: Representation of the measurement points in the xy plane and of the vectors and angles that can be calculated on this basis and that are necessary to orientate the system.

scanning measurement procedure with a relative measurement uncertainty ($k = 2$) of $\Delta x = 10$ µm at best. The operating distance is 1200 mm, which does not allow all characteristics of the system to be determined from one fixed position. The measuring arm can thus be mounted on any of the tree pillars of the 1-MN fsm, hereby keeping the software assistance of the coordinate system as long as a reference length exists. The reference length used here is the horizontal diagonal of the lever. The measured position coordinates are transmitted, and from these, the shifts and rotation angles required are calculated.

4.2 Vectorial representation

The vectors and positions in space are illustrated in Fig. 11. The points P_{m1}, P_{m2}, P_B, 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 anair 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 xy 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. In order to determine the rotation angle of a mass stack in such a way that the band can be applied tangentially to the

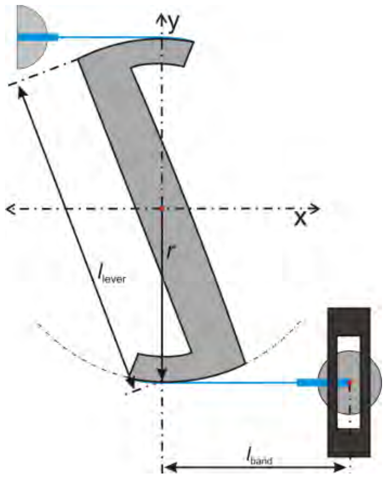

Fig. 12: Schematic top view of the lever system and bands once the adjustment of the mass stacks has been completed.

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 und P_B; from this, the angle α can be calculated 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.

4.3 Adjustment of the load masses

In the first adjustment phase, the spatial position of the mass stacks and of the force bands is determined as a function of the position of the lever. First, one of the massstacks is rotated until the band makes contact with the outer surface of the lever. This position is taken as reference, then the position of the second load mass stack is determined as a function of the position of the lever. The force application point of both bands on the lever must be identical. From the coordinate points of the second mass stack, the required shift and the rotation angle can be calculated in order to meet this condition. The second mass stack is shifted on a linear table and rotated by means of a rotary table according to the result obtained. Fig. 12 is a schematic representation of the completed orientation shown from above. The point of origin is defined as the centre of the lever arm.

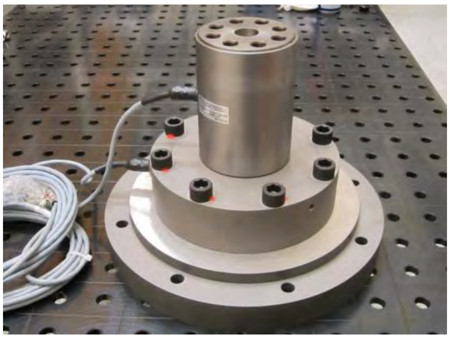

Fig. 13: The figure shows the tension/torsion transducer designed as a build-up system, for the calibration of the auxiliary device.

4.4 Force/torque sensor

The calibration of the auxiliary device requires a transfer standard with a sufficiently small relative measurement uncertainty of $< 1 \cdot 10^{-4}$ and crosstalk signal contributions as small aspossible. The transducer is a tension/torsion model of the type of a force/torque build-up system which can cover a measuring range of axial forces F_z ~ 500 kN and a range of torques M_z ~ 500 N·m. The adaption parts allow this force/torque sensor to be mounted into the 1-MN force standard machine for an axial load with a relative measurement uncertainty of 2×10^{-5}, as well as calibration in the 1-kN·m torque standard machine (1-kN·m-DmNME) [5] with a relative measurement uncertainty (k = 2) of 2×10^{-5}.

6 Outlook

The relative measurement uncertainty (k = 2) of the 1-MN force standard machine is 2×10^{-5}. According to first estimations, we are expecting a relative measurement uncertainty in a range smaller than 5×10^{-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 generation of bending moments. In the medium term, the investigation of multi-component transducers is planned.

References

[1] W. Weiler, M. Peters, H. Gassmann, H. Fricke, W. Ackerschott, “Die 1-MN-Normalmeßeinrichtung der PTB Braunschweig.”, VDI-Z 120, 1978, pp. 1-6

[2] A. Sawla, W. Weiler, M. Peters, A. Bray, R. Levi, M. Vattasso “A comparsion of force standards between the IMGC and the PTB.”, 6th Conf IMEKO TC3, Odessa, pp. 1-13

[3] W. Weiler, A. Sawla “Force Standard machines of the National Institutes for metrology”, PTB Bericht Me-22, 1978

[4] C. Ferrero, “The measurement of parasitic components in national force standard machines”, Measurement Vol. 8 No. 2, Apr-Jun 1990, pp. 66-76

[5] K. Adolf, D. Mauersberger, D. Peschel “Specifications and Uncertainty of Measurement of the PTB’s 1kNm Torque Standard Machine”, Proceedings of the 14th IMEKO TC3 Conference, September 5-8, 1995, Warsaw, Poland, pp. 178-182