Concept of a monolithic stiffness-compensated mechanism for high-resolution force sensors

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Abstract

Monolithic compliant mechanisms with concentrated compliances are often used in high-resolution force sensors and precision balances. Since the measurement resolution is vastly limited by the bending stiffness of the compliant joints, the thinnest part of the joints is reduced to down to 50 µm. A further reduction encounters technological limitations and creates new side effects. Compensation for the "positive" stiffness of the mechanism can be achieved by integrating an element with "negative" stiffness that generates a counteracting force or torque when deflected. In the literature, preloaded tension springs, buckled leaf springs as well as trim masses are predominantly for that purpose. However, most existing approaches are either not monolithic, elaborate to readjust, associated with parasitic forces and torques, or only applicable in a defined orientation relative to the vector of gravity.
This paper presents a new concept of monolithic stiffness-compensated mechanisms for use in high-resolution force sensors that is independent of spatial orientation. The negative stiffness is generated by a preloaded spring element of an integrated compensation mechanism. The preload force can be easily adjusted. The compensation force is generated simultaneously with the deflection and transmitted to the main mechanism by a lever and a dedicated coupling element to avoid parasitic effects as much as possible. A suitable design minimizes parasitic motions and avoids buckling of the thin joints as a result of the relatively high preloading force. Finite element simulations are performed to investigate the behavior of the mechanism and to validate the concept.

1 Introduction

Precision applications in force measurement technologies require highest resolution over relatively large measuring ranges, highest reproducibility, as well as traceability to a natural constant [1-3]. In high-resolution force sensors and precision balances, monolithic compliant mechanisms with flexure hinges are used as the main mechanical structure due to their many advantageous properties. Since the achievable measurement resolution is limited by the inherent stiffness of the mechanism [4], the thinnest part of the flexure hinges is usually reduced down to 50 µm. However, further downsizing cannot be realized for technological limitations and the increasing influence of disturbances [5]. Thin flexure hinges require relatively large lateral dimensions, hindering the stiffness reduction as well as the modeling accuracy [6]. To compensate for the remaining "positive" stiffness of the mechanism, an element with "negative" stiffness has to be integrated. This paper presents a novel concept of a monolithic stiffness-compensated mechanism for use in high-resolution force sensors.

2 State of the art

A force is determined by measuring its impact. Load cells convert the impact of a force either directly, indirectly, or via the compensation principle into an evaluable electrical signal. Limiting factors for the resolution and measuring range of the load cells include the system parameters of the mechano-electrical transducer and, depending on the working principle, the stiffness of the compliant mechanism as well as the specifications of the actuator.

2.1 High-precision load cells

The most precise load cells are used for mass determination at National Measurement Institutes. Weighing cells of precision balances and mass comparators can achieve sub-nanometric resolution over a measuring range of up to 20 mN [7]. These load cells are generally based on a beam balance and convert the impact of the force indirectly according to the principle of electromagnetic force compensation (EMFC) (**Figure 1**).

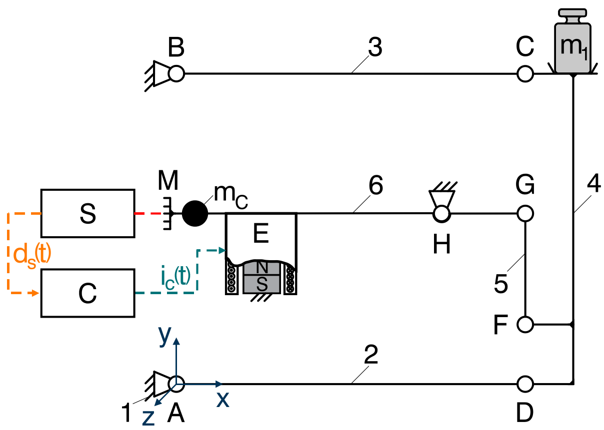

Figure 1 Working principle of a weighing cell with electromagnetic force compensation.

Weighing cells are mainly designed as monolithic mechanisms with flexure hinges due to their high motion accuracy and repeatability. The kinematic structure is com-

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posed of a parallelogram guide (2-4) and a pivoted transmission lever (6), linked by a coupling element (5). The position sensor (S) detects the deflection of the transmission lever produced by the weight force of the mass to be measured (m_l) at the measuring point M. A compensation force generated by the electromagnet (E) keeps the lever in equilibrium. The control unit (C) regulates the required electric current (i_C), which is used as a measure for the mass (m_l). A counterbalance mass (m_C) on the lever reduces the effective force on the electromagnet (E). As such, the ratio of the counterbalance mass (m_C) and the mass to be measured (m_l) is crucial for the resolution and the measuring range of the system.

2.2 Stiffness compensation

The force measurement resolution is limited by the stiffness of the compliant mechanism. As such, the thinnest part of the flexure hinges is typically reduced down to the technological limit of 50 µm. The highest possible resolution requires further stiffness reduction down to almost zero. To compensate for the stiffness of the compliant mechanism, an element that generates a counteracting effect to the restoring force or torque needs to be integrated. In positioning mechanisms, preloaded tension springs [8] and buckled leaf springs [9] are predominantly used. In addition to compensation by springs elements [10], trim masses [11] are also applied in force and mass measurement technologies.

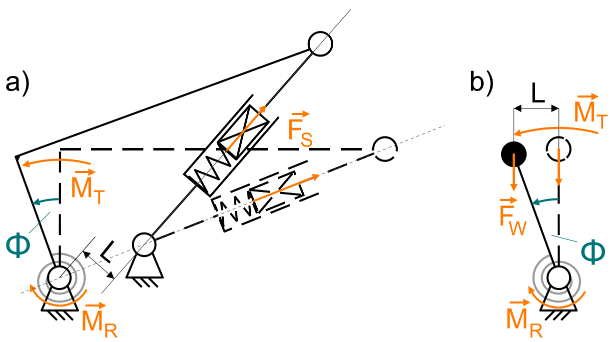

Figure 2 Stiffness compensation of a single flexure hinge by: a) preloaded spring. b) trim mass.

The existing principles for stiffness compensation are exemplified on a single flexure hinge (**Figure 2**). In the case of compensation with springs (**Figure 2a**), the mechanism is designed in such a way that the lever arm (L) of the preload force (F_S) increases with the deflection. This creates a torque ($F_S \cdot$ L) that counteracts the restoring torque (M_R) of the flexure hinge so that the required total or actuation torque (M_T) is reduced (**Equation (1)**). In the non-deflected position, the line of action of the preload force (F_S) goes through the rotation axis of the flexure hinge, i.e. L(0) = 0, keeping the system in static equilibrium.

$$M_T\,(\Phi) = M_R\,(\Phi) - F_S\,(\Phi) \cdot L\,(\Phi) \rightarrow 0 \qquad (1)$$

The stiffness compensation using trim masses (**Figure 2b**) works similarly, with the weight force (F_W) of the trim mass generating the counteracting torque ($F_W \cdot$ L) when deflected (**Equation (2)**).

$$M_T\,(\Phi) = M_R\,(\Phi) - F_W \cdot L\,(\Phi) \rightarrow 0 \qquad (2)$$

However, the preload force of the spring, as well as the weight of the trim masses, also represent a parasitic force on the flexure hinge, since only a component is required to produce the compensation torque, e.g. ($F_W \cdot \sin\Phi$). In the non-deflected state, these forces are purely parasitic.

3 General concept

The developed concept of a stiffness-compensated mechanism (**Figure 3**) is based on the structure of an EMFC weighing cell and can achieve outstanding force measurement resolution. In comparison to the trim masses on a weighing cell, the functionality of the stiffness compensation is realized by a preloaded tension spring and, thus, is widely unaffected by the orientation in the gravity field.

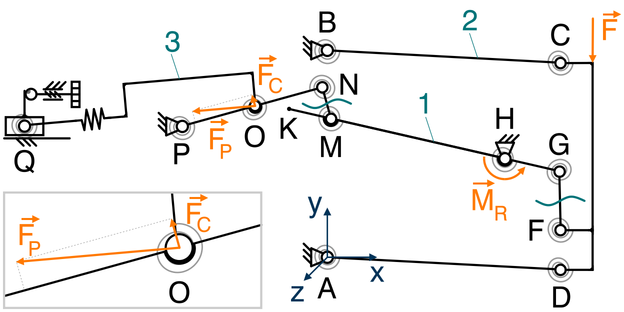

Figure 3 Rigid-body model of the mechanism.

3.1 Transmission lever

The concept of the mechanism is based on the beam balance. The pivoted transmission lever (1) serves as the weighing beam and is supposed to be kept in balance. The force (F) to be measured is transmitted to the lever via a coupling element at point G. The distance sensor and the actuator for electromagnetic compensation are positioned at point K to achieve the maximum resolution possible. A position control ensures a torque equilibrium at the transmission lever by selectively energizing the actuator according to the principle of an EMFC weighing cell.

3.2 Guiding mechanism

To enable translational force initiation, the concept provides a simple guiding mechanism (2), also following the model of an EMFC weighing cell. A parallel crank is used to realize a linear motion. Inherent lateral deviations of the coupler motion, as well as parasitic tilting caused by manufacturing deviations, are negligible. A coupling element connects the guiding mechanism to the transmission lever using the revolute joints F and G.

3.3 Compensation mechanism

To compensate for the restoring torque (M_R) caused by the compliant joints, a stiffness adjustment mechanism (3) is integrated. It uses a preloaded spring in combination with a lever element to generate a counteracting torque on the main mechanism. The spring element, exemplary conceived as tension spring, is preloaded up to the desired value by moving the slide joint at Q. When the transmission lever (1) is deflected by initiating a force at the contact point of the guiding mechanism (2), a component (F_C) of the preload force vector (F_P) is generated. The force (F_C) is transmitted to the transmission lever at point M by the lever NP and coupling element MN. An increasing deflection leads to an increasing lever arm and, thus, counteracting force, which compensates for the restoring torque. Through the use of the lever NP and coupling element MN, the remaining parasitic component of the preload force, especially in the non-deflected state, is decoupled from the main mechanism and is mainly supported by joint P.

4 Design of the mechanism

The mechanism is designed completely monolithic by replacing the revolute joints of the rigid-body model with flexure hinges and the tension spring with a leaf spring compound (**Figure 4**). Compared to conventional mechanisms with form-fit joints, monolithic mechanisms have the advantage of being free of clearance and wear, with negligible molecular friction occurring. Additionally, the lack of mechanical interfaces in the system, the elimination of error sources due to assembly and adjustment, and the high accuracy of the relative position of the joints enable the most precise and repeatable motion behavior.

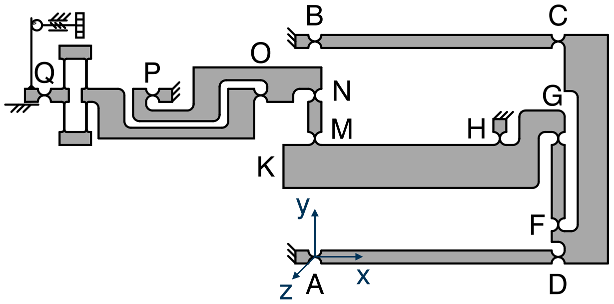

Figure 4 Embodiment design of the mechanism.

Flexure hinges with semi-circular contour meet the requirements of low complexity and motion behavior of a sheer rotation in the best way. The leaf spring compound used to replace the tension spring has the advantages of lower dead mass, high lateral stiffness, and a significantly lower tolerance range of the spring ratio. A symmetric dual configuration additionally eliminates parasitic lateral displacements. The embodiment design also avoids buckling due to high preload forces by stressing the thin joints on tension. Parasitic motions are minimized as well.

5 Simulation results

To validate the concept, the compliant mechanism is investigated by means of a 3D finite element model (**Figure 5**) in ANSYS Workbench. A mesh refinement was undertaken only on the flexure hinges to ensure the high accuracy of the results while minimizing computation time.

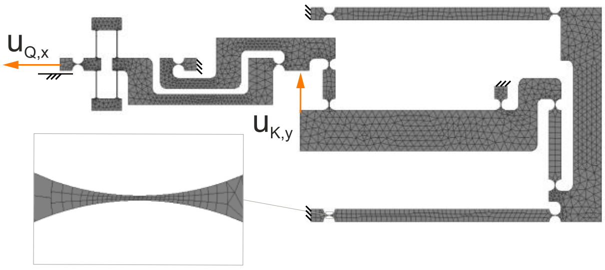

Figure 5 Finite element model of the stiffness-compensated compliant mechanism with boundary conditions

The geometrical parameters (**Table 1**) were prudently defined after an initial design iteration. The stiffnesses of the flexure hinges C_φ and the tension spring element C_f are determined using independent finite element models.

Parameter	Value	Parameter	Value
C_φ	18.03 Nmm/rad	C_f	7.78 N/mm
AD, BC	112.5 mm	HK	105 mm
AB, CD	100 mm	HM	85.5 mm
DF	15 mm	MN	10 mm
FG	40 mm	NO	25 mm
HG	27 mm	OP, OQ_{ini}	50 mm

Table 1 Parameters of the mechanism.

The stiffness characteristic curve of the mechanism at point K is investigated for different preloading positions $u_{Q,x}$. A setting resolution of 1 µm is assumed for realizing the preloading. The simulation results (**Figure 6**) show a reduction of the stiffness to about 0.2% of the initial value. The average stiffness $C = (F_{K,y}(u_{K,y}) - F_{K,y}(0))/u_{K,y}$ without the compensation sub-mechanism (**Figure 1**) for a deflection of $u_{K,y} = 0.1$ mm amounts to 3.65 N/m.

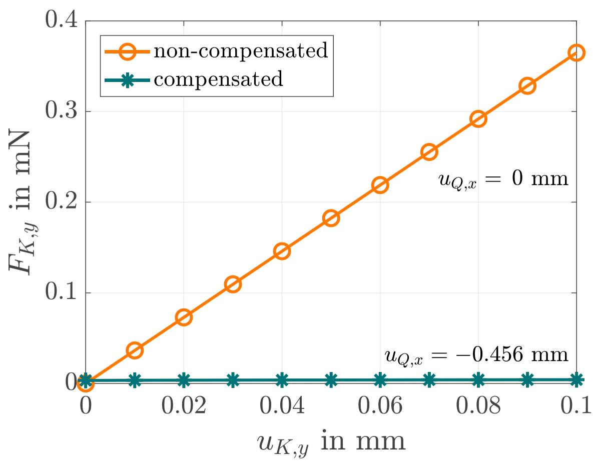

Figure 6 Stiffness characteristic curves of the mechanism with and without compensation.

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Without preloading, the total stiffness of the structure adds to 10.59 N/m. A preloading of $u_{Q,x}$ = - 0.302 mm is necessary to eliminate the additional stiffness of the compensation sub-mechanism. The preloading introduces a small parasitic force at $u_{K,y}$ = 0 mm (**Figure 7**), whose effect can be eliminated by subtracting it from the force value at $u_{K,y}$ = 0.1 mm. The minimum average stiffness with the current configuration of approximately 0.008 N/m is achieved with preloading of $u_{Q,x}$ = - 0.456 mm. Assuming a resolution of 1 nm of the position sensor, a theoretical force measurement resolution of approximately 31 pN can be achieved.

However, around zero average stiffness, the characteristic curve behaves nonlinearly (**Figure 7**). The nonlinearity causes the theoretical force resolution for $u_{K,y}$ = 0.1 mm to differ from its value for $u_{K,y}$ = 1 nm. For $u_{Q,x}$ = - 0.456 mm, the achievable force resolution is lower (> 31 pN) due to the degressive behavior of the stiffness. A verification for $u_{K,y}$ = 1 nm using the current finite element model is not possible due to numerical errors.

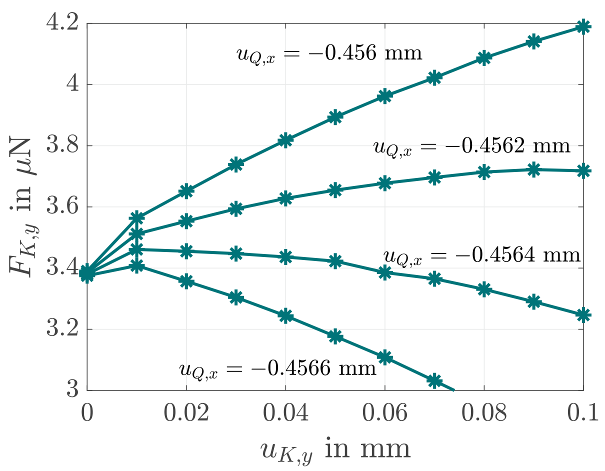

Figure 7 Close-up of the Stiffness characteristic curves around zero stiffness.

To achieve a higher measurement resolution, a higher resolution of the stiffness adjustment is required. This can be either realized by increasing the preloading resolution, reducing the stiffness of the tension spring element, or optimizing the elasto-kinematic design of the compensation mechanism.

6 Summary

This paper presents a concept of a monolithic stiffness-compensated mechanism for high-resolution force sensors. The working principle combines the advantages of an EMFC weighing cell and stiffness compensation by preloaded springs while having minimum side effects. The concept is validated using finite element models. With an adjustment resolution of 1 µm, the current configuration reduces the stiffness of the mechanism to about 0.2% of the initial state.

Future research will include the optimization of the mechanism to reduce the stiffness further, the influence of manufacturing tolerances on the required adjustment range, and investigations on the system's behavior with masses of the links enabled. In this context, the influence of the weight forces on the overall compensation concept as well as the change of orientation of the application in the earth's gravitational field will be considered. Experiments will be carried out to verify the theoretical investigations and simulation results.

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