INTRODUCTION

Precision mechanisms often use flexure based hinges for their deterministic behavior and the absence of friction, stiction and backlash. Motion in vacuum is easily enabled without contamination by particles. However, the initially high support stiffness decreases rapidly with deflection. Therefore, sound conceptual designs need to be optimized for stiffness and eigenfrequency over the range of motion with limited stress, actuation stiffness and in a confined volume. The dimensions of the flexures will strongly affect the occurring stresses and performance, making it complex and time-consuming to find the optimal parameters. Chen and Howell [1] proposed an analytical method with thickness-to-width ratio independent equations, but this method is only applied to simple notch hinges. With more complex shapes or flexure based hinges such as the Cross-spring Pivot [2], here called Three Flexure Cross Hinge, TFCH, a numerical method with which a parametric model is optimized is preferred. Boer et al. [3] use a multibody modeling approach, applied to the Curved Hinge Flexure, with which accurate results are obtained for the stiffness characteristics and maximum occurring stresses. Wiersma et al. [4] proposed a method to investigate the performance of several flexure hinge types for large deflection. The new Infinity Flexure (∞-FH) showed the highest first unwanted eigenfrequency. However, only one load case with a large moment of inertia and a single optimization criterion, the first unwanted eigenfrequency, were used.

This paper extends the method by researching the influence of a different load case, two optimization criterions and the influence of optimizing for different ranges of motion. The TFCH and the ∞-FH, the flexures that showed the best performance [4], are compared.

DESIGN AND OPTIMIZATION METHOD

The parameterized geometries of the TFCH and ∞-FH are shown in Figures 1 and 2 respectively.

The requirements, constraints and load case are based on a specific electron microscopy case. The load attached to the hinge is a rigid body with a mass of 100 g, with associated inertia. The orientation of the hinge with respect to the load is used as an optimization parameter.

Firstly, the hinges are uni-directionally optimized for stiffness in z-direction $c_z$ at $R = 50$ mm from the pivot point for the deflection angles $+5.7^\circ$ and $+20^\circ$. Secondly, for the first unwanted eigenfrequency the hinges are optimized for bi-directional ranges of motion of $\pm5.7^\circ$ and $\pm20^\circ$. The compliant actuation direction causes the lowest eigenfrequency. Therefore, the first unwanted eigenfrequency is $f_2$. Irrespective the optimization criterion, $c_z$ or $f_2$, all hinges need to satisfy the constraint that a minimum value of 300 Hz is required for $f_2$ at the maximum deflection. To make $c_z$ and $f_2$ comparable for the
two different ranges of motion, all hinge designs are constrained to the max stress level at ±20°. Furthermore, the actuation stiffness and Von Mises stress σ are constrained at 26 Nm/rad and 600 MPa respectively. The hinge height is fixed at \( H = 54 \) mm.

The models are based on a flexible multibody approach with non-linear finite beam elements such that it can capture the geometric non-linear behavior with a small number of elements and relatively low computation times.

RESULTS
The optimal geometries for the TFCH and \( \infty \)-FH are determined for the two criterions. Figure 3 shows the support stiffness \( c_z \) at \( R = 50 \) mm from the pivot point and Figure 4 shows the second eigenfrequency \( f_2 \) as function of the deflection angle for eight optimized geometries. The solid lines correspond with stiffness optimizations, whereas the dashed lines correspond with eigenfrequency optimizations. The effect of this varying stroke is clear when the stiffness of the \( \infty \)-FH is analyzed, see Fig. 3. The 20°-optimization yields a higher stiffness for deflections above about 11°, while below this intersection a higher stiffness is obtained with the 5.7°-optimization.

The stiffness obtained with the \( \infty \)-FH is higher than for the TFCH, especially at larger deflection angles. This matches the results shown in [4]. The internal frequencies of the \( \infty \)-FH are however significantly lower than for the TFCH. Since the mass and inertia of the rigid body attached to the hinges is relatively small, the first unwanted modes are internal modes of the hinges instead of the rigid body modes.

CONCLUSION
The \( \infty \)-FH improves the stiffness of the system, but the internal eigenfrequencies are lower than for the TFCH, causing a lower second eigenfrequency when the hinges are not subjected to a large load. The influence of the optimization criterion on the optimal geometry is significant, especially for the \( \infty \)-FH. The asymmetric shape of the stiffness lines shows that this hinge can be tuned to a specific load case very well. The TFCH is more symmetric and hence less sensitive to the optimization criterion.

These results will be verified by measurements on the prototype that is currently in production.

REFERENCES