IMEKO 2010 TC3, TC5 and TC22 Conferences
Metrology in Modern Context
November 22–25, 2010, Pattaya, Chonburi, Thailand

# PARAMETRIC MECHATRONIC MODEL OF A LOAD CELL WITH ELECTROMAGNETIC FORCE COMPENSATION

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**Abstract** – In this paper a mechatronic model for an electromagnetic-force-compensated (EMC) load cell is presented. Designed in MATLAB® Simulink® the model consists of four main modules: a rigid-body model delineating the mechanical behaviour of the load cell, an analytical model characterising the electrodynamic actuator consisting of a voice coil and a permanent magnet, a characteristic curve modelling the position detector, and a continuous or discrete model describing the operation of the controller.

Optimization of the mechanical, electromagnetic and controller components can be performed using this model; furthermore experiments can be performed to determine the sensitivity of the complete system to changes of defined parameters.

**Keywords:** Simulation of complex mechatronic systems, Optimisation of parameters, force measurement

## 1. INTRODUCTION

Concerning uncertainty of measurement and achievable resolution, balances based on the principle of electromagnetic force compensation represent the state of the art. The schematic setup of such a balance and its components is depicted in fig. 1.

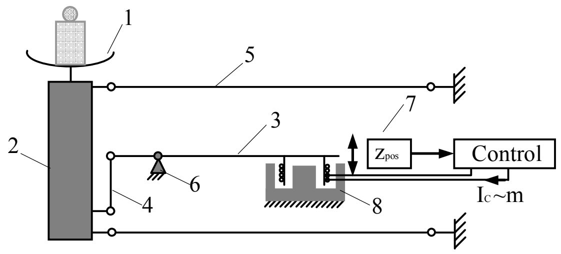


Fig. 1: Schematic setup of an EMC balance

1 – Weighing pan
2 – Load suspension
3 – Transmission lever
4 – Coupling member with flexure hinges
5 – Parallel lever system with flexure hinges
6 – Bearing of the transmission lever
7 – Position detector
8 – Electrodynamic actuator

When a weight is applied to the weighing pan (1), it lowers, guided by the parallel lever system (5). The movement is then transferred by the transmission lever (3) with a specific ratio. A position sensor (7) detects the displacement of the transmission lever. The resulting position signal is used as the input for a controller, which outputs a current to the coil (8). The resulting Lorentz force countervails the weightforce. The current in the steady state (lever returned to the zero position) can then be used as an accurate measure for the mass/force to be determined.

Due to many applications of these systems in industry and research, there is a strong interest in improving the performance in terms of speed and accuracy. Hence there is need for accurate models, describing the behaviour of the complex system to be optimized.

As a component of the industrial design process, finite element method (FEM) models are frequently used to investigate the behaviour of selected mechanical parts of a balance or to perform modal analyses. The mechanical behaviour of the system can be simulated very well. Currently, FEM models are bound to fixed geometries and cannot be set up parametrically. To investigate the influence of damping on the system dynamics as well as the interaction of mechanical, electromagnetic and controller components, a transient FEM-model is needed. This type of model is extremely memory and computing time intensive, hence impractical and rarely used.

In contrast to FEM models, analytical models can be set up completely parametrically. Theoretically, analytical models are completely invertible; hence every parameter could be optimized individually. The drawback of all these models is that if all relevant mechanical degrees of freedom for a balance are taken into account, the resulting differential equation is highly nonlinear. By coupling this differential equation to the equations for the electrodynamic actuator and the controller, the resulting equation has no closed analytical solution. In order to nevertheless derive a solution, the model is strongly abstracted (as in [1] and [2], with different levels of abstraction). The resulting model is a very rough approximation of the real system; thus it is not a good basis for optimization.

A tradeoff between FEM and analytical models can be found with rigid-body models. In [3] a very complex model is discussed which fits well with actual measurements. However, since only the mechanical part is modelled, the closed loop consisting of mechanics, electrodynamic actuator and controller cannot be investigated.

## 2. MECHATRONIC MODEL

In this paper a parametrical numeric model describing the complete system (See fig. 2) is presented.

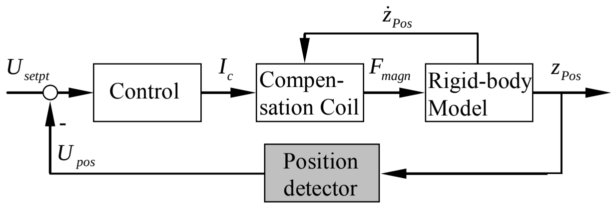


Fig. 2: Schematic setup of the complete model

The model consists of four main components: The controller, putting out a current to the electrodynamic actuator, resulting in force which acts upon the rigid-body-model. The motion of the transmission lever is then transformed into a position signal, which is fed back to the controller. The position detector can be well modelled by a PT1 element. As its time constant is very small and does not affect the bandwidth of the complete system, the model of the position detector was reduced to a static characteristic curve, correlating the mechanical position of the lever to the electrical position signal.

In the following paragraphs, the remaining three components will be described briefly.

### *2.1 Rigid-body model*

Rigid-body models consist of single bodies coupled by joints. Spring and damper constants can be assigned to these joints.

The flexure hinges of the lever and the parallel lever system can be modelled as torsion springs. Their spring ratio can be computed as described in [4]. To reduce the number of degrees of freedom, the flexure hinges with a factor of stiff to weak spring constant lager than 1e5 were assumed to have just one degree of freedom. In this case resilience can be neglected.

Rigid-body models are based on the assumption that all bodies under consideration are ideally stiff. This is an approximation that does not fit all parts of an EMC balance. Some mechanical parts of a weighing cell can be considered stiff due to their compact dimensions. The dimensions of the parallel lever system and the transmission lever are long and thin; when loads are applied, a strain is noticeable. Therefore, the parts of the parallel lever system cannot be assumed to be rigid bodies. The translation resulting from strain can be modelled as extension springs. A spring constant is assigned which is proportional to the strain of each of the bodies of the parallel lever system.

The masses and the moment-of-inertia matrices of each body have to be applied at the centre of gravity of the equivalent body. To obtain this information, the construction is virtually cut at the joints into geometries of less complex shape. Mass, centre of gravity and the corresponding matrix of moment-of-inertia can then easily be obtained. A schematic set-up of the rigid-body model is depicted in fig. 3.

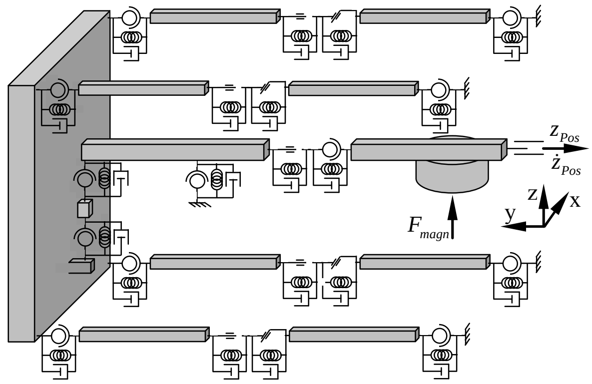


Fig. 3 Schematic set-up of the rigid-body model

Modelling the damping is more complex because there are three main sources: the magnetic system, inner friction and friction due to the movement in air. Damping due to self-induction in the moving coil is already accounted for in the analytical model of the magnetic system and does not need to be considered in the rigid-body model.

Damping induced by inner friction and the friction due to movement in air is difficult to determine. Because of this, it is estimated experimentally and fitted to the model. The system is deflected from the zero position and then allowed to settle again. From the decay process it is possible to obtain the damping ratio consisting of damping due to inner friction and damping caused by air friction. Since both effects are minor in size compared to the damping caused by the magnet system, merging both effects and applying the resulting factor to the joints causes negligible errors.

### *2.2 Analytical model of the magnetic system*

The electrodynamic actuator consists of two parts: the permanent magnet fixed to the system mounting (see fig. 1) and the coil connected to the transmission lever and moving in the magnetic field of the permanent magnet. The magnetic field can be fitted well by a parabolic function of the z-coordinate.

The magnetic system supplies the necessary force, induced by the current provided by the controller, to maintain the balance in the zero position. The acting Lorentz force can be calculated as:

$$F_{magn} = I_C \cdot l \times B(z) = I_C \cdot 2 \cdot \pi \cdot n \cdot r_{coil} \times B(z) \qquad (1)$$

With $F_{magn}$ – Lorentz force
$I_C$ – Current trough coil
l – Length of copper winding of coil
B(z) – Magnetic field depending on z-coordinate
n – Number of windings
$r_{coil}$ – Radius of the coil

Due to the movement of the coil in the magnetic field, a voltage $U_{ind}$ is induced (2):

$$U_{ind} = -z_{Pos} \cdot s \cdot l \times B(z) \qquad (2)$$

$U_{ind}$ is the feedback of the mechanical behaviour – expressed through $z_{pos}\cdot s$ which is the Laplace transformed velocity of the coil in the magnetic field – to the electrical characteristic of the coil. The behaviour of the coil can be described by its ohmic resistance and its impedance. We use the following network of ohmic resistances and inductances as shown in fig. 4.

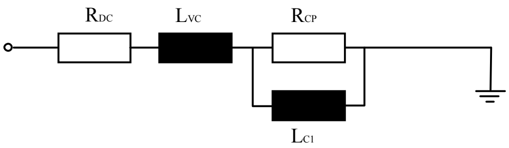


Fig. 4 Network of resistances and inductances describing the behaviour of the coil

The feedback of mechanical behaviour with the electric behaviour can be merged to eq. (3):

$$U = \frac{I_C}{R_{DC} + s\cdot L_{vc} + \dfrac{1}{\dfrac{1}{R_{CP}} + \dfrac{1}{s\cdot L_{C1}}}} - U_{ind}$$

$$U = \frac{I_C}{R_{DC} + s\cdot L_{vc} + \dfrac{1}{\dfrac{1}{R_{CP}} + \dfrac{1}{s\cdot L_{C1}}}} + z_{pos}\cdot s\cdot l \times B(z) \qquad (3)$$

For a less complex nomenclature, the equations (2) and (3) are denoted in Laplace domain.

### *2.3 Discrete model of the digital controller*

A controller is necessary for the operation of an EMC balance. In this paper a digital controller is used which has the form of a discrete PIDT1 controller. The transfer function of this controller is given by equation (4)

$$G_{PIDT1}(z) = k_p \left( \frac{1}{T_s\cdot(z+1)} + \frac{1}{2\cdot(z-1)\cdot T_N} + \frac{2\cdot T_V\cdot(z-1)}{T_s\cdot(z+1)\cdot[T_s\cdot(z+1)+2\cdot T_{d1}\cdot(z-1)]} \right) \qquad (4)$$

Where $k_p$ – Proportional Gain
Ts – Sampling time
$T_N$ – Integration time constant
$T_V$ – Differentiation time constant
$T_{d1}$ – T1 time constant
z – $z=e^{kTs\cdot s}$ with k=1,2,…

This controller operates in the same mode as an analogue controller, controlling the position of the transmission lever.

## 3. MEASUREMENT RESULTS AND SIMULATIONS

To verify the quality of the mechatronic model, several measurements were carried out.

In order to check thoroughly the mechanical behaviour of the system, the balance was operated in uncontrolled mode, which means the controller was turned off. The current through the coil was changed until the lever moved to zero position. When the lever had settled to a stable position, a small current step was added to the stabilizing current. The step response of the lever was monitored and compared to measurement results. Fig. 5 shows the position of the transmission lever for the simulation and for a measurement. The signals are normalized to the final value of the balances transmission lever.

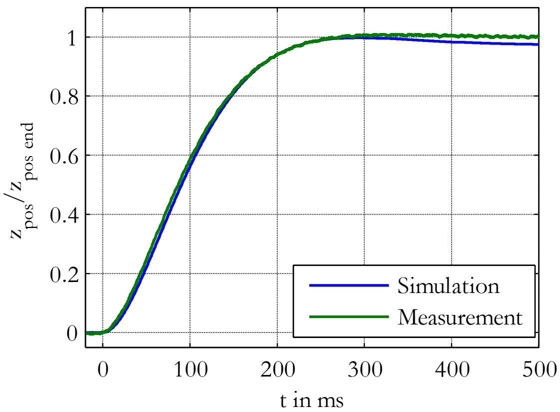


Fig. 5 normalized measured and simulated position of the transmission lever after applying a small current step to the coil in uncontrolled mode

The end position of the simulation differs by three percent of the measurement for this experiment. This result and the dynamic behaviour of the simulation come to lie within the tolerance band of the production spread of the investigated type of balances. For the uncontrolled mode, this shows a very good agreement of measurement and simulation.

In the next step, the balance was operated in controlled mode again. The progression of the position signal and of the current through the coil was observed for a load step on the weighing pan. Several experimental investigations have shown that removing a piece of mass from the weighing pan very quickly generates an nearly ideal step function.

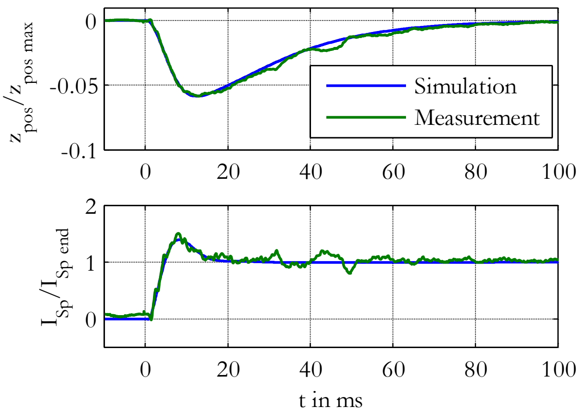


Fig. 6 Normalized position and normalized current showing a load step of 20g

In fig. 6, the response of the position signal (normalized to the maximal possible deflection of the investigated balance) and of the current (normalized to the final current of the balance) to a load step of 20g on the weighing pan is shown. Both for the position signal and for the current, the simulated data come to fit well in the range of manufacturing tolerance.

Design and optimization of controllers require excellent knowledge of the behaviour of the system to be controlled. This can be modelled by transfer functions, giving the dynamic ratio of output signal to input signal. The input of this transfer function is the current through the coil, since this is directly proportional to the acting force on the lever (see eq. 1). The position signal of the lever is applied as the output of the transfer function. Based on these data, a non-parametrical frequency resolved transfer function can be estimated.

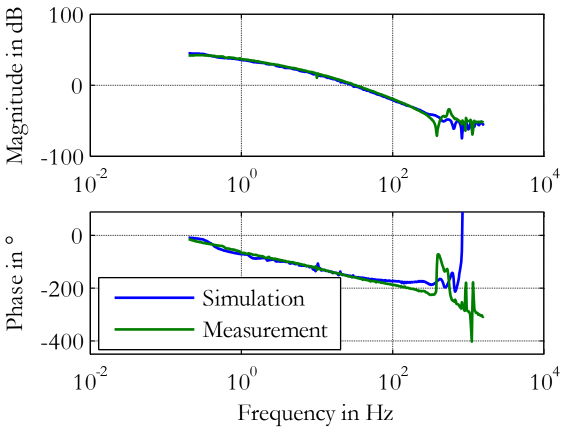


Fig. 8 Bode diagram of transfer functions estimated from measured and simulated data

For the frequency band that is of interest (>5 Hz and <5 kHz), a very good agreement for magnitude and phase of the models estimated from measured and simulated data is achieved (see fig. 8).

The model could be verified in time domain and in frequency domain.

## 4. CONCLUSION

In this paper a parametric model is introduced, which provides an easily manageable tool for the simulation and optimisation of the mechatronic behaviour of EMC balances. The necessary parameters are easy to access experimentally.

In comparison to an FEM model applied to the same set-up, this model stands out due to the smaller computing time. However, this model represents a reduction compared to the FEM model, while still providing crucial data.

Exemplary measurements show that the assumptions and simplifications, if made appropriately, fit to actual practice. Manipulations of all system components (mechanics, electronics and controller) can be carried out on the basis of this model. This model is then able to provide a good characterisation of the behaviour of EMC balances.

On the basis of this model simulations for new controller concepts can be carried out, and new balance configurations can be simulated and verified.

## ACKNOWLEDGMENTS

The project is sponsored by the Federal Ministry of Education and Research (BMBF): Innoprofile – Entrepreneurial Regions, "Innovative Kraftmess- und Wägetechnik" in cooperation with Sartorius AG and SIOS Messtechnik GmbH.

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