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A Contribution to Adjustable Counter Weight Balance for Actuators

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Abstract
PI (Physik Instrumente GmbH & Co. KG) offers a wide portfolio of positioning technologies. Out of this variety, magnetic drives surpass piezo- and spindle-driven stages in high dynamic motion over a wide travel range. This advantage is getting more and more attractive for positioning applications even in vertical direction. Thereby one big challenge for those applications is to compensate the lack of a self-locking mechanism, so that the stage won't accelerate downwards and crash into its hard stops when powered off or due to a loss of power. The favored solution at PI is a counter weight balance, so that the drive is released as much as possible from the payload. Such counter weight balances should be adjustable in force for varying payloads and should work even when systems are powered off. Furthermore, friction is a very important criterion in applications which require nanometer resolution. This paper shows the state of the art of counter weight balance mechanisms and shows a possible solution adjustable in force based on a passive spring system. The general working principle of the spring system will be explained. The discussion of the results will include the adjustable force range and the linearity of force over the travel range compared to calculated values.

Direct Drives, Vertical application, Counter weight balance, Spring system, Adjustable force

1. Introduction

To reduce the motion time in high precision applications the high dynamic characteristics of magnetic drives are especially in vertical direction on the rise. With this tendency a solution for the missing self-locking mechanismis necessary. A counter weight balance solution is preferred over a break system, because it releases the actuator of the tare weight and the payload. Ideally, the counter weight balance is adjustable for varying payloads, so the actuator can exclusively use its force for the positioning task despite a change of the load.

1.1. State of the Art

The first approach for a vertical positioning application should be to avoid a counter weight balance and use a drive technology that includes a self-locking mechanism e.g. spindle drives. If alternative drive concepts won't fulfill the requirements, then a counter weight balance suitable for the application is required.

There are several options for counter weight balance devices depending on the main parameters travel range, force and cost. Here the technologies are differentiated between adjustable in force and not adjustable in force.

A not adjustable magnetic solution for a very small travel range (~1mm) is a magnet in an "U"-shaped iron. Here the geometries must be optimized with a FEM analysis to keep the forces nearly constant. Another magnetic solution has the trade name MagSpring®. These devices are more sophisticated to keep the force constancy over longer travel ranges [1].

A mechanical based solution is a bended coil tension spring [2] are a good solution for travel ranges of some mm. For longer travel ranges this solution gets space consuming. Balance springs with constant force [3] have a very good constancy over long travel ranges. Their disadvantage is a limited lifetime of some thousand movement cycles. Gas springs offer a long travel range, but are very nonlinear and have a friction based hysteresis [4].

A simple adjustable solution is to attach a counterweight via cable over a pulley to the moved mass. The counterweight is changed according to the payload. The disadvantages are more added mass to inertia, additional space requirements and limitation of the acceleration to 1g (taut cable). A pneumatic solution for an adjustable counterweight balance is a pressure controlled pneumatic cylinder [5]. This solution needs pressured air and a pressure control. This makes the solution very expensive (compressor) and space consuming (hoses). Adjustable mechanical solutions are spring systems with additional mechanisms to keep the force of the spring(s) constant over the travel range. These mechanisms vary in complexity and constancy of the resulting force.

This work describes an easy implementation based on a passive spring system to generate an adjustable constant force. The idea of similar adjustable spring systems is used in plant engineering and construction [6]. These systems hold heavy objects and compensate for thermal expansions. The range of generated forces is from 10^3N to 10^5N for static use. The difference to positioning is the range of force and the dynamic use for positioning. With the reduction of the force range to some ten newtons it is also possible to reduce the size of the system.

Described below is the general idea of the function principle and the calculation basis. A possible design example is shown, measured and compared to the calculations. The planed future work concludes this contribution.

2. Function Principle

The spring system consists of two compression coil springs, one main and one compensation spring. The force of the compensation spring (F_{comp}) counterbalances the force of the main spring (F_{main}) to achieve a constant resulting force (F_{res}).

This is done by pivoting the compensation spring next to the main spring, so that the effective force (F_{eff}) is a function of the angle (ϕ). F_{res} is the superposition of F_{main} and F_{eff}. ϕ is depending on the distance between pivot point to the joint at a linear bearing (a) and the position (s) of the linear bearing. F_{res} is guided by the linear bearing.

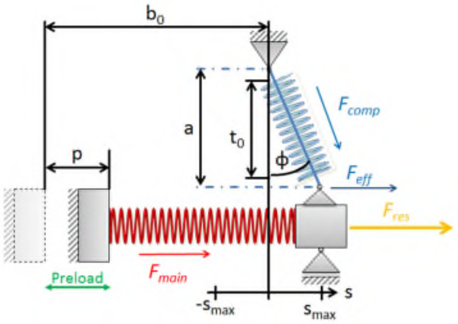

Figure 1. Schematic design of the spring system

s is limited via hardstops symmetrically to the limit positions $\pm s_{max}$. With moving the spring seat by a certain distance (p), a preload is applied to the main spring. The preload makes F_{res} adjustable to the payload. The assumption is that F_{comp} is nearly linear for small ϕ, which also applies for F_{eff}.

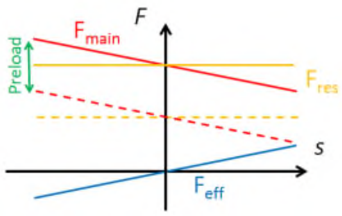

Figure 2. Simplified force path and influence of the preload

To calculate F_{eff} as a function of s the smallest length of the compression spring at s=0 (t_0) is required. With the spring rates (k_{comp}, k_{main}) and the relaxed spring lenghts (L_{comp}, L_{main}) it is possible to calculate F_{eff} and F_{main} as a function of s.

Equation 1 F_{eff} as a function of s

$$F_{eff}(s) = sin\left(arctan\left(\frac{s}{a}\right)\right) \cdot k_{comp} \cdot \left(L_{comp} - t_0 + \left(\sqrt{s_{max}^2 + a^2} - \frac{a}{cos\left(arctan\left(\frac{s}{a}\right)\right)}\right)\right)$$

Equation 2 F_{main} as a function of s

$$F_{main}(s) = k_{main} \cdot (L_{main} + p - b_0 - s)$$

The limitation of this model is that the angle of ϕ won't be so small that the F_{eff} has linear behaviour. Friction is also not included in the calculations.

3. Mechanical Design and Calculation

The actual design consists mostly of standard parts and some simple milling parts to keep the assembly low-cost. The difference to the schematic design is, that F_{res} is used as a pulling force through the preload part with the thread.

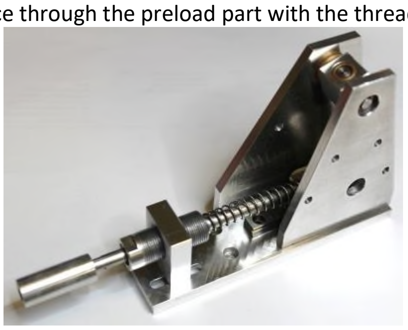

Figure 3. Prototyp of spring system for testing

The prototype is calculated with the following parameters: a: 46.5mm, b_0: 47.5mm, k_{main}: 0.35N/mm, L_{main}: 54mm, k_{comp}: 1.32N/mm, L_{comp}: 28.8mm, p: 18.65mm, s_{max}: 10mm and t_0: 16.2mm.

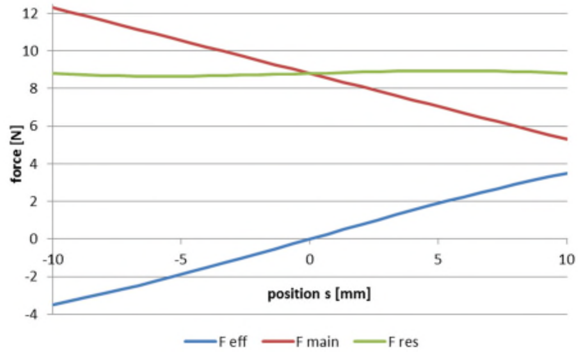

Figure 4. Calculation example of forces over travel s

4. Measurements and Qualification

The forces were measured on a teststand with a load cell. The measurements show a hysteresis between positive direction (red curve) and negative direction (blue curve) caused by internal friction of the springs. The calculated curve (violet) fits in the middle of both curves and confirms the calculation. The hysteresis (green curve) has a mean value of about 1.5N, which means a deviation of the calculated force of ±0.75N at a mean force of roughly 8.75N.

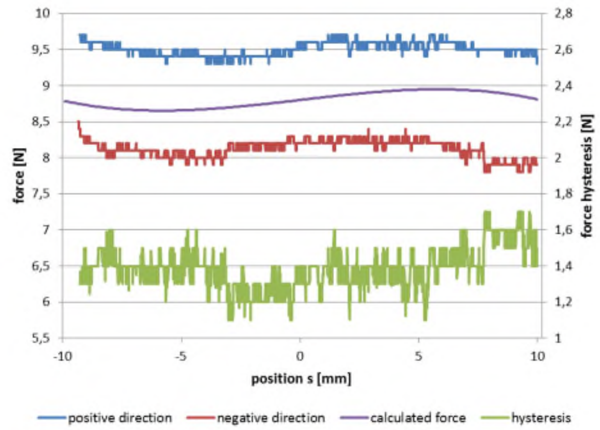

Figure 5. Comparison between calculated and measured values

5. Summary and Future Work

The shown counter weight balance based on a passive spring system generates an adjustable mean force from 3.5N to 11N. The tolerances from friction based losses is within an acceptable range of ±10%. The next step is to create a closed system device which can be attached to an actuator as a kind of "backpack". The final step is to integrate the mechanism directly into an actuator.

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