Numerical simulation and statistical analysis of a cascaded flexure hinge for use in a cryogenic working environment

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Abstract. Due to their many advantages, flexible structures are increasingly being used as guide and transmission elements in handling systems. Prismatic solid-state joints with a concentrated cross-sectional reduction are predominantly used as flexure pivots for both microscopic and macroscopic designs. A transfer of these geometries to applications in cryogenic working environments is not easily possible at temperatures below -130 °C due to the changed material properties. In this paper, the further development of swivel joints as cascaded solid state joints for such a cryogenic environment is illustrated by the targeted adaptation of certain joint parameters and dimensions. By means of a comprehensive FEM simulation, it can be shown how the influence of specific parameters affects movement accuracy, process forces and shape stability and to what extent these geometric parameters influence each other in their effect.

Keywords: compliant mechanisms, flexure hinges, cryogenic workspace.

1 Introduction

In today's industry, automation is an omnipresent factor or will be, even in niche applications such as cryopreservation. Nevertheless, Manual handling of biological or toxic samples is still the rule in research facilities. Such samples are conserved in so-called biobanks. In their simplest form, these are Dewar vessels are cooled with liquid nitrogen. However, large semi-automated systems are also used, mainly for an indefinite storage time. Even in such systems, the samples are still stored, rearranged or transferred by hand using bulky protective clothing and equipment. Doing so involves a significant risk of injury due to cold burns for the users on the one hand. On the other hand, the sample integrity can be compromised by any kind of heat exposure during the handling process. To overcome the dangers of manually handling test tubes while circumventing the technical challenges a cryogenic work environment presents, a full automation is desirable. The logical consequence is to develop a robot based handling system.

As part of the DFG funded project “Methods for the automation of handling processes under cryogenic environmental conditions”, a parallel robot is being developed at the Institute of Assembly Technology at Leibniz Universität Hannover to implement such full automation. Especially the passive joints of the robot have to meet very high

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demands: The extreme temperatures of below -130 °C do not allow the use of classic rigid body systems such as ball joints due to the freezing of lubricants or the clamping of components due to cold shrinkage. To avoid these disadvantages, solid-state joints in the form of cohesive swivel joints are used. Due to their monolithic construction, clamping is not an issue since no parts moving against each other are involved. In addition, the use of lubricants is also unnecessary.

Due to these and other advantages, joints with concentrated resilience designed as monolithic coupling mechanisms have already become widely used in technical and industrial handling technology. In these one-piece constructions, an inherent compliance is achieved exclusively by reversible deformation of certain areas. Thus, these mechanisms can be used for small angular deflections or displacements instead of conventional form- or force-paired pivot hinges. The most common application to date has been prismatic solid state joints with areas of reduced cross sections as bending points. These mechanisms are mainly found in precision systems in precision mechanics and metrology as well as in applications in precision assembly, i.e. areas in which high demands are placed on positioning accuracy and resolution [1–4]. Applications in macroscopic constructions and for big angular deflections are still the exception.

Solid-state joints are material-coherent mechanisms and their ability to move is based on the deformation of the entire joint or on the deformation of areas with a concentrated reduction in cross-section. Therefore, their bending range is severely limited to a few degrees by the maximum allowable stresses in the component. In addition, a rotational movement of a material-locking joint always causes a load-dependent displacement of its axis of rotation, which has a negative effect on the positioning accuracy of the mechanism. It is therefore always desirable to achieve the smallest possible ratio of maximum stress to the rotation achieved, while at the same time minimizing the displacement of the axis of rotation. To achieve this, increasingly complex geometries have been developed in the recent past, or an increasing number of simple solid-state joints have been installed in a common structure.

1.1 Related work

In contrast to cohesive joints with only one area of concentrated flexibility or with multi-unit flexible coupling mechanisms, the structural and dimensional synthesis of complex flexure pivots has not yet been discussed in detail. The literature mainly describes constructions with concentrated cross-section reductions in rectangular or circular contours, as well as the effect of these contours on the achievable range of motion [5–17]. There is also work on the geometric design of complex swivel joints on the basis of interlinked tape spring joints [18], which allow large angular deflections. The relevant influences on the geometric design of such joints with regard to their ability to move along the main axes and parasitic axes have hardly been investigated. The project ”Methods for the automation of handling processes under cryogenic environmental conditions”, however, was preceded by the work of Raatz et al. [19–21]., in which the use of cohesive joints in parallel robot structures and the kinetic and kinematic behavior of these joints were examined in detail. The use of solid-state joints in a cryogenic working environment has not yet been investigated. The work on the use of flexible

joints for use in aerospace engineering has been described [3, 4, 22], but not for the environmental conditions envisaged in this project.

Following on from the work of Robert Fowler [4] and Simon Heinein [22], in this paper the finite element method (FEM) and the ANSYS program will be used to present a joint geometry that is capable of achieving a required angle of rotation in a cryogenic working environment. The maximum permissible stresses within the components are never allowed to exceed a certain value, so that the deformation of the joint is completely reversible at all times. In addition, by varying certain design parameters, their influence on the necessary process forces, the undesired deformation along certain axes (henceforth called parasitic axes) and the effect of these parameters on each other will be investigated.

1.2 Materials and methods

The aim of this work is the further development of cascaded solid state joints based on the work of Robert Fowler [4], who in turn bases his work on the Butterfly Pivots by Heinein et al [22, 23]. Especially the Flex-16 large-displacement compliant hinge offers a good starting point for the development and investigation of own joint geometries (Fig.1). Due to the continuum mechanical problem, a sequential approach of concept synthesis and constructive development of the design is not easily transferable to the existing compliant hinges. The displacement of the axis of rotation allows only limited formation of a replacement model based on rigid state mechanisms. In addition to the desired rotational motion, a force-deformation characteristic along the main and parasitic axes as well as the shape stability must also be taken into account in the structural development of compliant mechanisms. The procedure here follows the idea of a topology-optimized structure. The relevant load paths and most stressed points of a prototype are identified. Then the material distribution is varied within the framework of certain constructive and production-related restrictions and the change of the load cases is evaluated afterwards. In particular, it will be investigated whether the motion and deformation behavior can be represented as a function of the geometry parameters and whether there are interdependencies between the geometry parameters with respect to their influence on the motion and deformation behavior. For this purpose, four criteria are to be considered qualitatively and quantitatively as result variables:

- The maximum von Mises stresses occurring in the critical areas when a defined bending angle is reached
- The deviation of the rotary axis from the initial position
- The deformation of the entire joint along two (parasitic) axes perpendicular to the axis of rotation.
- The forces required to reach the defined bending angle

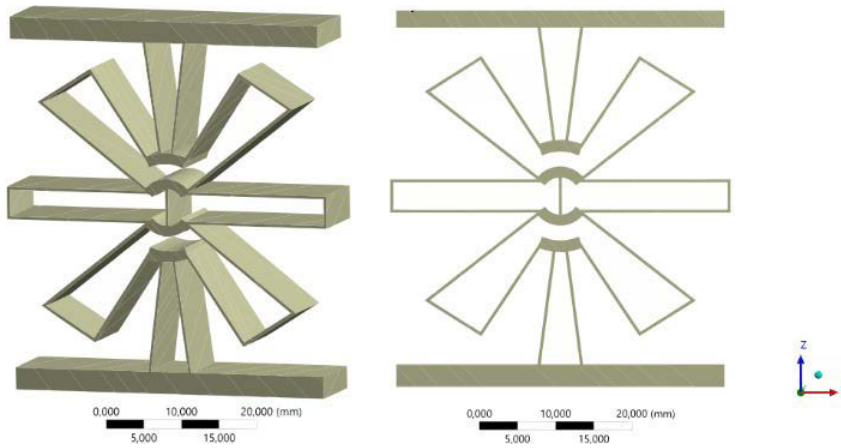

Fig. 1. Prototype Geometry based on Robert Fowlers Flex-16 large-displacement compliant hinge

The simulation of the joints in ANSYS is to be carried out under the conditions typical for a cryogenic working environment, taking into account the changed material properties of the joint material at temperatures below -130 °C. However, since these temperatures are reached in a biobank by cooling with liquid nitrogen, the components undergo the danger of being exposed to much lower temperatures. For the simulations, the adjusted material parameters Young's Modulus and transverse contraction number are used from tabulated sources [24, 25] for a temperature of -196 °C, which corresponds to the boiling temperature of liquid nitrogen and includes the case that the joints are immersed in the liquid.

2 Prototype geometry

The starting point of the joint construction is the Flex-16 joint, which was presented in multiple variations in [4]. After initial estimations and simulations, the overall dimension was fixed at 50 x 50 x 10 mm³. This limitation was chosen because of the space available in the final construction of the parallel robot, which will be examined in more detail in other works. For a better understanding, a design of this robot and a Cardan joint consisting of six flexure pivots are shown in Fig. 2. In [4] it has already been shown, that this flexible mechanism is able to achieve a bending angle of 90° at room temperature, if titanium was used as the base material.

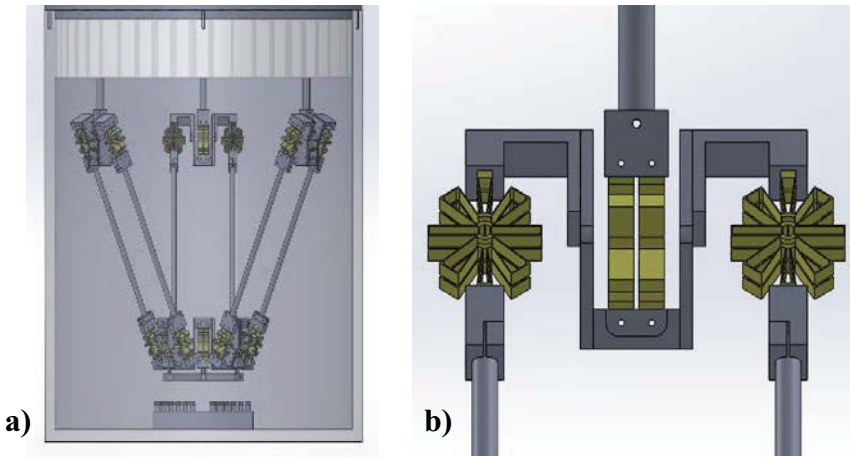

Fig. 2. a) Parallel robot structure and b) Cardan joint

2.1 Simulations of the prototype geometry

The rotational movement of the joint in the ANSYS simulation is realized by fixing the joint at one end and applying various forces at the other end to investigate all possible load cases. Care has been taken to ensure that the maximum equivalent stresses do not exceed the maximum yield strength that the material can withstand under cryogenic conditions. This is made possible by the fact that the bending angle to be achieved for the entire joint is divided between the 16 serially linked thin sections of the joint. The simulation results of the bending are shown in Fig. 3 a). From this preliminary examination, the most heavily loaded or most heavily deformed sections were determined. As can be seen in Fig. 3a), the sections 3 and 4 are most strongly involved in the rotational movement about the intended bending axis. In Fig. 3 b), the sections 1 and 2 show the strongest local deformation in positional deviation along the parasitic axes under a tensile load.

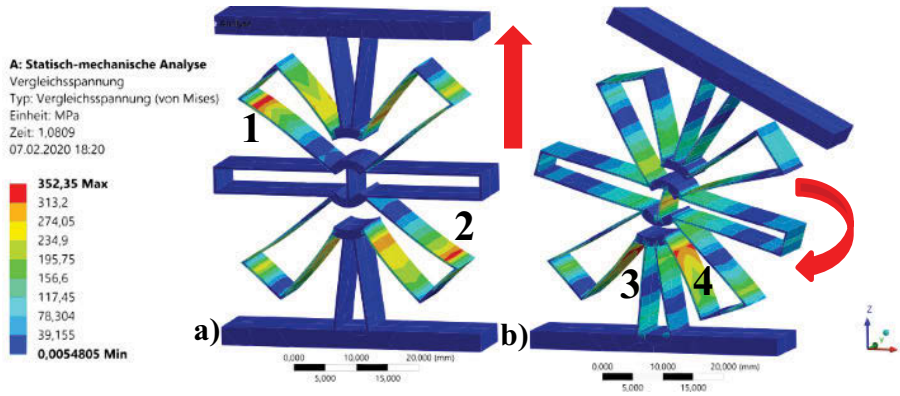

Fig. 3. Flex 16 under a) tensile load and b) bending load

While the bending movement around the axis of rotation should be achieved with as little process force as possible, it is desirable to stiffen the joint in the direction of the parasitic axes as much as possible by adjusting the geometry. The geometry parameters described in the following and their effects on the result variables mentioned in section 1.2 have particularly this stiffening as their goal. After several simulations with different materials, the choice of material falls on titanium. The minimum 20° bending angle of the joints required for robot design could not be achieved with any other material within the limited installation space.

3 Development of the joint geometry

The flexure joint developed here will later be manufactured from the titanium alloy TiAl6V4 in a laser sintering process. This choice was made, because it is by far the most common titanium alloy available for industrial additive manufacturing processes. The hinge consists of 16 individual thin sections arranged around a central connection point. Four areas with a comparatively large cross-section - so-called shuttles - are arranged around the fulcrum to ensure axial stability. The thin sections form six "wings".

The first geometric parameter to be varied is the thickness of the material. It is expected that an enlargement of the cross-section *h* (see Fig. 4b) will lead to a stiffening of the joint. The extent of this stiffening is to be investigated for a range between *h*=0.8 mm and *h*=1.2 mm in 0.1 mm steps. The second parameter describes the shortening of the thin spots called wings. Here it is assumed that a shortening of the wings will lead to a stiffening of the joint. The wing proportions are shown in here Fig. 4c). Their length *l* is varied between *l*=15 mm and *l*=5 mm in steps of 2 mm. The stiffening by an enlargement of the cross section and the shortening of the thin sections mainly involved in the bending are logical steps. The last geometry parameter to be varied is derived from the results of the prototype geometry from Fig. 3b. As can be seen, the upward bending of the wings seems to be the main reason for the deformation in the y-direction. To minimize this deformation, small corner shuttles are provided in

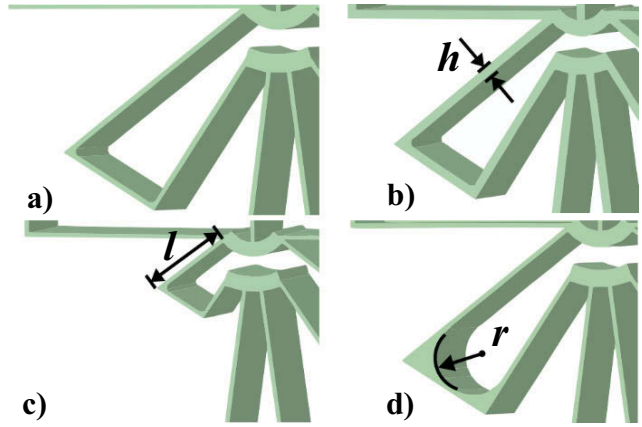

Fig. 4. a) Prototype proportions, b) widened cross-section *h*, c) shortened wing length *l*, d) increased radius for corner shuttles *r*

the wings, whose radius r is varied between r=0.4 mm and r=3.2 mm in steps of 0.2 mm (see Fig. 4d). The minimum variation of the geometries is based on the manufacturing parameters of a laser sintering process, which can generally be guaranteed by the manufacturers [26, 27]. The number of possible parameter combinations resulting from this is two hundred and twenty-two simulations.

Depending on the selected geometric proportions, different bending angles can be achieved before the deformation causes critical stresses in the components. Since a minimum angle of twenty degrees for each solid state joint is sufficient for the parallel robot structure to be realized, it was checked whether all joint variations can represent this angle, and if so, which maximum angle can be achieved by all joints. The preliminary investigations showed that the smallest bending angle that can be achieved by elastic deformation of all joints is 27.75 degrees. This angle was assumed as a fixed value for the simulations. Thus, the necessary applied bending force as well as the maximum von Mises stresses occurring can be used as target values. In addition to the bending, the joints are also loaded with 16 N for tension in the y-direction and with 16 N for shear in the x-direction in order to investigate a deformation along the parasitic axes. The target values for these load cases are the deviations from the initial position along these axes.

3.1 Simulation of the joint geometry

The basic geometry of the flexure pivot based on the Fowler model [4] is created with solid works. The geometry parameters to be adjusted are then inserted and the model parameterized in ANSYS. The mesh quality of the model significantly influences the calculated stiffness in the simulation [28]. In order to find a compromise between the necessary simulation time and sufficient accuracy, the number of nodes used per model is limited to 8200. This number resulted from the specification that every element has an edge length of no more than two times the material thickness. The aim was a multi-zone mesh. Areas of the solid state joint that have a simple geometry (mainly cuboids or circular sections) can be swept and thus meshed very evenly. For the areas where these joint sections meet, a finer mesh structure has been strived for. Here a hex-dominant mesh was preferred, if possible. The use of linear tetrahedrals was avoided as this element class usually implies too high a stiffness. The skewness, which should not exceed the value 0.5, was considered as a measure for the quality of the mesh. These restrictions were met for all simulated joints. Due to the quality of the net, it was possible to forgo center nodes, which can also cause excessive stiffening of the simulated component [28]. A Sparse Matrix direct solver was used with 40 substeps and a minimum increment of 0.51×10^{-4} for the convergence criteria (CC).

4 Graphical evaluation of the results

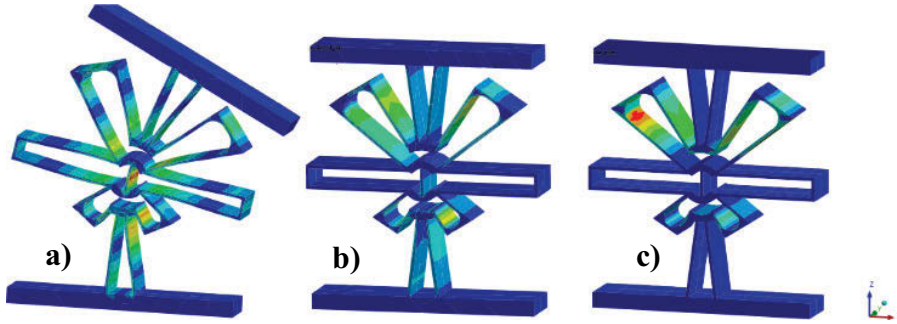

Fig. 5. Deflection of the flexure pivot und a) bending load, b) shear load, c) tensile load

Two hundred and twenty-two simulations were carried out in this way. Based on this, a comprehensive examination and analysis of the data was carried out. Fig.5 shows an example from the simulation with the corresponding images of the deformed joint under the given load cases. The consideration of the target values (see section 1.2) shows a trend in the effect of the varied geometry parameters. Fig. 6 shows the maximum stresses as an example.

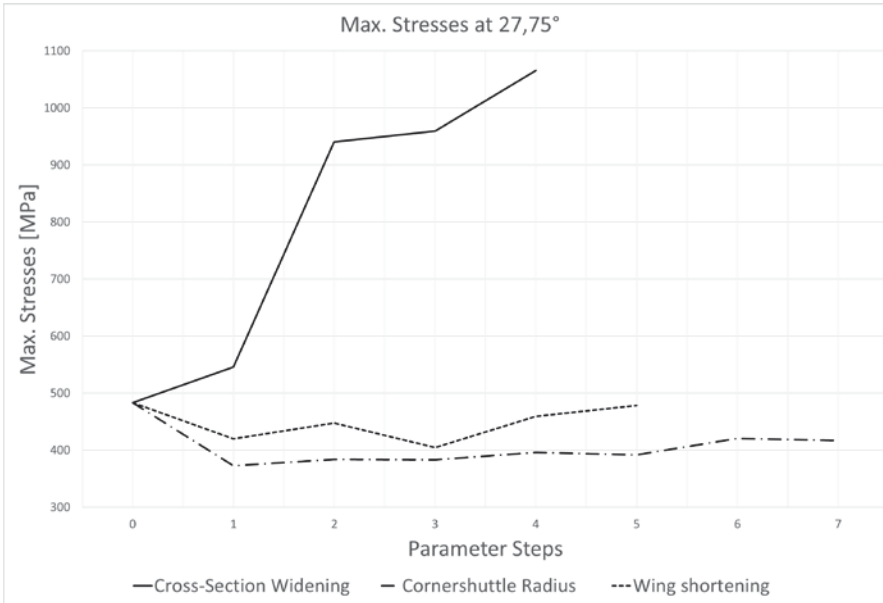

Fig. 6. Maximum stresses, sorted by the varied geometric Parameters

The course supports the assumption that the geometric parameters are not independent of each other with respect to the effect they have on the target variables. It can also be seen that the effect of the changed parameters material thickness, wing length and corner shuttle have a different influence on these target values. The widening of the cross-

section has by far the strongest influence on our target values, whereby the increase of the necessary bending forces is rated as negative, and the reduction of the deflection of the joints along the parasitic axes is rated as positive. Analogous conclusions result for the wing length and the radius of the corner shuttles in descending significance. Here, too, it can be seen that the effect of the parameter change is not linear, which again supports the assumption that the geometric parameters are not independent of each other. In order to investigate this in more detail, the results of the simulations were divided into different classes. The simulated feature carriers are first divided into six classes sorted according to the cross-section thickness and then the target values are displayed as a function of the wing shortening. The corner shuttles are set to a radius of 0,4 mm. As can be seen in Fig 7, the graphs do not run parallel to each other. Analogous to this procedure, six subclasses are formed for the wing length and then the target values are displayed as a function of the radius of the corner shuttles. This time the material thickness is set to 1 mm. The result forms a similar picture as in the previous consideration. The course of the curves is not constant between the subclasses.

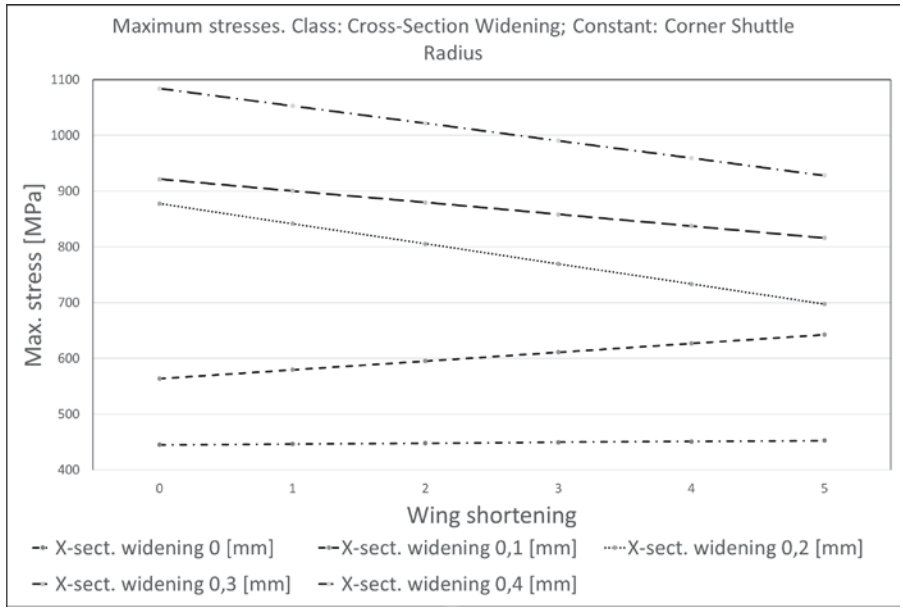

Fig. 7. Class comparison: cross-section widening

5 Analysis of the results

The graphical evaluation of the results suggests that there is no linear correlation between the target values and the individual geometry parameters. For a multivariate analysis approach, a regression analysis based on a polynomial is suitable. Based on Fig.7 however, it is assumed that the parameters themselves depend on each other in their effect on the target parameters and therefore no linear regression model can be used as

a basis. At this point, however, qualitative statements about the influence of the geometry parameters on the target variables can already be made. These are summarized below. A further procedure on the basis of the obtained data is discussed in the outlook. Based on the results obtained so far, the following qualitative correlations can be concluded:

- The three selected geometry parameters material thickness, wing length and corner shuttle radius all have a negative influence on the target parameter "bending force" and a positive influence on the target parameter "deviation along the parasitic axes".
- The significance of the influence varies between the geometry parameters
- The geometry parameters are not independent of each other. They influence each other in their effect on the target parameters.
- It appears that the decrease in wing length and the increase of the corner shuttles both have a positive influence on the maximum stresses. The reasons for this are not yet clear, but an assumption can be made, that this is the result of a better overall distribution of the maximum stresses throughout the flexure hinge.

Because of the data obtained, the following quantitative statements can be made, even if the formula-based relationship is still unknown. Fig.8 sums the results up graphically:

- A cross-section widening of 0.4 mm increases the necessary bending forces for reaching the defined angle by 711 % from 17.7 N to 127,26 N. The position deviation in the x-direction due to the shear load is reduced by 85% from 1.39 mm to 0.20 mm. Similarly, the position deviation in the y-direction due to the tensile load is reduced by 87% from 0.89 to 0.12 mm. The maximum stresses are increased by 121%, from 482 N/mm to 1065 N/mm.
- Shortening the wings by 10 mm increases the bending forces required to achieve the defined angle by 42 % from 17.7 N to 25,08 N. The position deviation in the x-direction due to the shear load is reduced by 45% from 1.39 mm to 0.76 mm. Similarly, the position deviation in the y-direction due to the tensile load is reduced by 46% from 0.89 mm to 0.48 mm. The maximum stresses are decreased by 1%, from 482 N/mm to 478 N/mm.
- Increasing the radius of the corner shuttle by 2.8 mm increases the bending forces required to achieve the defined angle by 20 % from 17.17 N to 21.25 N. The position deviation in the x-direction due to the shear load is reduced by 30% from 1.39 mm to 0.96 mm. Similarly, the position deviation in the y-direction due to the tensile load is reduced by 42% from 0.89 mm to 0.52 mm. The maximum stresses are decreased by 14%, from 482 N/mm to 416 N/mm.
- In the extreme case that all geometry parameters are set to the maximum of their values, the necessary bending forces for reaching the defined angle increases by 1052 % from 17.7 N to 204 N The position deviation in the x-direction due to the shear load is reduced by 94% from 1.39 to 0,09 mm. Similarly, the position deviation in the y-direction due to the tensile load is reduced by 95% from 0.89 to 0.05 mm. The maximum stresses are increased by 91%, from 482 N/mm to 919 N/mm.

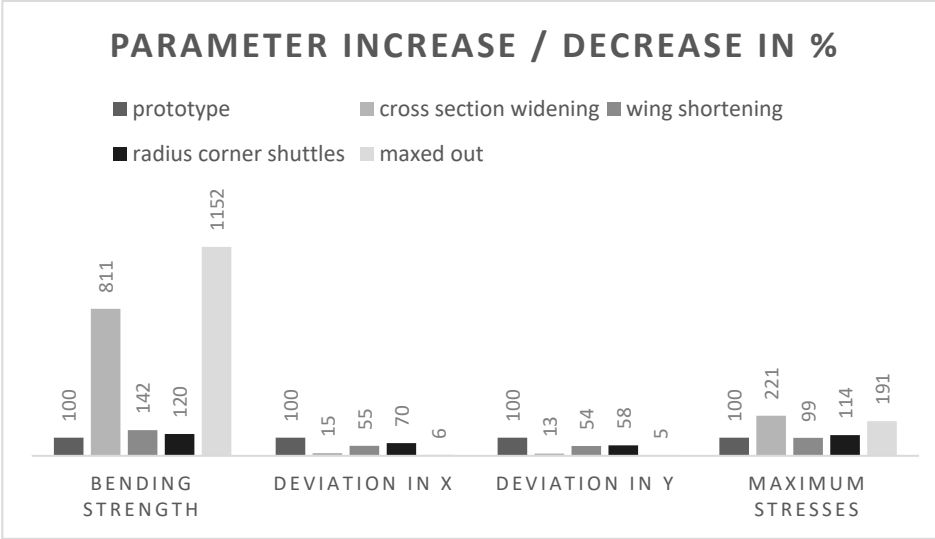

Fig. 8. Summary of the results

6 Conclusion and outlook

Due to the assumed non-linearity and lack of independence of the varied geometry parameters from each other, no formula-based procedure for the optimal design of the flexure pivot described here can be proposed at present. However, it could be shown that the geometric adaptation of the joint allows a significant reduction of deviations from the nominal position without limiting the functionality or exceeding the critical comparative stresses. In the following, further analyses of the existing database will be carried out. This includes non-linear regression analyses as well as optimization strategies. In addition, a characteristic value is to be developed which describes the usefulness of the geometric adaptations as a ratio of the positive and negative effects. If this procedure proves to be promising, further joint geometries will be created and evaluated.

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