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In-situ fine adjustment system for in-vacuo weighing cells

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Abstract

For traceable determination of mass, mass comparators and precision balances with a weighing cell as core unit are used. A further reduction of the measurement uncertainty is achieved by operation in a hermetically sealed chamber under vacuum conditions. The adjustment of the weighing cell required for high-resolution measurements is currently carried out manually under atmospheric conditions. This limits the achievable adjustment quality since the verification of the adjusted state is only possible under working conditions in the closed chamber.
This paper presents a novel system that enables the in-situ adjustment of high-precision weighing cells. The result of the adjustment can be directly controlled in a closed loop, thus fulfilling the conditions for a determined fine adjustment. The required high sensitivity of the moving parts of the weighing cell to be adjusted demands the minimization of disturbing influences to the largest possible extent. Based on a systematic investigation, sources of disturbances are eliminated and unavoidable residual systematic deviations are significantly reduced. The adjustment is performed by high-resolution actuators integrated directly into the moving elements. Power is supplied exclusively during the adjustment process via highly flexible electrical wires, which are connected via a separate rack-mounted drive system with low disturbances. The contact is released before the actual measuring process. Therefore, disturbances by the adjustment system are completely avoided during the measurement process. With the novel low-disturbance in-situ adjustment system, a significant increase in the performance of high-precision weighing cells can be achieved.

Keywords: weighing cell, in-situ adjustment, adjustment system, disturbance effects

1. Introduction

Comparison between mass standards at the top end of the dissemination chain demands mass measurements with highest resolution and least possible uncertainty. State-of-art mass comparators and precision balances use force-compensated weighing cells as core unit [1,2]. A monolithic linkage with filigree flexure hinges serves as mechanical structure to achieve highest sensitivity and repeatability. To avoid deviations from external sources, such devices are often operated in hermetically sealed vacuum chambers. Measurement uncertainty is further reduced by compensating remaining systematic deviations through fine adjustments of movable elements of the mechanism [3].

Fine adjustments are carried out on the arrested weighing cell using manually-driven built-in mechanisms under atmospheric conditions [4]. However, the adjusted state can only be verified under working conditions inside the vacuum chamber. Due to the large adjustment loop, an ex-situ adjustment is associated with a high uncertainty. The achievable performance of the device is, thus, limited. Conventional remotely-controlled drives cannot be integrated due to the highly unstable mechanical disturbances brought by their electric cables.

The following contribution presents a novel system for the in-situ adjustment of high-precision weighing cells. The system achieves the adjustment inside the sealed vacuum chamber with minimal disturbances and without arresting the weighing cell. Any sources of disturbance are eliminated from the weighing cell after the adjustment process, allowing for high stability of the adjusted value.

2. Adjustment of weighing cells

Figure 1 shows the functioning principle of a weighing cell based on the principle of electromagnetic force compensation. The weight force of the measurand induces a deflection of the compliant mechanism, which is detected by a position indicator at point M. Then, the force of the voice coil actuator at point K is regulated to compensate the deflection. The electric current required by the actuator is used as indirect measure of the mass of the measurand. A compensation mass is often built on the balance beam (GK) to reduce the electromagnetic compensation force and its associated disturbances.

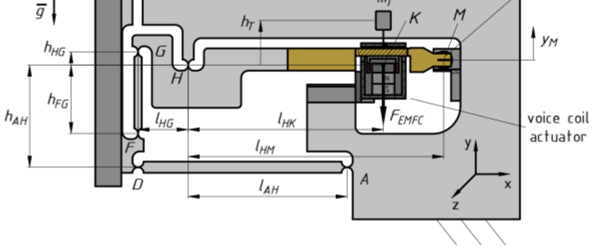

Figure 1. Functioning principle of a weighing cell based on the principle of electromagnetic force compensation

The inherent mechanical stiffness of the compliant mechanism C limit the achievable measuring resolution of the weighing cell, i.e. $r_m = C \cdot g^{-1} \cdot r_{y_M}$. Furthermore, restoring forces due to residual deflections u_{y_M} around the zero position represent a major source of uncertainty, i.e. $u_{m,y_M} = C \cdot g^{-1} \cdot u_{y_M}$. Thus, the stiffness of the system must be adjusted to an approximate zero value for maximum performance. Height-adjustable trim masses are built on movable elements of the mechanism for this purpose [3]. Highest sensitivity is achieved with adjustments on the balance beam, see Figure 1. The variation in stiffness ΔC as a function of the vertical position h_T of the trim mass m_T can be approximated by the following equation:

$$\Delta C \approx -\frac{m_T \cdot g \cdot h_T}{l_{HG} \cdot l_{HM}} \tag{1}$$

While other adjustments, e.g. tilt and corner-load sensitivity, are required maximize performance, this contribution focuses solely on the adjustment of the mechanical stiffness. Subject of this investigation is a weighing cell mechanism widely treated in previous works [3-6]. Table 1 shows the the parameters of the compliant mechanism. Due to its high sensitivity, it is a suitable application example for the targeted adjustment system.

Table 1 Ideal parameters of the weighing cell

Parameter	Value	Parameter	Value
Young's modulus E	71 GPa	length l_{HG}	27 mm
Poisson's ratio ν	0.33	length l_{HK}	105 mm
Density ρ	2.8 g/cm²	length l_{AD}	112.5 mm
gravity acceleration g	9.81 m/s²	length l_{BC}	112.5 mm
min. hinge height h	0.05 mm	length l_{HM}	137.5 mm
hinge radius R	3 mm	length l_{AH}	85.5 mm
hinge width w	10 mm	height h_{AH}	55 mm
height h_{HG}	3.15 mm	height h_{FG}	40 mm

Figure 2 shows the adjustment behavior of the investigated weighing cell according to equation 1, finite element simulations as well as measurements on a prototype. The finite element model is constructed as suggested in previous works [3,6]. The adjustment on the prototype is realized manually on the arrested weighing cell by means of built-in screws. The results are in good accordance with each other. The adjustment sensitivity amounts to $dC/dM = -0{,}2369$ N/m/Nmm, where the momentum is defined as:

$$M = m_T \cdot g \cdot h_T \tag{2}$$

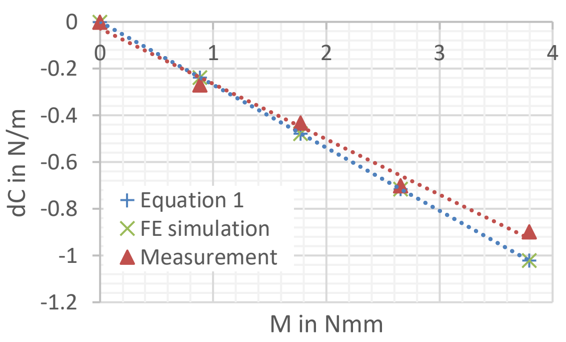

Figure 2. Stiffness adjustment of weighing cell according to analytical and finite element models as well as measurements on prototype.

On the current prototype, the stiffness can be adjusted to a value of $C < 0.01$ N/m. This corresponds to a contribution to the measurement uncertainty of $u_{m,C} < 1$ ng for an error of the position sensor of $u_{y_M} = 1$ nm [7]. To further increase the performance, the novel in-situ adjustment system must realize an adjustment of the momentum better than 4,21 mNmm. Due to the high sensitivity of the mechanism, disturbances introduced by the fine adjustment system on the adjustment location (point K in Figure 1) require special consideration.

3. Behavior regarding disturbances

A previous investigation in [6] shows that mechanical and thermal disturbances on the adjustment location must be completely avoided during the measurement process. Limiting these down to acceptable values regarding the admissible mass measurement deviation may not be technologically possible. However, the püermissible limit values regarding the admissible stiffness deviation during the adjustment are orders of magnitude higher. Disturbances could be, therefore, tolerable only during the adjustment process and must be eliminated afterwards.

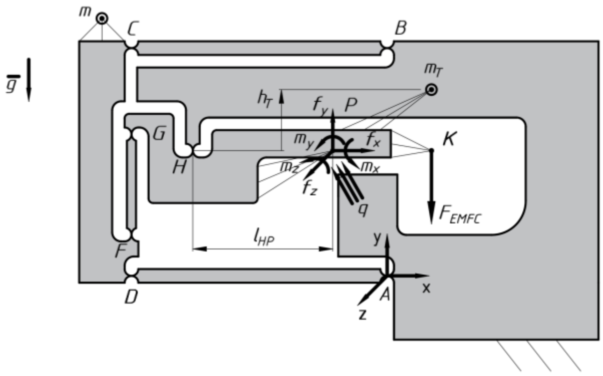

Figure 3. Model of the weighing cell for investigation of disturbance sensitivities and their application point

To further reduce the influence of disturbances by the adjustment system, the investigation based on the finite element model is extended to find a location on the balance beam with minimum sensitivity (see Figure 3). Figure 4 shows the variation of the critical disturbance sensitivity to the forces in x-direction f_x due to the position l_{HP} in x-direction of the application point relative to the main pivot H. A strong reduction can be observed as the application point P tends towards H, $l_{HP} \longrightarrow 0$. This is due to the reduction of the deflection-dependent lever arm, and thus, of the torque.

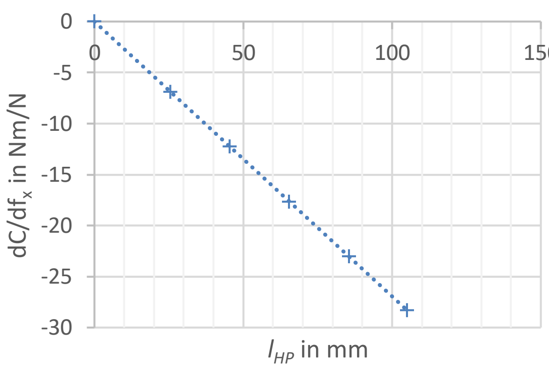

Figure 4. Variation of sensitivity of the stiffness to parasitic force f_x due to the application point l_{HP}.

On the contrary, the sensitivity to forces in y-direction f_y increases as its application point approaches the main pivot H (see Figure 5). This is due to the effective centre of rotation of the balance beam not coinciding with the geometric centre of the flexure hinge, where the load is applied. Thus, an effect similar as the one described in Equation 1 occurs. Further investigations have shown that the sensitivity to f_y is dependent on the loading of the weighing cell (m and m_T), which induces an displacement of the effective centre of rotation. The sensitivity to forces on the z-direction f_z decreases progressively as l_{HP} reduces to zero. Other sensitivities do not show any change in behavior.

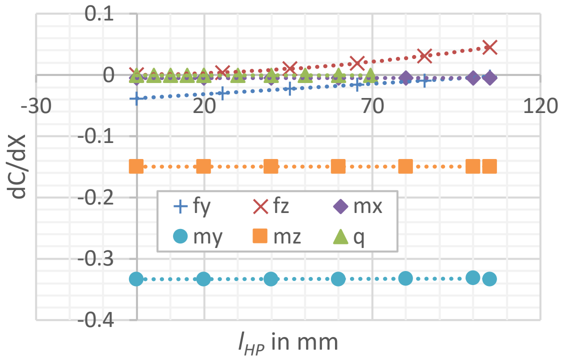

Figure 5. Variation of sensitivity of the stiffness to parasitic forces f_y, $f_z = 0{,}1$ N , parasitic torques $m_x, m_y, m_z = 10$ Nmm and heat $\dot{q} = 10$ W due to the application point l_{HP} (linearized values)

Based on the sensitivities on the main pivot H, new limit values for the disturbances during the adjustment are derived. These are presented in Table 2. A maximum deviation of the stiffness of 0,1 mN/m is used to set the limit values. Whereas the allowable forces in the x- and z-directions can be significantly increased, allowable forces in the y-direction are highly critical.

Table 2 Disturbance sensitivities of the weighing cell and limit values on the adjustment location K and main pivot H

X	Disturbance sensitivity			Limit value		
	point K	point H	unit	point K	point H	unit
fx	2,8E+01	4,6E-03	N/m/N	3,5E-06	2,2E-02	N
fy	2,6E-03	3,8E-02	N/m/N	3,9E-02	2,6E-03	N
fz	3,0E-03	1,9E-05	N/m/N	3,3E-02	5,4E+00	N
mx	5,0E-03	5,0E-03	N/m/Nm	2,0E-02	2,0E-02	Nm
my	2,2E-02	2,2E-02	N/m/Nm	4,6E-03	4,6E-03	Nm
mz	4,3E-01	4,3E-01	N/m/Nm	2,3E-04	2,3E-04	Nm
q	1,1E-02	1,1E-02	N/m/W	9,0E-03	9,0E-03	W

4. Design of in-situ adjustment system

A novel concept for an in-situ adjustment system is developed based on the knowledge gained from the investigation in Section 3. The main idea of the system design is the reduction of the influence of parasitic loads during the adjustment by purposely locating them in the least sensitive spot on the balance beam. Figure 6 shows an scheme of the concept system.

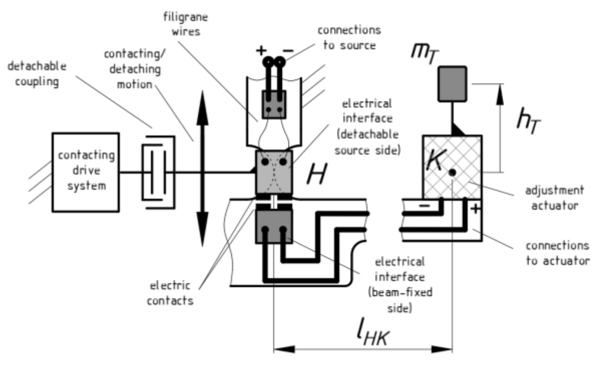

Figure 6. Schematic representation of concept for the fine adjustment system.

Since critical mechanical loads are mainly introduced through the electric connections for energy transfer to the adjustment actuator built on the balance beam, these are laid out as filigrane wires and short-circuitted through the main pivot H. Gold-plated copper wires with a diameter of 50 µm are used due to their very low stiffness. To minimize the number of wires, a drive element with two lines is selected. The adjustment actuator stays on point K and energy is transferred from point H through electrical connections fixed on the balance beam. A detachable electrical interface is required to eliminate the mechanical coupling between the source and the balance beam after the adjustment. A separate rack-mounted drive system is used to realize the contacting and detaching motion.

Particular focus lies on the design of the detachable electrical interface. The interface must enable an electric connection between the electric cables of both sides with least electrical resistance. The weight of the source side also is the main source of parasitic mechanical loads during the adjustment, particularly f_y. A change in stiffness is produced after it is detached, leading to a different adjusted state. Thus, the mass of the source side must be as low as possible and the centre of mass must accurately coincide with the centre of rotation of the balance beam (point H). In addition, a mechanical interface is required for its manipulation with the contacting drive system.

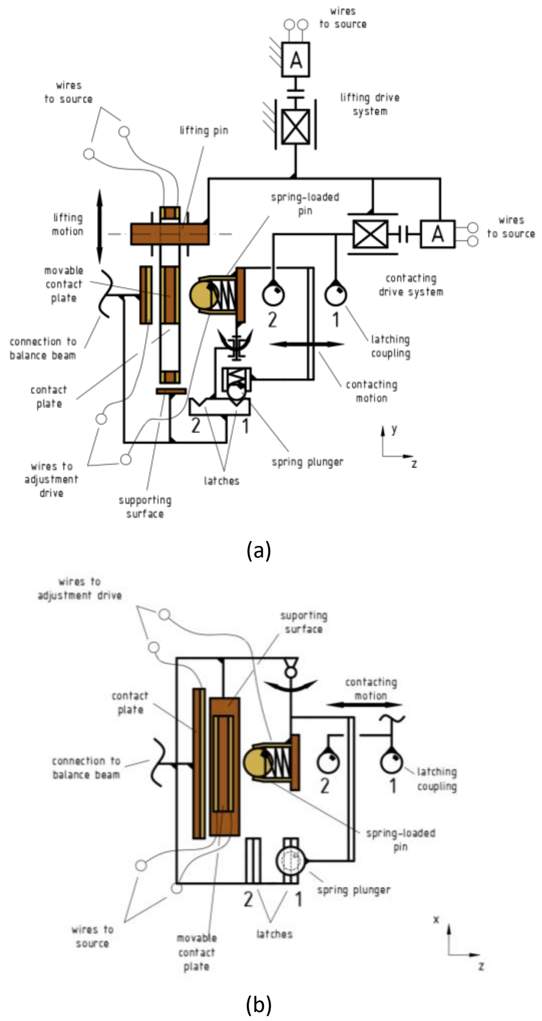

Figure 7. Functioning principle of detachable electrical interface: (a) in yz-plane, (b) in xz-plane. Gold color indicate conducting elements, brown color indicate insulating elements. Dimensions are not proportional.

Figure 7 shows the functioning principle of the detachable electrical interface. A movable contact plate based on a double-sided gold plated substrate is transported in the y-direction (gravity vector direction) using an lifting mechanism. The mechanical coupling is done via a form contact pairing with an insulated pin on the lifting mechanism. Clearance is necessary to allow self-aligment of the contact plate. Stops on the pin avoid excessive lateral motions. A supporting flat surface on the beam side of the interface defines the position of the centre of mass in y-direction to avoid the stiffness variation. The contact plate is designed symmetrical for a better definition of its centre of mass, see Figure 8. When the contact plate touches the locating surface, mechanical coupling with the lifting mechanism is interrupted. Thus, no forces in y-direction are transmitted. The pin stays in its position without mechanical contact until the adjustment is done and the contact plate can be retired.

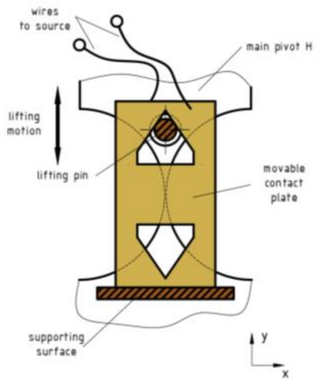

Figure 8. Positioning of contact element relative to main pivot H in xy-plane. View during contact with supporting surface without beam-fixed contact plate.

To mechanically secure the contact plate and ensure the electrical connection, a contacting drive system mounted on the lifting mechanism pushes the movable plate against the contact plate fixed on the balance beam. The contacting mechanism mounted on the balance beam composes of a latching lever with two secured positions: open (1) and closed (2). On the lever there is a spring-loaded electrical pin which is used to push the movable contact plate. The spring-loaded pin forms with the double-sided contact plate the other electrical connection with the adjustment actuator. The motion of the lever from position 1 to 2 and viceversa by introducing external forces in the stiff z-direction via ball-plane contact pairings with the contacting mechanism. Actuation of the contacting mechanism in this direction allows for higher forces to be used for establishing the electrical connection. The latching coupling can move in one direction until a position (1 or 2) is secured, then move in the other direction to eliminate the contact. To avoid any influence on the stiffness, the position of the centre of mass of all components of the beam side of the interface in y-direction does not change between position 1 and 2.

A two line linear actuator (built on the balance beam) is required for producing the in-situ adjustment motion. The actuator must be able to maintain the adjusted position after the energy supply has been interrupted as well as to limit heating nor magnetic fields. Piezoelectric actuators store the electrical energy while also convert it into deformation. They can hold the adjusted position for relatively long periods of time. In addition, in quasistatic applications, current requirements and heat losses are very low. A vacuum-compatible piezoelectric stack actuator from Physik-Instrumente GmbH [7] was used in a preliminary investigation on the unconstrained weighing cell. Due to its high resolution, a theoretical resolution of the stiffness adjustment better than 0,1mN/m is possible with a 100g trim mass. After separating the electrical contact, the adjusted position varies over time slightly due to the self-discharge of the actuator. This voltage loss is also problematic for readjustments due to the difference between the voltage of the source and the actuator, which produce current peaks during coupling. These problems could be avoided using long-term stable piezoelectric actuators, which maintain their position at 0V [8].

5. Conclusions and outlook

This contribution presents a novel concept for an in-situ fine adjustment systems on highly sensitive movable elements of weighing cells inside hermetically sealed vacuum chambers. The system is designed to produce minimal disturbances during the adjustment process and completely avoid them afterwards. Thus, a highly accurate and stable adjustment can be produced. The adjustment can be achieved without arresting the weighing cell, further increasing its performance.

Disturbances by the adjustment system are mainly introduced by the electric cables for energy transfer to the actuator built on the balance beam. These are designed as detachable filigrane wires purposely connected to the frame through the least sensitive location of the balance beam. The electric interface is securely connected and detached via a contacting system which introduces forces only in the stiff z-direction of the weighing cell. The only significant parasitic effect to stiffness adjustment is due to the momentum of movable contact plate, which is kept minimal and its application point on the centre of rotation with its embodiment design. To minimize heat generation, while simultaneously maintaining the adjusted position, a built-in high-resolution piezoelectric actuator is used for the in-situ adjustment. A theoretical stiffness adjustment resolution better than 0,1 mN/m is possible with the aforementioned system.

Experimental validation of the system represent the ongoing work. In addition, further geometry and material optimization of the electric interface is required to reduce the weight and to ensure a secure electric contact with low disturbances.

Acknowledgments

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