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Contribution to the mechanical enhancement of load cells in precision weighing technology by means of advanced adjustment strategies

Maximilian Darnieder[1], Rafael R. Marangoni[2], René Theska[1], Thomas Fröhlich[2], Ilko Rahneberg[2]

Technische Universität Ilmenau, Department of Mechanical Engineering
[1]Institute of Design and Precision Engineering, Precision Engineering Group
[2]Institute of Process Measurement and Sensor Technology, Process Measurement Group

maximilian.darnieder@tu-ilmenau.de

Abstract
The accuracy of force measurement systems is predominantly influenced by its stiffness towards deflection. Monolithic mechanical systems in precision weighing technology rely on ultrathin flexure hinges. Further stiffness reduction by a decrease of the minimum hinge thickness is unfavourable. Consequently, the present concept relies on compensation rather than a reduction of the stiffness. Based on precise adjustments, the system's state is altered towards an astatic state. Hereby, the overall stiffness and the tilt sensitivity is reduced. These properties have been determined as major contributions to the measurement error. The results of the theoretical investigations form a basis for future experiments and a further improvement of load cells.

Keywords: load cell, stiffness compensation, astatic system, precision mass measurement, weighing technology, mass comparator

1. Introduction

Weighing technology represents an omnipresent and inevitable part of the global economy. Despite its importance and the outstanding advances in technology in the past decades, the unit kilogram is still not based on a satisfactory definition, [1]. In this context, highly precise mass comparators are of great importance, both for the dissemination of the present definition and for scientific efforts aiming at a redefinition of the kilogram, [2]. State of the art load cells combine a monolithic structure with an electro-magnetic force compensation (EMFC), [3]. This relatively recent concept replaces equal arm beam balances, which have been studied extensively in the past, [4]. The complex mechatronic system has been calculated numerically in [5]. There is consensus about restrictive factors limiting the precision of mass determinations like the stiffness of the mechanical structure, ground tilt, ground vibrations and none centric loads.
The objective outlined in the present paper is a mechanical enhancement of load cells to minimize the effects of the mentioned factors on the measurement. This includes the reduction of the stiffness of the structure and the minimization of its response to ground tilt. Stiffness reduction by means of thinner compliant joints is a measure that is already applied extensively. A further reduction of the minimum hinge thickness is accompanied by several disadvantages like a reduced load capacity of the single joint, increasing parasite deformations and rising manufacturing costs.

An alternative approach constitutes a principle that is applied in seismological instruments, [6]. The geometry of the kinematic structures is designed to be in close proximity to an astatic system-state over the entire deflection range. This in turn equals a high sensitivity to the measured value. For the load cell, this principle is realized by raising the centre of gravity (CG) above the centre of rotation. This leads to a counter-torque compensating the residual restoring torque of the compliant joints, [7].

2. Methodology

The kinematic structure of the load cell is examined in detail throughout several modelling stages. A simplified *analytical model* based on Lagrange equations of second kind constitutes the starting point of the theoretical investigation.

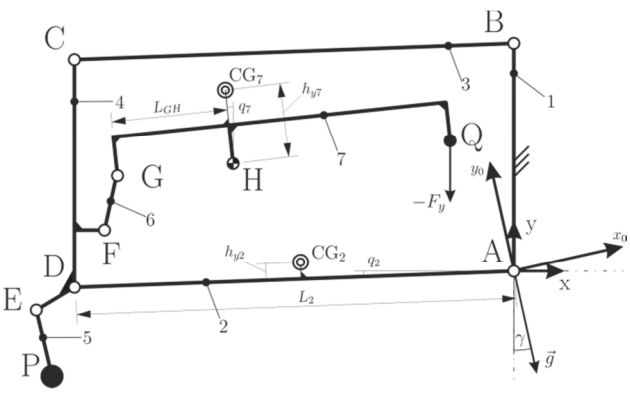

Figure 1. Basic mechanical structure of an EMFC load cell. The joints at points A to D and F to H are modelled with constant torsion spring stiffness.

In order to consider nonlinear effects, the kinematic structure is modelled as a *multi-body system* (MBS). The most detailed modelling stage constitutes a *structural finite element* (FE) *model*. The results from the three modelling stages are compared and conclusions for the enhancement of load cells by means of advanced adjustment strategies are drawn.

3. Results and discussion of the analytic model

Reduced to a system of rigid bodies, the load cell's kinematic structure in the xy-plane can be depicted as shown in Figure 1.

The compliant joints are modelled as pivots with a constant stiffness and without friction. Their torsional stiffness is determined based on the geometry of a sample flexure hinge that is used throughout the models. The stiffness amounts to $c_{rot} = 22.7e^{-3}$ Nm/rad. The location of the CG's and the masses itself are the variables of the models. For the formulation of the static behaviour of the system it is fallen back on *Lagrange's equations of second kind*. The general equation can be simplified, since only conservative forces are present and all time derivatives equal zero. With $L := L_{GH}/L_2$ follows:

$$C = m_2 g h_{y2} L^2 + m_7 g h_{y7} - 2c_{rot} - 4c_{rot} L^2 \quad (2)$$

$$D = -m_2 g h_{y2} L - m_7 g h_{y7} \quad (3)$$

Once this simplified model is linearized, the resulting equations (2-3) provide a comprehensive picture of the behaviour of the load cell. Clearly, nonlinear effects are excluded from the consideration, but they can be expected to vanish due to the marginal deformation of the system. To keep the relations as simple as possible, the angle of part 6 in Figure 1 is not considered. The resulting static equation can be divided into terms dependent on the deflection of the system (2) and those sensitive to the tilt angle γ (3).

The stiffness C and the tilt sensitivity D represent the properties of the weighting cell this study focuses on. The objective is a minimization of both unfavourable properties. The theoretical results show that this condition can be reached by a shift of the CG's:

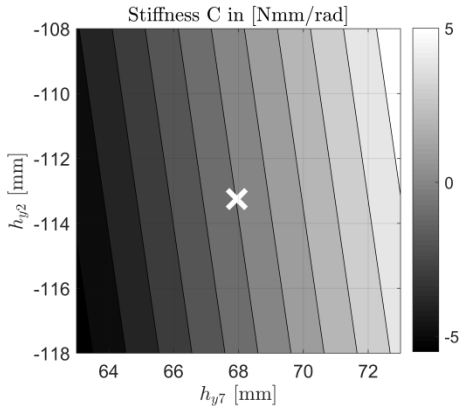

Figure 2. The stiffness of the structure C as a function of the y-displacement of the lever masses. The optimal configuration is marked by the white cross.

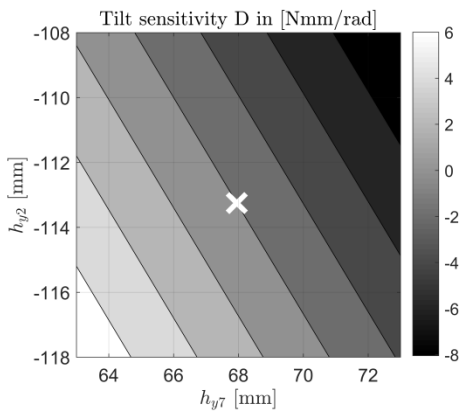

Figure 3. The distance of the CG's to their respective centres of rotation can be employed to minimize the tilt sensitivity D.

Consequently, the optimum for the load cell is the common zero-crossing of the C-plane and the D-plane as depicted in Figures 2 and 3 (white cross). It is found that the prerequisite for the presence of the common zero-crossing is that the levers L_{GH} and L_2 are not equal.

4. Multibody- and Finite Element model

The analytic model confirms the theoretical viability of the proposed method for the enhancement of the sensitivity of the load cell based on simplifying assumptions. To check the applicability of these model assumptions, first a multibody simulation (MBS) and second a parametric FE-model is developed. Whereas the MBS is based on concentrated masses in the CG's and torsion spring stiffness, the FE-model is capable of computing the actual geometry of the load cell. In a parameter study, the locations of the CG's are varied as in the analytic model and the results are generally in agreement, see Table 1.

Table 1. Comparison of the resulting optimal configurations of the load cell for the different models with the resulting adjustment sensitivities. The masses of the components are $m_7 = 0.10$ kg and $m_2 = 0.25$ kg and the lever ratio L_{GH}/L_2 amounts to 0.24.

	analytic	**MBS**	**FE**
$\mathbf{h_{y7}}$ [mm]	67.88	67.88	68.86
$\mathbf{h_{y2}}$ [mm]	-113.08	-113.08	-112.09
$\Delta\boldsymbol{C}/\Delta\boldsymbol{h_{y7}}$ [Nmm/rad mm^{-1}]	0.981	0.920	0.832
$\Delta\boldsymbol{D}/\Delta\boldsymbol{h_{y2}}$ [Nmm/rad mm^{-1}]	0.589	0.621	0.657

The comparison reveals that analytic model and MBS are in good agreement. The MBS proves to be slightly stiffer than the analytic model. The deviation of the optimal configuration of the FE model might be justified by an additional stiffening of the compliant joints by the vertical load of the structure.

4. Summary and future work

The theoretical investigations outlined in this paper reveal that the commonly used structures of present load cells can be adjusted to an optimal configuration where the sensitivity to ground tilt and the stiffness of the load cell vanish. The residual amount of C and D depends on the step size of the grid, thus on the accuracy of the adjustment of the CG's. The theoretical model requires experimental verification in future work as well as a dynamic consideration of the load cell in view of ground vibrations.

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