LOW CREEP AND HYSTERESIS LOAD CELL BASED ON A FORCE TO LIQUID PRESSURE TRANSFORMATION

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0. Abstract

Important problems in load cells are creep and hysteresis. Expensive high grade steels are used in order to reduce these effects. In this paper a load cell design based on a force to liquid pressure transformation is presented. The design is insensitive to hysteresis and creep and can be made at very low costs. Analytical, numerical and experimental results are in very close agreement with each other.

1. Introduction

A load cell is a sensor for the measurement of forces and masses. Load cells are for example used in weighing bridges for lorries, cars and trailers. Also in industries where bulk material is worked up, it is necessary to measure masses as accurate as possible.

Most current load cells are made of steel. The performance of these load cells is limited by hysteresis and creep even when expensive high grade steels are used. Hysteresis means that the output is a function of the loading history. Creep means that for a fixed load the output changes in time. In order to increase stability and to decrease the costs, a new type of load cell is presented which is nearly free of hysteresis and creep and which has a very simple design. The load cell we are aiming at has the following specifications:

- maximum load: 10000 N (1000 kg)
- accuracy of full scale: 0.03 % = 0.3 kg
- temperature range: -10 up to 50 °C
- production costs: less than US $ 75
- calibration: once in two years

The load cell discussed in this paper transforms the force into a fluid pressure. This pressure is measured by a commercial silicon pressure (pressure range: 0-8.8 bar; price: US $ 25) sensor which is because of its mono cristallinity not sensitive to creep and hysteresis. In [2], a fluid pressure load cell was introduced as a new kind of load cell. It consist of a piston under which the fluid is locked up by a seal. Characteristic for this kind of load cells is the high sensitivity. However, the load cell presented in [2] is still rather sensitive to hysteresis. Another disadvantage is that it needs a standard Teflon seal which requires a proper surface finish of the mating steel parts. The load cell discussed in this paper does not have these drawbacks.

In section 2 the concept of the force to liquid pressure transformation is explained. Section 3 considers the modeling of the load cell. The realization and experiments are treated in section 4. Finally, in section 5 some conclusions and recommendations are drawn.

2. Concept of force to liquid pressure transformation

In most current load cells the force F is applied to a steel bar which will be deformed (see figure 1a). The deformation is sensed by strain gages. This is a force to strain transformation and the strain e of this type of load cell can be expressed as

$$e = C\frac{F}{E}, \tag{1}$$

where C is a constant and E Youngs modulus of the bar. As the output signal depends on Youngs modulus, this sensor is sensitive to hysteresis and creep.

Another option is to use a force to liquid pressure transformation. The most simple example of this kind of load cells is given in figure 1b. The force to pressure transformation is given by

$$P = \frac{F}{A}, \tag{2}$$

where A is the area of the piston. The pressure is sensed by a silicon pressure sensor which is because of its mono cristallinity independent of hysteresis and creep. It follows from (2) that the signal is independent of Youngs modulus and, therefore, insensitive to hysteresis and creep. The drawback of this kind of load cell is that a seal has to be used which imposes demands on the surface finish of the piston and bucket. Besides, the friction of the seal may be a new source of hysteresis. In order to eliminate these drawbacks the load cell presented in this paper is designed.

Figure 2 shows a schematic drawing of this load cell which is axial symmetric. A liquid is locked up between the steel membrane and the steel bucket. The force F is applied to the steel boss which is attached to the steel membrane. A silicon pressure sensor measures the difference between the fluid pressure P_{fluid} and the air pressure P_{air}. It is seen that no seal is needed. In the next section it will be shown that this load cell has a same kind of relation as in (2). The only difference is that the area for calculating the pressure is some effective area (A_{eff}) which only depends on the ratio of radii r_1 and r_2:

$$P_s = P_{fluid} - P_{air} = \frac{F}{A_{eff}}. \tag{3}$$

In the next section this load cell is modeled and the function A_{eff} is determined.

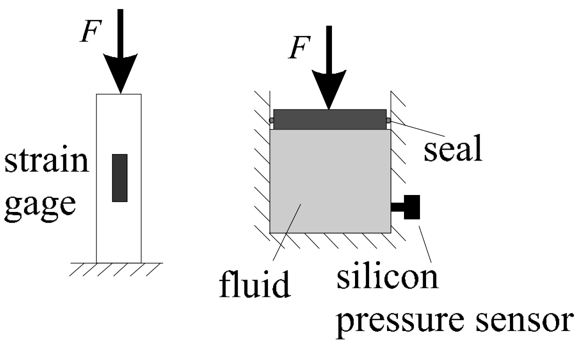

figure 1: (a) conventional steel load cell. (b) fluid compressed by a piston.

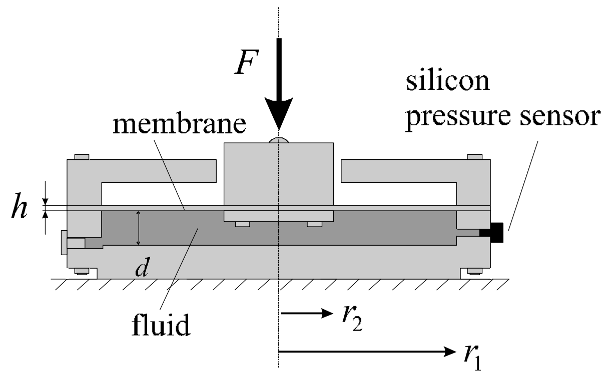

figure 2: Layout load cell.

3. Model

In this section the load cell is modeled mathematically. In §3.1 it is assumed that the water is incompressible. In this paragraph the pressure-force relation, the deformation profile of the membrane and the stresses in the membrane are calculated. Then, in §3.2 the compressibility of the water is taken into account. In §3.3 the temperature dependence of the load cell is considered.

3.1 Incompressible fluid

In figure 3 the deformation of the membrane and the deflection of the boss for some load F is shown. The pressure-force relation and deflection profile of the membrane are derived in the next way:

- Use classical plate elasticity theory [1].
- The deflection profile $w(r)$ of the membrane has to satisfy some boundary conditions:

$$w(r=r_1)=\frac{dw}{dr}(r=r_1)=\frac{dw}{dr}(r=r_2)=0 \qquad (4)$$

- Incompressibility of the fluid is satisfied if the following condition is fulfilled:

$$p r_2^2 w(r=r_2)+\int_{r=r_2}^{r_1} 2prw(r)dr=0 \qquad (5)$$

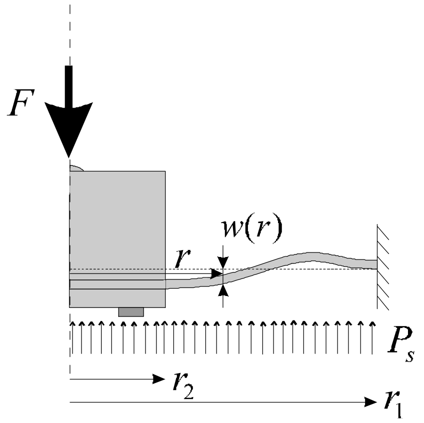

figure 3: Deformation of membrane and boss.

The general expression for the deflection $w(r)$ of the membrane is given by

$$w(r)=\frac{F(n^2-1)}{pEh^3}\, l(\frac{r_1}{r_2},r_1,r), \qquad (6)$$

where n is Poissons ratio and h the thickness of the membrane. $l(r_1/r_2, r_1, r)$ is a rather complex function. An impression of the deformation of the membrane is given in figure 4.

The pressure-force relation is given by:

$$P_s=\frac{F}{A_{eff}},\quad A_{eff}=\frac{pr_2^2}{p}\frac{\left(1-4s^2+6s^4-4s^6+s^8\right)}{\left(3.82\ln(s)\left[s^2-s^4\right]+0.95\left[1-s^2-s^4+s^6\right]\right)},\quad s=\frac{r_1}{r_2}. \qquad (7)$$

From (7) it follows that there is no dependence on the elastic modulus of the steel membrane so that it is concluded that the load cell is insensitive to hysteresis and creep. A_{eff} is the effective area which was discussed in section 2. An impression of the effective area in

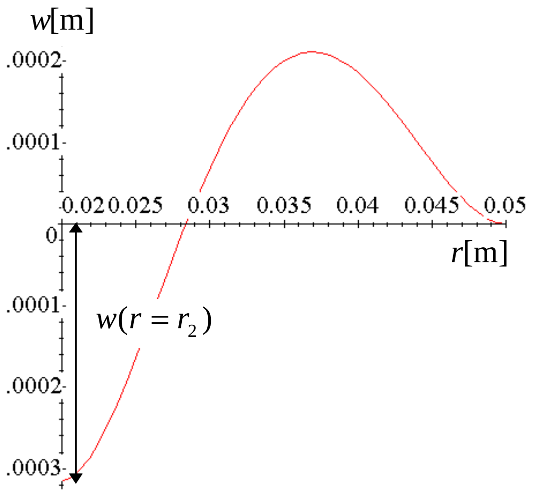

figure 4: Deformation of the membrane ($F = 10000\ N$, $E = 210\ Gpa$, $n = 0.3$ $h = 1\ mm, r_1 = 5\ cm, r_2 = 2\ cm, d = 1\ cm$).

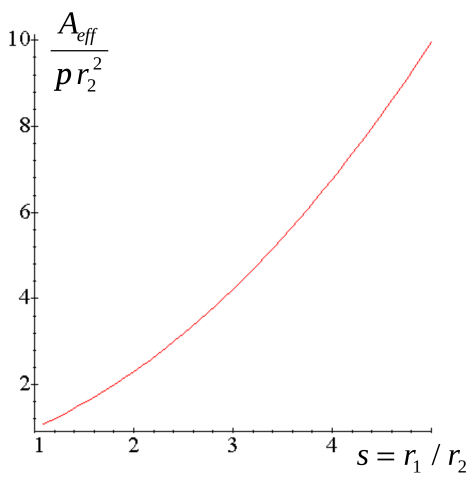

figure 5: Effective area divided by the area of the boss as a function of the ratio r_1/r_2.

comparison to the area of the boss and to the total area of the bucket (=area of boss + membrane) are respectively shown in figure 5 and figure 6. From these plots, it is concluded that the effective area lies somewhere between the area of the boss and the bucket.

As the membrane is rather thin it must be checked whether the critical stress is not exceeded. In the membrane three stresses are present: radial, tangential and shear stress acting in the direction of force F (s_r, s_t and t respectively). The first two are calculated from their corresponding moments [1] and they have a maximum on both sides of the membrane. As all three stresses occur at the same place in the membrane, the Von Mises stress criterion is used for comparison to the maximum allowable stress. It is given by

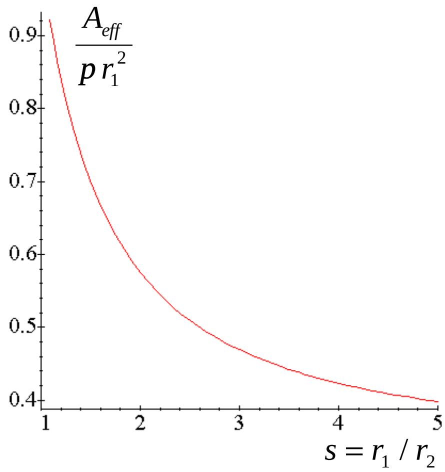

figure 6: Effective area divided by the area of the bucket as a function of the ratio r_1 / r_2.

$$s_v = \sqrt{\frac{1}{2}\left[(s_r - s_t)^2 + s_t^2 + s_r^2 + 6t^2\right]}. \tag{8}$$

A plot of all the stresses is given in figure 7. The ultimate stress of steel is about 1 to 1.5 Gpa, so it should just be possible to bear a load of 10000 N. As the shear stress t is very small in comparison to $s_{r,\max}$ and $s_{t,\max}$ this stress can be neglected. Besides, it is seen that the maximum stresses occur at $r = r_2$. Therefore, the maximum Von Mises stress in the membrane is approximately given by

$$s_{v,\max} = \frac{3F\sqrt{(1-n-n^2)}}{4ph^2}g(s), \qquad g(s) = \frac{\left(4\ln(s)s^4 - 5s^4 + 8\ln(s)s^2 + 4s^2 + 1\right)}{(1-2s^2+s^4)}. \tag{9}$$

The function $g(s)$ is drawn in figure 8. It is concluded that the maximum stress strongly depends on the thickness h of the membrane. The dependence on the ratio r_1 / r_2 is less strong.

For the parameters given in the caption of figure 4 the pressure sensitivity is calculated from (7) giving $¶P_s / ¶F = 250.03\ \mathrm{Pa / N}$.

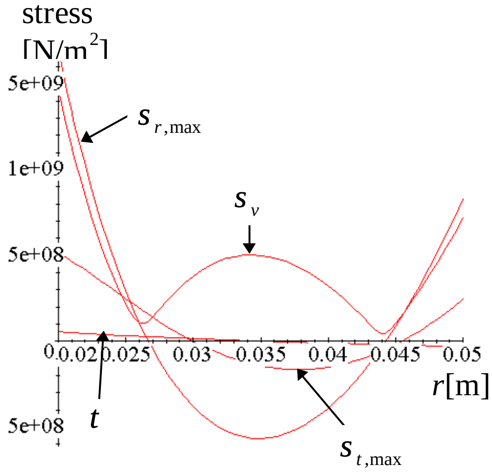

figure 7: Radial, tangential, shear and Von Mises stresses in the steel membrane.

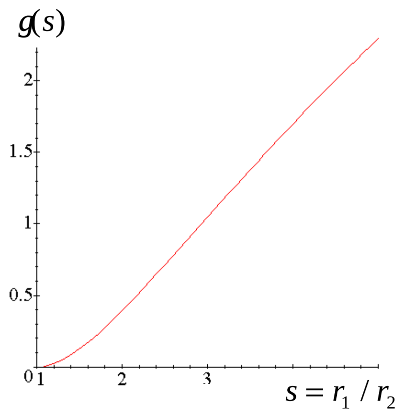

figure 8: Von Mises stress parameter $g(s)$.

3.2 Compressible fluid

By including compressibility of the fluid, equation (5) is changed to

$$pr_2^2 w(r = r_2) + \int_{r=r_2}^{r_1} 2prwdr = \Delta V = V_0 \frac{P_s}{E_v}, \tag{10}$$

where E_v is the bulk compressibility modulus, V_0 the initial volume and ΔV the decrease in volume of the compressed fluid. For water $E_v = 2.24 \text{ GN / m}^2$. The effective area is calculated in the same way as done in §3.1 giving

$$A_{eff} = pr_2^2 \frac{\left\{1 - 3s^2 + 3s^4 - s^6 + \dfrac{16 V_0 E h^3}{p E_v r_1^6 (n^2 - 1)} s^6)\right\}}{3.82 \ln(s) s^2 + 0.95\left[1 - s^4\right]}. \tag{11}$$

From this equation it is concluded that hysteresis and creep are present, because of Youngs modulus. Therefore, the term $\dfrac{16 V_0 E h^3}{p E_v r_1^6 (n^2 - 1)}$ must be chosen as small as possible. The most simple tool for reducing this term is by giving a very small value to the initial volume of the fluid. For the parameters given in the caption of figure 4 the pressure-sensitivity equals $\P P_s / \P F = 248.92 \text{ [Pa / N]}$. So it is concluded that the condition of incompressibility can be maintained.

The load cell is also analyzed in the finite element program Ansys. For the (compressible) fluid, FLUID79 elements are taken. The deformation profile and displacements in the direction of the force are shown in figure 9. The deformation profile appears to be the same as was obtained with analytical formulas. The pressure sensitivity equals $\P P_s / \P F = 247.05 \text{ [Pa / N]}$ which is about the same as the analytical result.

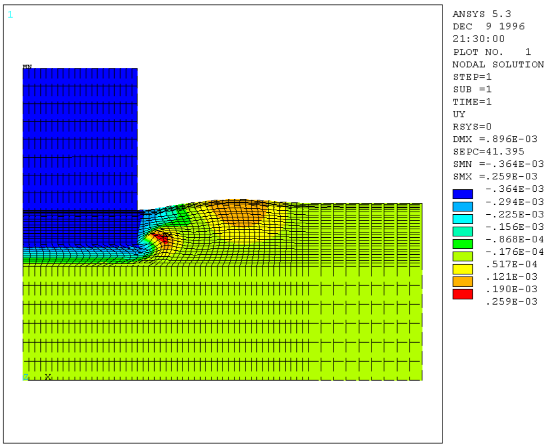

figure 9: Displacements in direction of applied force.

3.3 Dependence on temperature

The effective area on which the fluid presses, changes linearly with temperature:

$$\frac{dA_{eff}}{dT} = 2a_{steel} A_{eff}. \tag{12}$$

T is the temperature and a_{steel} the temperature expansion coefficient of steel. However, due to large differences in temperature expansion coefficients of the steel and fluid, this effect is overruled. Practical values are $a_{Steel} = 12 \cdot 10^{-6}\ ^\circ\text{C}^{-1}$ and for water at 20 °C, $a_{water} = 207 \cdot 10^{-6}\ ^\circ\text{C}^{-1}$. For these parameters Ansys calculates a temperature sensitivity of $\P P_s / \P T = 4849 \text{ Pa / C}^\circ$. If $a_{fluid} = a_{steel}$ is taken, then it follows from Ansys that $\P P_s / \P T = 0.54 \cdot 10^{-7} \text{ Pa / C}^\circ$ which is very low so that it is concluded that the temperature coefficients of the steel and fluid should match as close as possible.

The temperature sensitivity can also be decreased by reducing the volume of the fluid. Decreasing this volume by a factor 10 gives a decrease in temperature sensitivity by about a factor 10.

4. Realization and experiments

The load cell is tested by using a commercial Honeywell pressure sensor. The repeatability and hysteresis of this sensor is less than 0.15 %. For the fluid, water is used. Hysteresis of the load cell is tested by loading, then unloading and hereafter loading the load cell with some weight. The output of the Wheatstone bridge of the pressure sensor is shown in figure 10. The load cell behaves linearly and the pressure sensitivity equals $\partial P_s / \partial F = 249\ [\mathrm{Pa} / \mathrm{N}]$ which is in very close agreement with the analytical and numerical results. The sensor could not be loaded above 400 kg, because the pressure sensor is not able to withstand a higher pressure. The hysteresis as a percentage of the full scale maximum load (that is 1000 kg) is less than 0.009 % which is far more accurate than the 0.03 % that was specified in section 1. For the calculation of the full scale output of the pressure sensor linear behavior of the pressure sensor is assumed which is correct, because its linearity error is less than 0.25 % of full scale.

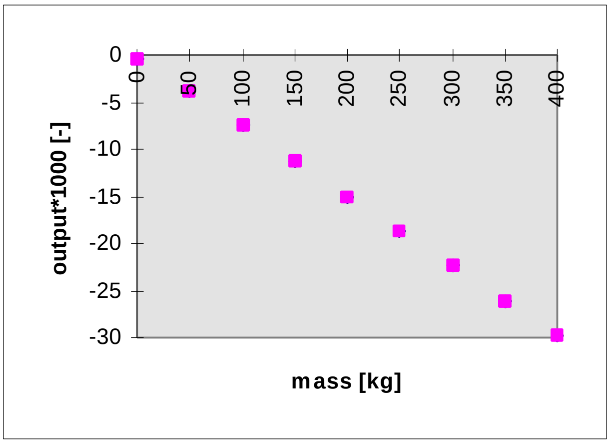

figure 10: Hysteresis in the load cell.

So far, it was not possible to test creep, because the pressure sensor and the force to fluid pressure transformation are rather sensitive to temperature variations.

5. Conclusions and recommendations

It is proved that the force-pressure relation of the designed load cell is independent of the elastic modulus and therefore is independent of hysteresis and creep. The analytical, numerical and experimental results are in very close agreement with each other. It is shown by experiments that the hysteresis of the load cell is less than 0.009 %. As the hysteresis of the pressure sensor can be greater than 0.009 % the hysteresis in the force to liquid pressure transformation might even be smaller. The dependence of the fluid pressure on temperature variations can be eliminated by choosing a very small initial volume of the water and by matching the temperature expansion coefficients of the fluid and steel.

At the moment the load cell is being made completely of silicon. Calculations have shown that this is indeed possible. The advantage is that load cells can be made in large amounts and at very low costs.

In the future creep must be investigated experimentally. In addition, a pressure sensor has to be used which is insensitive to temperature variations and which has an accuracy better than 0.03 %. Fluids must be found with temperature expansion coefficients which equal the coefficient of the steel as close as possible.

6. References

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[2] H. Kazerooni, M.S. Evans and J. Jones, Hydrostatic force sensor for robotic applications, Journal of dynamic systems, measurement and control, 119(Issue 1), 1997, 115-119.