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A high-resolution and large force-range load cell by means of nonlinear cantilever beams

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ABSTRACT

In the present work, we describe a nonlinear stiffening load cell with high resolution (the ability to detect 1% changes in the force) that can function over a large force range (5 orders of magnitude), and exhibit minimal hysteresis and intrinsic geometric protection from force overload. The stiffening nature of the load cell causes its deflection and strain to be very sensitive to small forces and less sensitive to large forces. High stiffness at high forces prevents the load cell from over-straining. We physically implement the nonlinear springs with cantilever beams that increasingly contact rigid surfaces with carefully chosen curvatures as more force is applied. We analytically describe the performance of the load cell as a function of its geometric and material parameters. We also describe a method for manufacturing the mechanical component of the load cell out of one monolithic part, which decreases hysteresis and assembly costs. We experimentally verify the theory for two load cells with two different sets of parameters.

1. Introduction

Load cells are useful for applications ranging from material strength testing to prosthetic limb sensing [1], monitoring infusion pumps delivering drugs [2], agricultural product sorting [3], suction cup strength measuring [4], and human–robot collision force sensing [5].

Load cells can measure forces via several different methods, including hydraulic or pneumatic pistons and deforming materials. For hydraulic or pneumatic load cells, the force is applied to a piston that covers an elastic diaphragm filled with oil or air respectively, and a sensor converts a pressure measurement to a force measurement. Use of hydraulic load cells is limited by high cost and complexity. Pneumatic load cells are limited by slow response times and a requirement for clean, dry air [6]. The most common load cells are solid materials that deform when subject to an applied force.

Deforming load cells come in many different shapes, such as bending beams (a cantilever), S-beams (an "S"-shaped configuration of beams), single point load cells (a double-clamped beam, for which the force measurement is insensitive to the position of the load along the beam), shear beam load cells (an I-beam produces a uniform shear across its cross-section that can be measured by strain gauges), and "pancake" load cells (bending disks) [6]. All of these load cells deflect linearly.

Traditional linear load cells can be designed for almost any force capacity. Bending beam load cells are typically used for force ranges of 5.0×10^1–2.5×10^4 N and pancake load cells can be used for force ranges up to 2.5×10^6 N [6]. Many linear load cells are designed to withstand a limited amount of force overcapacity using overstops that prevent over-deflection; typically up to 50–500% load capacity before breaking [7]. Because they deform linearly, these load cells also have constant resolution (that is, the smallest force increment that they can measure) for their entire force range.

There are several challenges to designing a load cell. One wants to reduce the load cell mass and volume to minimize its effect on the test sample. Additionally, the load cell should have minimal hysteresis for accurate measurements in both up-scale and down-scale, and low side-load sensitivity (response to parasitic loads) [6]. One of the most critical design challenges is the trade-off between force sensitivity and range: It is desirable to maximize strain or deflection in the load cell in order to increase force measurement resolution because strain and deflection sensors have limited resolution; typically 14-bits between 0 and their maximum rated measurement [8–10]. Simultaneously, one wants to maximize the load cell's functional force range and protect it from breaking due to forces that exceed that range, which requires limiting its strain.

Different studies have made various modifications to the traditional linear load cell to increase its force range and sensitivity, and minimize side-load sensitivity. Chang and Lin [3] studied a "capital G-shaped" load cell with two force ranges: for small forces, a top

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sensitive flexure deflects alone. For large forces, the sensitive flexure contacts a stiffer flexure, and the two flexures deflect together at the higher stiffness. In this way, the load cell is more sensitive to small forces and does not yield for large forces. Other devices use multiple linear load cells of increasing stiffnesses in series, as described in several U.S. patents [11,12]. The multiple load cells of a single device deflect together until overload stops prevent the weaker load cells from deflecting too far, after which the stiffer load cells continue to deflect. A microcontroller determines which load cell measurement to display. Using this approach, Storace and Sette [11] were able to measure weights over a range of 1 g to 30 kg. One way to minimize sensitivity to side-loads such as undesired moments is to use multiple load cells (i.e. 3) and take the average force measurement [11]. Challenges with these designs are that the linear load cell components have limited resolution, and using multiple load cells in one device may be bulky or expensive.

Another approach for designing a load cell with high force resolution and capacity is to use a nonlinear mechanism rather than a linear one. A nonlinear load cell may have a low stiffness at low forces (and therefore high force sensitivity) and a high stiffness at large forces (and therefore protection from yielding due to over-deflection). The design may also be volume compact and inexpensive due to requiring only one nonlinear spring and sensor per device.

A nonlinear spring may be physically realized in many different ways. The simplest form of a nonlinear spring is a cubic spring. One way to implement a cubic spring is by linear springs supporting a proof mass at various angles to its direction of travel. For example, MacFarland et al. [13] investigate a nonlinear spring realized by a thin elastic rod (piano wire) clamped at its ends without pretension that displaces transversely about its center. To leading order approximation, the stretching wire produces a cubic stiffness nonlinearity. Similarly, Hajati et al. [14] describe a spring made out of a doubly-clamped piezoelectric beam. The double-clamps cause the beam to axially stretch as it bends, resulting in a nonlinear stiffness. Mann and Sims [15] describe a spring that is implemented by a magnet sliding in a tube with two opposing magnets as the end caps. This configuration causes the stiffness to be the summation of a linear and cubic component. Kantor and Afanas'eva [16] describe the nonlinear stiffness of a clamped circular plate with variable thickness along its radius, which has a force-displacement curve similar to that of a cubic spring.

This paper describes a nonlinear stiffening load cell with high resolution (within 1% of the force value) that can function over a large range (5 orders of magnitude), with minimal hysteresis and intrinsic geometric protection from force overload. The stiffening nature of the load cell causes its deflection and strain to be very sensitive to small forces and less sensitive to large forces. When used with a constant-resolution sensor, this causes the load cell as a whole to have higher resolution for smaller forces. High stiffness at high forces prevents the load cell from over-straining. In Section 2, we develop the theory for this load cell, which uses cantilever beams that increasingly contact surfaces with carefully chosen curvatures as more force is applied. In Section 3, we describe a method for manufacturing the mechanical component of the load cell out of one monolithic part, which decreases hysteresis and assembly costs. In Section 4, we experimentally verify the theory for two load cells fabricated using the described method. Our findings are summarized in Section 5.

2. Theoretical modeling

We design the load cell as a 2×2 symmetric grid of nonlinear spring elements, as shown in Fig. 1. Load cell deflection occurs between the top and bottom rigid blocks. The nonlinear springs are physically realized by cantilevers that make contact with rigid surfaces as they deflect (splitting each cantilever length into a "segment in contact with the surface" and a "free segment"). As the contact length increases, the shortening length of the free cantilever segment causes the stiffening spring behavior. The single cantilever-contact surface nonlinear spring mechanism was first described by Timoshenko [17]. We analyze a similar nonlinear spring in Kluger et al. [18] and Kluger [19] in the context of energy harvesting from ambient vibrations. The tips of the bottom cantilevers connect to the tips of the top cantilevers by vertical rigid bars, which cannot rotate due to symmetry. To further ensure symmetry, we design the device with a rigid horizontal crossbar connecting the vertical rigid bars, as shown in Fig. 1(b), which strengthens the load cell's resistance to parasitic moments and horizontal loads. In this paper, we study load cells where the cantilever tips are either rigidly connected to these vertical bars (Fig. 1(a)) or connected to the vertical bars via moment-compliant flexures that are physically realized by three-quarters of a circular ring (Fig. 1(b)). As we show in Section 2.5, adding the 3/4-rings reduces the maximum stress in the load cell at a given applied force.

Table 1
Nomenclature for load cell components.

1/4 load cell	One of the four symmetrical spring elements, as shown in Fig. 1.
Rigid block	One of two symmetrical rigid blocks, each with four surfaces with a carefully chosen curvature.
Cantilever	One of four cantilevers with length L_{Cant}, width b, and thickness t.
Contact point, x_c	Point that separates the cantilever segment in contact with the surface and free cantilever segment and is a function of the applied force.
Cantilever segment in contact with the surface	Segment of the cantilever that is tangent to the surface, with a length from $x=0$ to $x=x_c$.
Free cantilever segment	Segment of the cantilever that is not tangent to the surface, with a length from $x=x_c$ to $x=L$, as shown in Fig. 1.
Moment compliance ring, or 3/4-ring	270° circular arc used as a rotational spring to connect the cantilever tips to the rigid vertical bars.
Rigid connection	Junction of the cantilever tip and rigid vertical bar when the load cell does not have a 3/4-ring.
Rigid vertical bar	Component connecting the top and bottom cantilevers. It cannot rotate due to symmetry when the load cell is in pure tension/compression loadings.
Rigid crossbar	Horizontal component connecting the left and right rigid vertical bars that stiffens the load cell's response to parasitic moments and horizontal forces.
Root gap	Location of removed material near the root of the surface curve that may be required by machining limitations described in Section 3.
Root insert	Rigid blocks that follow the surface curve and can be inserted into the root gaps, described in Section 3.

Throughout this paper, we will use notation for the nonlinear spring components listed in Table 1.

We set the 1/4 surface shape to follow the curve

$$S = D\left(\frac{x}{L}\right)^n, \tag{1}$$

where $L = L_{\mathrm{Surf}} = L_{\mathrm{Cant}}$ is the cantilever and surface length (assuming small cantilever deflections), x measures the location along the length of the beam from its root, D is the end-gap between the surface and undeflected cantilever, and n is a power greater than 2. In theory, any curve with a monotonically increasing curvature