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A GEOMETRIC APPROACH TO TOLERANCE ANALYSIS: CONTRIBUTION TO THE ROBUST DESIGN OF FLEXIBLE ASSEMBLIES

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Abstract

Tolerance analysis is an important element for virtual product development. It makes it possible to record the effects of permissible deviations for the complex assemblies. The component deviations will only be represented by a significantly simplified structure in commercially available 3D tolerance analysis programs and elastic deformations in the components, as well as complex contact regions, can only be partially represented. The objective of this paper is therefore to present a geometric-based approach for tolerance analysis. This already represents the tolerance-related deviations for components in a 3D data set as realistic imperfections. The assertion, that the geometric imperfections have a significant impact, will be verified on the basis of a bolted articulated joint assembly. The numerical results will be subsequently validated experimentally. A test bench will be designed for this purpose and various test sequences with realistic bolted articulated joints will be examined. The results indicate the relevance of realistic consideration of tolerances in product engineering processes and therefore create an element for the robust design of flexible assemblies.

Keywords: Robust design, Computational design methods, Tolerance representation and management

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1 INTRODUCTION

All the components we can produce in today's world are not ideal! We need to take this into account for the engineering design process of new products. The engineer defines nominal dimensions for the geometrical representation of a new product with computer aided design software (CAD). But to gain control of the imperfections of the manufacturing and assembling process, we need limits for each geometric dimension of each single work piece. The geometrical dimensions need to be inside these limits, which are also known as tolerances.
State of the art structural analysis workflows makes it possible to use the nominal CAD geometry with changes of the dimensions inside the given tolerances (see Figure 1a and 1b) (ISO 17450-1, 2012). But the shape is still ideal, because the geometric tolerances are not taken into account in the CAD Model.

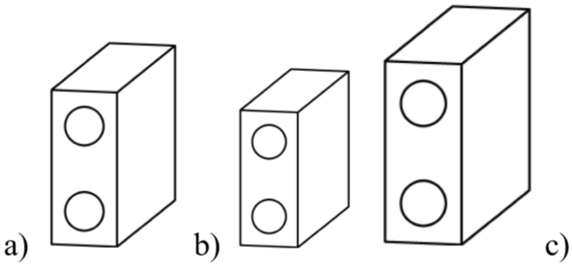


Figure 1. a) The nominal Geometry, b) The nominal geometry with maximum and minimum dimensional tolerances c) The non-ideal representation of a geometry, Source (ISO 17450-1, 2012)

In this paper the impact of realistic non-ideal geometry (see Figure 1c) data to the stress distribution and magnitude under loaded conditions is shown for a well-known machine element, an assembly containing three main parts (clevis joint with bolt linkage, see Figure 2). With a subsequently performed experimental investigation on a test rig, the numerical results shall be validated.

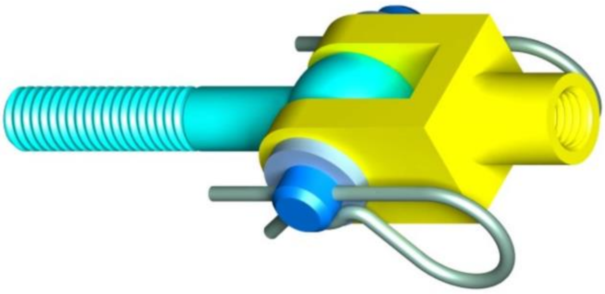


Figure 2. Machine Element: Clevis joint with bolt linkage

2 GEOMETRY BASED TOLERANCE ANALYSIS WITH NON-IDEAL PARTS

The developed workflow for the new approach of geometry based tolerance analysis is shown in Figure 3. In a first step the non-ideal geometry has to be generated directly according to the geometric product specifications (GPS) with a therefor developed software tool.
Afterwards the FE simulation model has to be defined and solved initially. With the suitable results as output-parameters a design of experiment (DoE) is performed to investigate the influence of the single tolerances regarding to the chosen output parameter of the model.
After a so called sensitivity study, a robust design optimization can follow as a last step, to ensure that the output parameters of the optimized model are as independent as possible due to variations of the input parameters (the given tolerances).

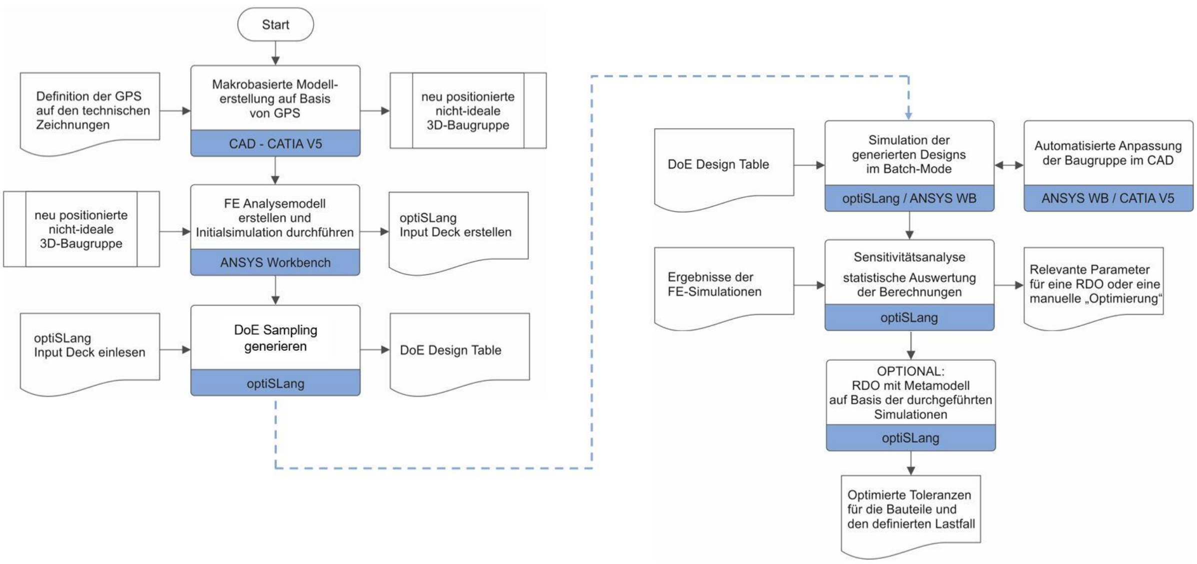


Figure 3. Flowchart of the new geometric tolerance analysis approach

2.1 Non-ideal model generation

To create the non-ideal CAD geometry an in-house tool was developed. In a graphical user interface (GUI) the parameters of the surface deviation for the description of the dimensional and shape tolerances could be entered. The visual basic (VB)-based tool is adapted to the commercially available CAD system CATIA V5.

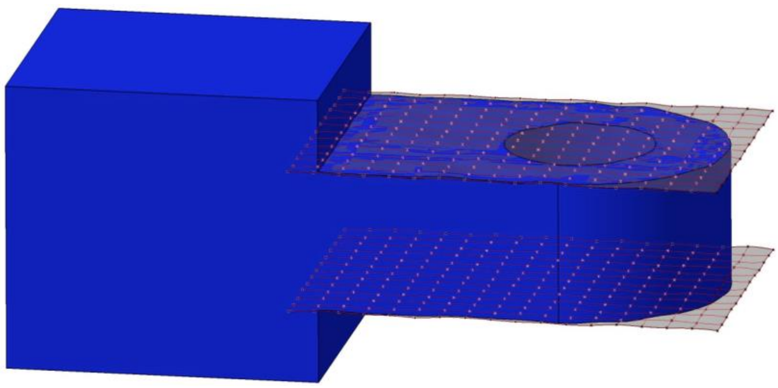


Figure 4. Surface generation for the non-ideal geometry

After the import of the parameters, the non-ideal CAD model is generated automatically according to the given tolerances. Figure 4 shows exemplary the way of the surface generation which is performed in several steps. First the datum is defined and afterwards a grid of points and curves. Than surfaces are approximated through this supporting wireframe which form the exterior of the part.

Continuous / realistic topology generation: In this context, the term "realistic" means that the virtual deviations from those of real-made components as close as possible.
The difference between two adjacent support points for the production of tolerated geometry elements must not be more than 20% of the permissible tolerance zone (measured in the normal direction to the limiting plane of the defined tolerance).
As a result, it is ensured that a continuous deformation of the component shape takes place (within the scope of the defined and thus permissible tolerances).
In Figure 5 the part with the number 2 shows how a non-ideal surface can form, in which this criterion is not applied. In the case of part 1, the geometry has been generated as described above and shows a realistic course.

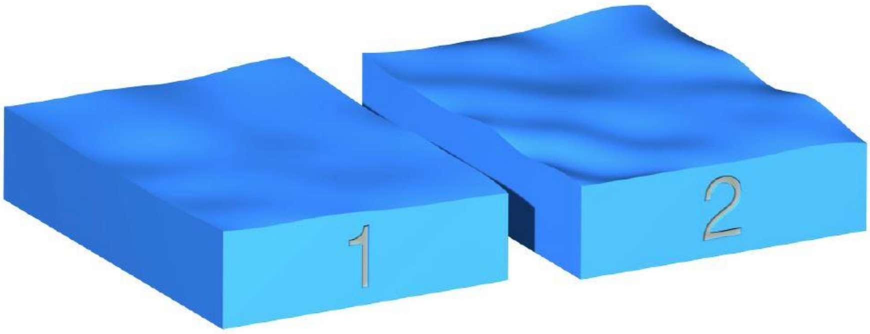


Figure 5. Variants for the creation of non-ideal surfaces (amplified)

Use of the specified tolerance range: The non-ideal geometries are generated by wire geometry models. These are based on approximation polynomials which extend through support points and can also exceed the permissible tolerance zones, depending on the position and arrangement of the support points.
To ensure that the basic geometry and the non-ideal surfaces generated below do not exceed the permissible tolerance ranges at any point, an empirically determined utilization factor is taken into account. This serves as a correction factor for the virtual tolerance zones and reduces the usable tolerance range for the support geometry. This ensures that the final geometry always lies within the defined tolerances.
A check of the generated geometry is then carried out for each component produced. For this purpose, references are generated in a first step on the basis of the non-ideal geometry, conforming to the specifications of the technical drawing.
The respective tolerances are then determined as actual values and converted into a data file. Designs that are not within the allowed tolerances are automatically discarded.

Table 1. Orders of geometric deviations

1.Order: Shape Non Uniformity		linear, planar and roundness Non-Uniformity
2.Order: Waviness		Waves
3.Order: Roughness		Groves
4.Order: Roughness		Scratchs, Scurfs, …
5.Order: Roughness	not presentable	Microstructur
6.Order	not presentable	Atomic Lattice

Only a geometric deviation of the first and second order is taken into account during the generation of the non-ideal geometry. Geometric deviation with higher order cannot be sufficiently considered because of the high difference between the dimensions of the roughness and the geometrical dimensions of the parts. In Table 1 an overview of the different orders of surface deviation is depicted.

2.2 Mesh generation

Before the stresses and deformations can be determined, the computational domain has to be discretized. Depending on the dimensions of the part and the values of deflection the element size has to be chosen small enough to represent the geometry with a high accuracy.
To avoid a too large number of elements a mesh with hexahedron elements was generated with a structured blocking. In Figure 6 is a stable and suitable mesh for the whole assembly depicted.

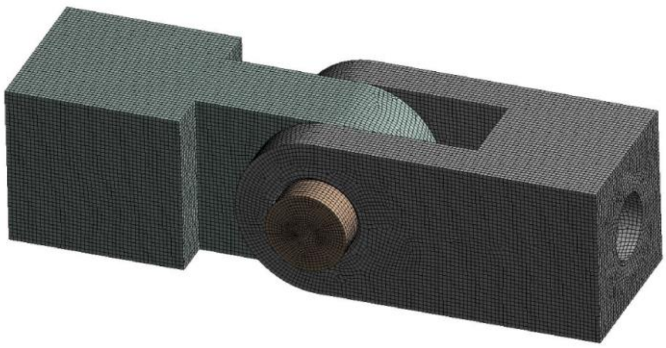


Figure 6. Mesh of the assembly

2.3 Boundary conditions

According to the shop floor drawings the non-ideal CAD geometry is generated. As a result we have three non-ideal shaped parts which are nevertheless valid when being compared to the drawings. The rod is loaded with a force F = 6,3 kN in axial direction. The bracket is fixed in all degrees of freedom at the flange surface. Contact regions are frictional with a friction coefficient of μ=0,1 (see Figure 7).
In addition to the non-ideal model a nominal model is generated as reference model which is treated with the same boundaries.

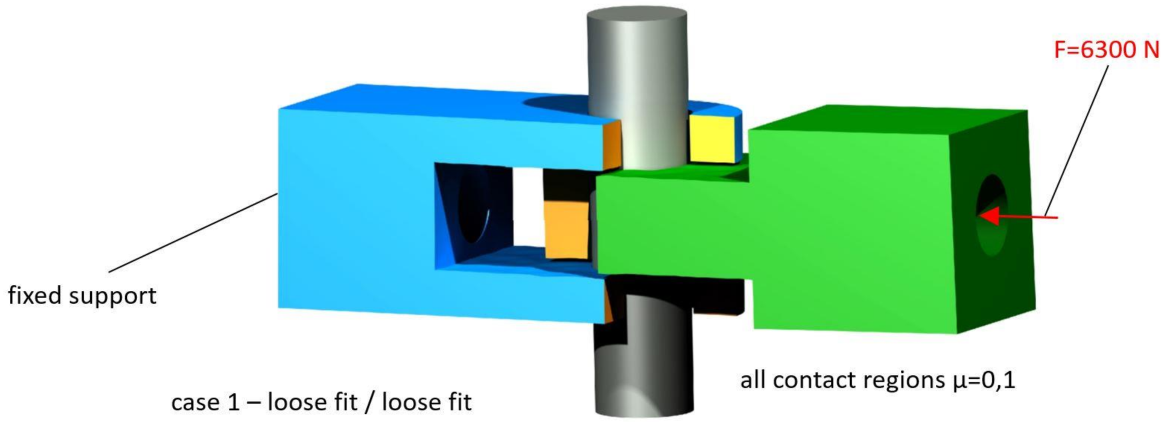


Figure 7. Boundary conditions for the simulation of the non-ideal assembly

2.4 Finite Element Analysis

In a first step the nominal simulation of the assembly was calculated. This reference simulation is based on the same boundary conditions with the same geometry like the simulation with non-ideal geometry. After the reference analysis the assembly with non-ideal geometry is calculated. The simulations are performed with the commercially available FEM package ANSYS v16 as a nonlinear simulation with plasticity of the material.

2.5 Design of Experiments and results

The design of experiment analysis is performed subsequently according to the flowchart of the new method. With respect to the number of tolerances a sample size of 120 was chosen to get statistically valid results.
The result of the sensitivity analysis is depicted on the left hand side of Figure 8 as the Coefficient of Importance (CoI) (Dynardo, 2015) and on the right hand side of Figure 8 as linear correlation coefficient. The most significant influence is assigned to the bolt diameter (parameter „dia_bolzen_DS“). This parameter with his negative value says, that for small diameter-values the output parameter rises, thus the stress in the bolt rises. Although important parameters are perpendicularity of the hole in the bracket (parameter „ort_bo2_stange_DS“) and the perpendicularity of the clevis joint flanks (parameter „ort_fluegel_gabel_DS“) together with the diameter of the hole in the bracket (parameter „dia_bo2_stange_DS“).

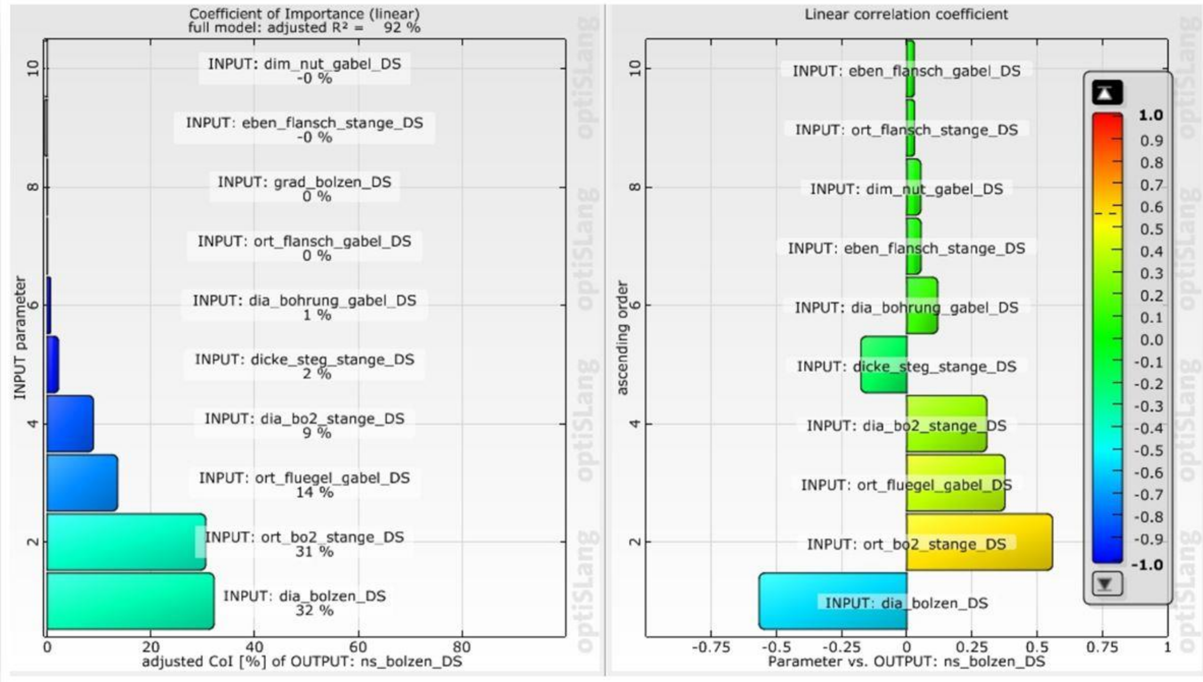


Figure 8. Result after 120 DoE runs with Latin Hypercube Sampling (Florian, 1992)

3 EXPERIMENTAL VALIDATION

In addition to the statistical analysis carried out, real assemblies with defined deviations are required for experimental validation. For this purpose, six different non-ideal variants of the assembly are generated and their tolerances and characteristics are documented in a simplified manner. These assemblies are investigated numerically and experimentally as a further step in order to create a validation option.

3.1 Numerical results

The analysed stresses in the bolts shown in Figure 9 show a significant variation. In particular the crack-induced flexural pull side of the bolt shows great differences between the individual variants. These are presented both in the intensity of the stresses and in the extent of the strain distributions. The saturation of the false colour representation provides as the beginning of the red range the average tensile strength of the material S235JR (Rennert et al., 2012)

These results are particularly relevant, because bending stress is often a failure-critical variable in bolt design.

The stress values shown in the table in Figure 9 illustrate the influence of the non-ideal geometry of the six test variants on the stress in the bolt. The percent deviations are shown in relation to the results of the ideal assembly.

On the basis of the results obtained in this chapter in addition to the defined workflow from Chapter 2, conclusions can be drawn from the material-specific Wöhler diagram for the expected load cycles of the clevis bolt joints. These evaluations and the validation of the simulations with the data from the life tests are presented in the next section.

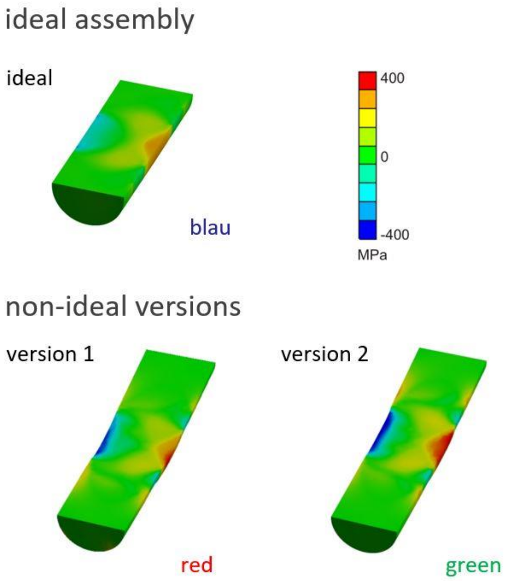
3.2 Experimental results

For the validation of the simulation models a suitable experimental setup is required. The final concept of the test bench provides a special hydraulic cylinder with a gas reservoir in order to apply the dynamic test load to the compounds to be tested. In Figure 10 the test stand is shown as a photo of the real laboratory setup.

The bolt test stand is designed with the purpose to determine the load cycles the bolt joints can endure for a swelling load case with a given load amplitude.

For the validation of the numerical simulation, "ideal" and "non-ideal" bolt joints are made within allowed tolerances and analysed on the test stand. In order to achieve an optimal comparison option despite a limited number of samples, the hardware tests were also classified into the six known variants.

Contrary to the ideal CAD model, it is not possible to create "ideal" geometries for real assemblies. For this reason, an "ideal" assembly is understood to mean that the components are manufactured according to drawing tolerances, which are at least a tens of poles smaller than those of the "non-ideal" assemblies.



Assembly variant	Stress in the bolt MPa	Deviation from the ideal variant %
Ideal	248	0
1	326	31
2	300	21
3	315	27
4	312	26
5	304	23
6	292	18

Figure 9. Comparison of the stresses in the different non-ideal assemblies

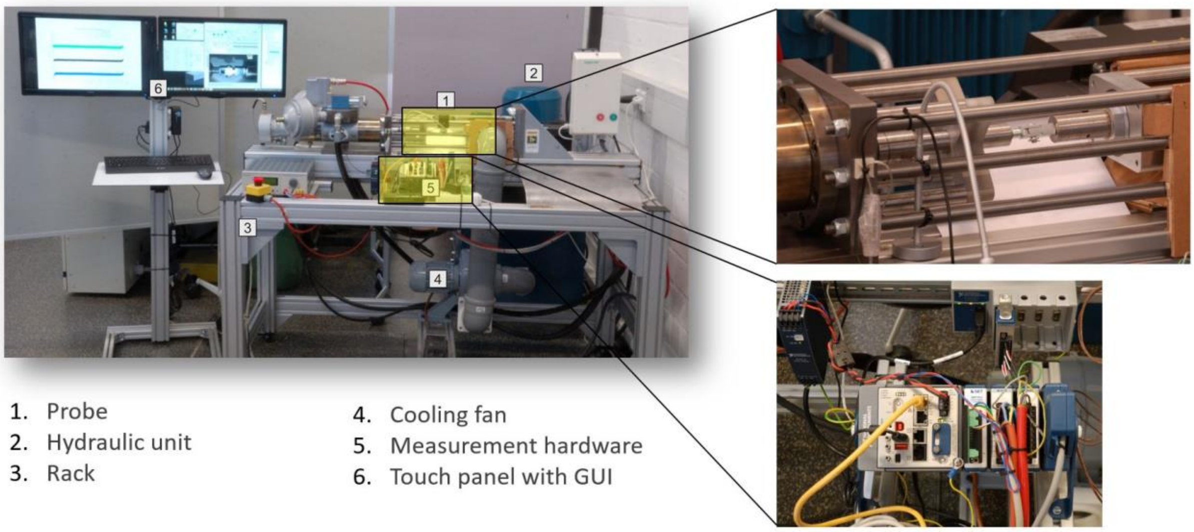


Figure 10. Overview of the experimental test rig

The results of the tests are displayed as load cycles "LW" in the graphic in Figure 10 in dependence on the upper force F2 = Fmax. The blue data series stands for the "ideal" assemblies of the connection and lies with their median (vertical bar) with 840.000 load cycles.

In comparison, two other "non-ideal" variants are defined and analysed. This is variant 1 (shown in red n Figure 9) and variant 2 (shown in green in Figure 9).

The vertical bars in the respective colours also represent the median of both variants.

It can be clearly seen that the influences of the tolerance-induced deviations on the bearable cycle number under load are significant.

The variant 1 is with the median at 115,000 load cycles and thus more than seven times below the cycle number of the "ideal" assembly.

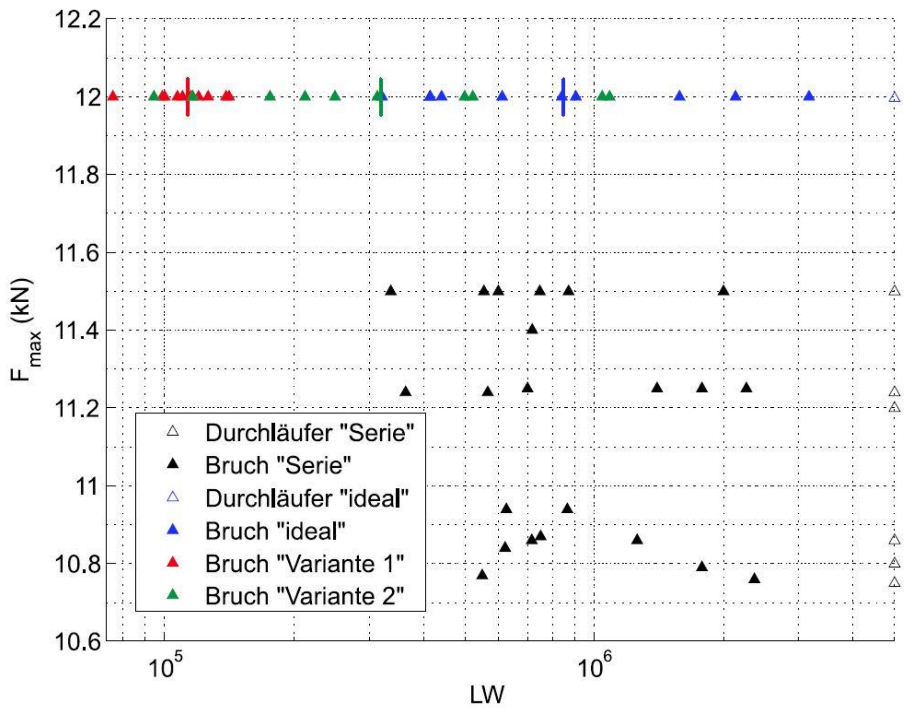


Figure 11. Experimental results after respectively ten samples per variant

Variant 2 is due to the defined tolerance values (dimensional tolerance of the bolt diameter at the upper tolerance limit) higher than the value of variant 1. However, variant 2 also has a pronounced influence of the non-ideal geometry. The median of the perceptible load cycles lies with 324,000 load changes more than 2.5 times below the value of the "ideal" variants.

4 SUMMARY AND CONLUSIONS

In this paper, an innovative methodology has been developed for geometrically-based tolerance analysis of non-ideal 3D assemblies. This includes a workflow and combines existing techniques with new approaches to the use of realistic, tolerated CAD geometry. The relevance of the methodology was illustrated by a practical example and the numerical results were confirmed by a subsequent validation by investigations with real assemblies.

In summary, it can be stated that the geometrically attributed consideration of dimensional and positional tolerances in the product development process brings additional benefits with respect to the specific product knowledge. Individual tolerances can be specifically optimized in this way in order to save material or production costs and to design complex assemblies with minimizing the needed resources.

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