A massively overconstrained and statically balanced flexure mechanism for a 20 kN load capacity

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INTRODUCTION

In high-precision applications, flexure mechanisms allow for highly predictable, accurate motion as their flexural elements bend with neither friction, wear nor play. However, flexure mechanisms also come with a tradeoff between range of motion and load-bearing capacity, since stress due to deflection limits the thickness of the flexures [1], [2]. Also, the actuation stiffness needs to be limited, as to avoid large actuators, heat generation and undesired thermal expansion [3], [4].

In pursuit of a high load capacity of flexure elements, we propose a massively overconstrained parallel flexure guidance. By placing a multitude of flexures in parallel we obtain the desired loading capacity of over 20 kN. This approach alone leads to excessive actuation forces. By careful selection of the flexure thickness in conjunction with the applied constant load, we may balance the flexures such that (close to) zero actuation force is required. In previous research [5] we presented that this is also advantageous for lowering the stress in flexures, allowing for a higher thickness and load capacity. Such high-load, statically balanced flexure mechanisms may be used in passive vibration isolation systems such as the ASML balance mass or seismic attenuation filters [6].

METHOD

The actuation force F_a of a parallel flexure guidance (Figure 1) with n flexures becomes zero if the applied load F_l equals the system's balance load F_b

$$F_l = F_b = n\frac{\pi^2 EI}{l^2} \qquad (1)$$

where E, I, and l are the Young's modulus, the flexures' moment of inertia, and length, respectively.

The overconstrained nature of the design may lead to undesired internal loads due to manufacturing tolerances and thermal gradients [7]. Deleterious consequences include additional stress, binding [8] and unexpected buckling behavior. Previous work [9]–[11] has shown that these adverse effects may be avoided if the tolerances stay within certain limits.

Here, we investigate the robustness of the proposed design, by simulating the effect of load misalignment distance d on the actuation force, stress in the flexures and support stiffness for various number of flexures $n = [2, 5, 20]$. The parameters used in the simulation are found in Table 1. Load misalignment from the shuttle center is evaluated over the full width of the shuttle.

RESULTS

Figure 2 shows that the moments introduced by a misalignment cause uneven loading over the different flexures resulting in a reduced performance on all metrics. As the unloaded actuation force for $n = 20$ is $F_{\mathrm{a,unload}} = 1300\ \mathrm{N}$, still a formidable reduction of actuation force is achieved even for the worst load misalignment case. The stresses in the flexures, however,

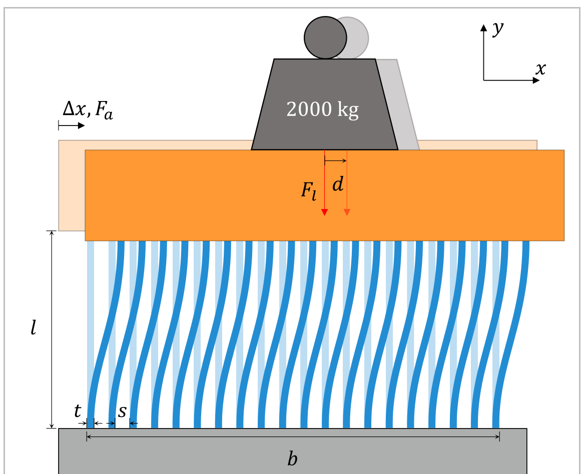

FIGURE 1. A massively overconstrained parallel flexure guidance with 20 flexures supports a large constant load F_l. This load statically balances the mechanisms leading to zero actuation force F_a.

TABLE 1. The design parameter of the parallel flexure guidance

	Symbol	Value	Unit
Displacement	Δx	5.00	mm
Elastic modulus	E	210	GPa
Guidance width	b	100	mm
Flexure length	l	100	mm
Flexure width	w	50.0	mm
Flexure thickness	t	1.07	mm

increase strongly as half of the flexures are loaded under tension. Although the support stiffness k_{yy} only slightly decreases, the lowest internal resonance frequencies reduce by a factor of two, indicating an increased susceptibility to buckling. It should be noted that we evaluate a rather large load misalignment distance. In practice the load will be applied close to the shuttle center and these adverse effects may be minor due to the quadratic nature of the response.

In the full version of this paper we will add a sensitivity analysis for internal stresses due to production tolerances or thermal gradients.

CONCLUSION

In this paper, we present the idea of massively overconstrained flexure mechanisms with near zero actuation stiffness for high load capacity and investigate its robustness to production tolerances, thermal gradients and load variation. We will show that using the principle of static balance, the load capacity of a parallel flexure guide may be increased by at least one order of magnitude while keeping the actuation forces within limits.

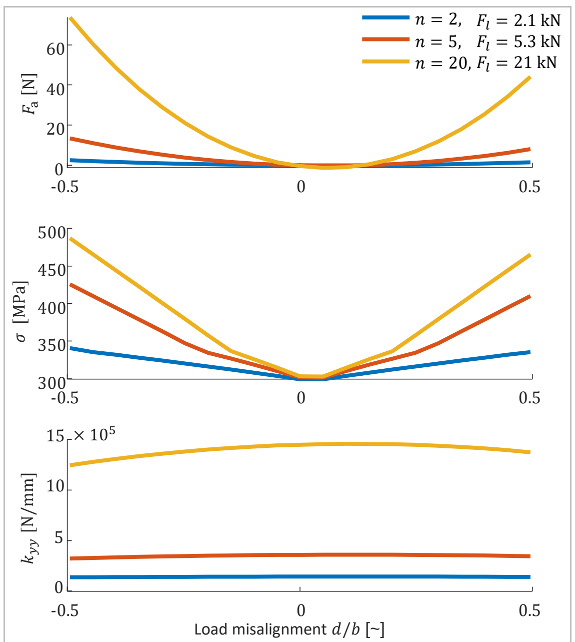

FIGURE 2. Variation in the load misalignment distance d may lead to increased actuator forces (top), increased stresses (middle) while the vertical support stiffness is lowered only slightly (bottom). All simulations are performed for a shuttle displacement of $\Delta x = 5\ mm$.

REFERENCES

[1] J. Van Eijk, "On the design of plate-spring mechanisms," 1985.

[2] S. T. Smith, *Flexures: Elements of Elastic Mechanisms*. Taylor & Francis, 2000.

[3] Q. Yang, S. Bi, and T. Liu, "Static balancing of flexural pivots with two symmetrically arranged pre-compressing springs," in *IOP Conference Series: Materials Science and Engineering*, 2020, vol. 892, no. 1, doi: 10.1088/1757-899X/892/1/012121.

[4] E. G. Merriam and L. L. Howell, "Non-dimensional approach for static balancing of rotational flexures," *Mech. Mach. Theory*, vol. 84, pp. 90–98, 2015, doi: 10.1016/j.mechmachtheory.2014.10.006.

[5] J. J. de Jong, S. Theans, L. J. Epping, and D. M. Brouwer, "Improving Support Stiffness of Flexure Mechanisms By Statically Balancing," *Proc. - 36th ASPE Annu. Meet.*, pp. 45–48, 2021.

[6] G. Bergmann *et al.*, "Passive-performance, analysis, and upgrades of a 1-ton seismic attenuation system," *Class. Quantum Gravity*, vol. 34, no. 6, 2017, doi: 10.1088/1361-6382/aa5e1f.

[7] H. Soemers, *Desing Principles for Precision Mechanisms*. Herman Soemers, 2011.

[8] A. H. Slocum, *Precision machine design*. Prentice Hall., 1992.

[9] S. Awtar, K. Shimotsu, and S. Sen, "Elastic averaging in flexure mechanisms: A three-beam parallelogram flexure case study," *J. Mech. Robot.*, vol. 2, no. 4, 2010, doi: 10.1115/1.4002204.

[10] J. P. Meijaard, D. M. Brouwer, and J. B. Jonker, "Analytical and experimental investigation of a parallel leaf spring guidance," *Multibody Syst. Dyn.*, vol. 23, no. 1, pp. 77–97, 2010, doi: 10.1007/s11044-009-9172-4.

[11] M. Nijenhuis, J. P. Meijaard, and D. M. Brouwer, "Misalignments in an overconstrained flexure mechanism: A cross-hinge stiffness investigation," *Precis. Eng.*, vol. 62, pp. 181–195, 2020, doi: 10.1016/j.precisioneng.2019.11.011.

PERFORMANCE OF A VIBRATION-DESENSITIZED SCANNING WHITE LIGHT INTERFEROMETER

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ABSTRACT
Scanning white light interferometers (SWLI) normally require vibration isolation for optimum performance; frequently they are located in "metrology labs" far removed from the shop floor. A new commercial "vibration tolerant" scanning white light interferometer suggests that precision surface measurements might be made much closer to the shop floor. This paper examines what is meant by "vibration tolerant" and describes a method for evaluating the performance of similar instruments.

INTRODUCTION
To enable surface quality measurements in manufacturing environments, optical profilometry may not be the first traditional choice due to measurement sensitivity to instrument vibrations. To address this sensitivity a commercial scanning white light interferometer (or SWLI) has been introduced that promotes increased vibration tolerance (the Zygo ZeGage). In this work, the SWLI's mechanical response to forced vibration was evaluated using a frequency sweep technique. Replicated nickel surface roughness standards were measured while the SWLI platform was subjected to forced vibration from a modal shaker. Surface topography responses to vibration frequency and amplitude were found to correspond with the mechanical resonances of the instrument structure. Additionally, the influence of environmental disturbance due to an operating machine tool was evaluated. The degradation of the measurement result for optical-quality surfaces was detectable at lower vibration amplitudes than for conventionally-machined (rougher) surfaces. However, the nature of the measurement result degradation was similar.

PERFORMANCE EVALUATION
In previous work, reference surface topographies measured by the SWLI in a manufacturing environment and with a stylus profilometer in a metrology lab were found to agree within the slope range of the SWLI [1]. Mechanical frequency response functions for the SWLI were determined and surface topography response to forced vibration was investigated.

Response to Milling Machine Vibration
The SWLI was placed on a table in close proximity to a horizontal milling machine during machining. Measurements of etched silicon and machined aluminum surfaces were performed. The amount of data loss was found to vary inversely with magnification. Conventionally machined surfaces exhibited less data loss than optical-grade surfaces.

Instrument Structural Evaluation
The SWLI was placed on a rolling hydraulic lift cart which provides some vibration isolation from the operating environment. A modal shaker was mounted to the cart base to provide an input force and low-mass accelerometers were used to measure the corresponding vibration. Direct and cross frequency response functions were measured from the shaker to the cart top (where the SWLI was mounted), the SWLI sample table, and the SWLI optical column, in response to nominally vertical and horizontal forced vibrations at the cart base. Example results for the response of the optical column to nominally vertical cart base input are displayed in Figure 1.

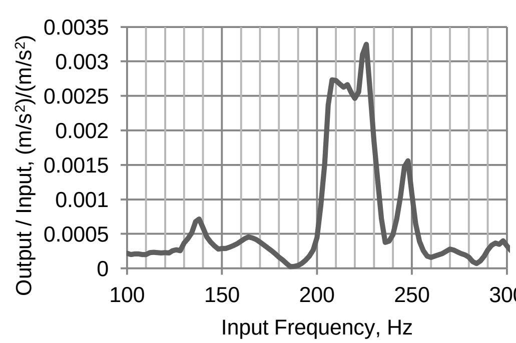

FIGURE 1. Vertical response of SWLI optical column to vertical modal shaker input.

Response to Forced Vibration

Using the modal shaker mounted on the cart base, a sinusoidal vibration with constant 500 nm amplitude at the cart top was applied. Deviation in the measured areal arithmetic average roughness, Sa, over the frequency range is shown in Figure 2. These Sa variations appear to correspond to the frequency response characteristics of the SWLI optical column.

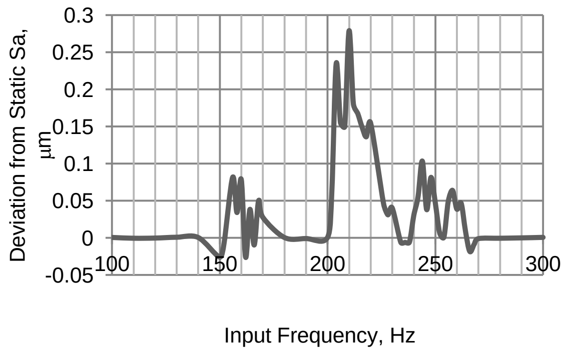

Figure 2: Sa deviation from static, 0.04µm Sa reference surface with 50x Mirau objective.

For frequencies near a mechanical resonance, the surface topography shows peaks and valleys parallel to the interference fringes, occurring at twice the fringe frequency. Figures 3 and 4 illustrate fringes and surface topography with forced vibration near a mechanical resonance. Slight data loss is visible in some of the surface “valleys” at regions of high local slope. Similar phenomena were exhibited by rougher reference samples. Using the modal shaker to force constant-amplitude vibrations to the instrument as a whole, as well as constant amplitude vibration of the optical column and sample table, amplitude and spatial parameters of surfaces were investigated.

SUMMARY

Through modal testing, mechanical resonances in the SWLI instrument were located. Forced vibration of the SWLI indicated maximum deviation from static surface roughness measurements near resonances, exhibiting a surface topography with peaks and valleys at twice the interference fringe frequency. This effect was found to be less obvious with surfaces of increasing roughness. By forcing constant amplitude vibration to different components of the instrument, the effect of vibration at each location on surface measurement was investigated. Classification schemes beyond Sa were applied to the resulting measurements.

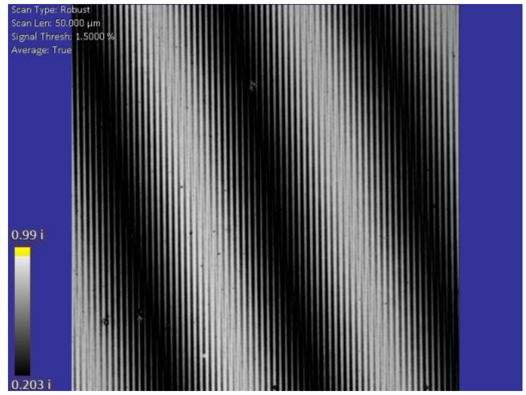

Figure 3: Interference fringes at static condition.

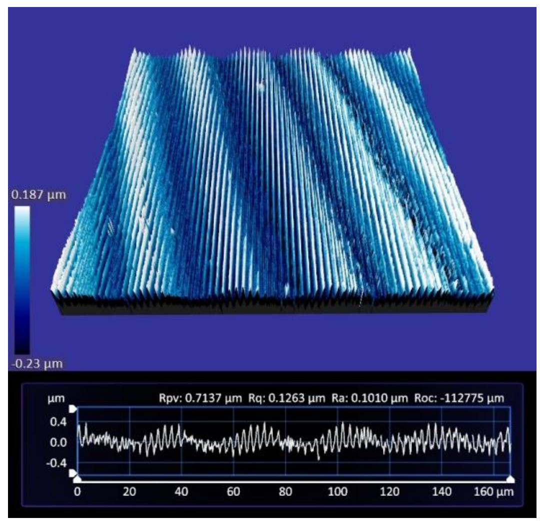

Figure 4: Indicated surface topography at 210 Hz, 500 nm vibration amplitude at cart top.

REFERENCES

[1] Badami V, Liesener J, Evans C, de Groot P. Evaluation of the measurement performance of a coherence scanning microscope using roughness specimens. Proceedings of ASPE Annual Meeting. Denver, 2011.