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USE OF VECTORIAL TOLERANCES IN CAD DURING THE DESIGN PROCESS

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1. Introduction

Mechanical engineers have to consider all product characteristics (e.g. geometry, material, surfaces) and properties (e.g. functional behaviour, assembly) during the design process. An important fact is, that in reality all product characteristics have deviations from their nominal values. Therefore, the engineer also has to investigate the effects of deviations of the characteristics on the properties, especially with regard to kinematic behaviour and assembly [Wittmann et al. 2009]. Currently, the permitted deviations of geometry parameters as well as of the positions and orientations of elements are described by tolerances in drawings and/or by adding semantic annotations to digital product models. Base are usually ISO standards of tolerancing. Since CAD-systems can only evaluate linear, one-dimensional tolerance chains, additional CAx-components (CAT – Computer-Aided Tolerancing systems, e.g. 3DCS, VisVSA) are often necessary for advanced tolerance representation, analysis and synthesis [Ziegler et al. 2013]. Using different systems – CAD and CAT – means that engineers often do not represent the tolerances in the product model and are not aware of the impact of the defined tolerances.

An interesting possibility are vectorial tolerances, which can be represented directly in the parametric CAD-model using surface-type specific attribute containers. These containers are attached to the desired nominal surfaces of the digital product model. Since most engineers think in the ISO-system, a user-controlled translation between ISO and vectorial tolerances is necessary. By direct representation of mathematically evaluable tolerances in the CAD-model, the designer is better motivated to define the necessary tolerances him-/herself and can analyse the impact of deviations along the tolerance chain during the design process.

New from a scientific point of view in the paper is the possibility to analyse partially closed tolerance loops. This is a main focus of the paper. Most CAT-systems only handle open tolerance chains. In contrast, a real product often consists of partially closed tolerance loops. Even if the total tolerance chain can be considered as an open chain, many technical products have partially closed tolerance loops just in the joints.

2. Concept

There exist several types for tolerance representation. The most known and standardised types are the ISO and ASME tolerances (e.g. ISO 1101:2012, ASME Y14.5M-2009), which divide tolerances into dimension, form and position tolerances. These are usually represented using semantic information in 2D-drawings or 3D-models. The tolerances as standardised by ISO or ASME have advantages for conventional manufacturing and metrological inspection (e.g. two-point measurement and use of measuring gauges). However, they are not directly mathematically evaluable and often lead to

misunderstandings concerning the 3D-consequences of tolerances (position and orientation of the tolerance zone) and the combination of tolerances [Britten et al. 1999]. Besides the standardised ISO or ASME tolerances, science has investigated several different tolerance representations, often concentrated on proper mathematical evaluation [Weber et al. 1998], [Krimmel et al. 1999], [Gaunet 2003]. One concept uses vectorial tolerances [Stark 1994]. This concept originally comes from coordinate measuring technology in the 1980s.

2.1 Vectorial tolerance representation

In contrast to current standard tolerance specifications according to ISO or ASME vectorial tolerances are attached to surfaces only. The standardised tolerances describe the limits of possible deviations of points, lines, faces or derived entities. Current investigations in the field of vectorial tolerances focus on five standard surfaces (plane, cylinder, sphere, cone and torus – also see Figure 1), although torus-type surfaces play a secondary role.
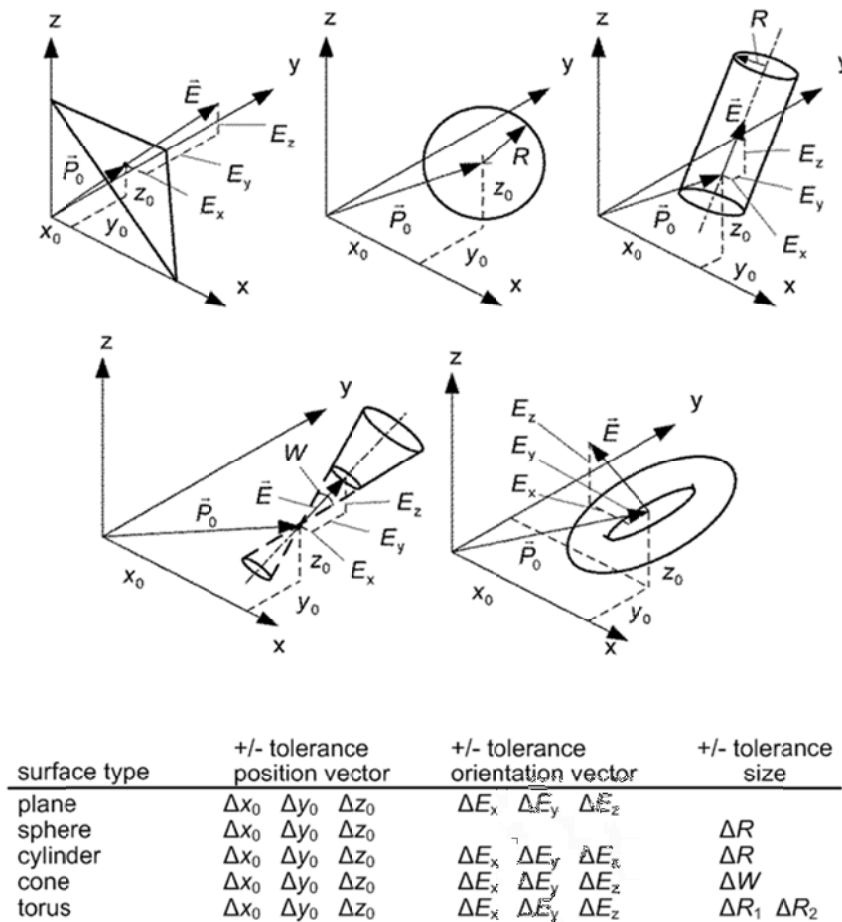

surface type	+/- tolerance position vector			+/- tolerance orientation vector			+/- tolerance size	
plane	Δx_0	Δy_0	Δz_0	ΔE_x	ΔE_y	ΔE_z		
sphere	Δx_0	Δy_0	Δz_0				ΔR	
cylinder	Δx_0	Δy_0	Δz_0	ΔE_x	ΔE_y	ΔE_x	ΔR	
cone	Δx_0	Δy_0	Δz_0	ΔE_x	ΔE_y	ΔE_z	ΔW	
torus	Δx_0	Δy_0	Δz_0	ΔE_x	ΔE_y	ΔE_z	ΔR_1	ΔR_2

Figure 1. Surface types and parameters for vectorial tolerances (according to [Stark 1994] and [Wirtz 1988])

For each surface type up to two tolerance vectors exist (one for the absolute surface position and one for the orientation of the surface; only a sphere does not need an orientation vector). Some surface types need an additional size parameter (see Figure 1). The surfaces in the tolerance representation are boundless (except of sphere and torus, which are selfcontained). The bounded faces of a solid body are results of intersections of several surfaces. Since the vectorial tolerance representation is almost similar to the B-Rep (boundary representation) description in current CAD-systems, it can be integrated directly into current parametric 3D-CAD-systems. The tolerance parameters can be represented by surface-type specific attribute containers, which are attached to the desired nominal surface.
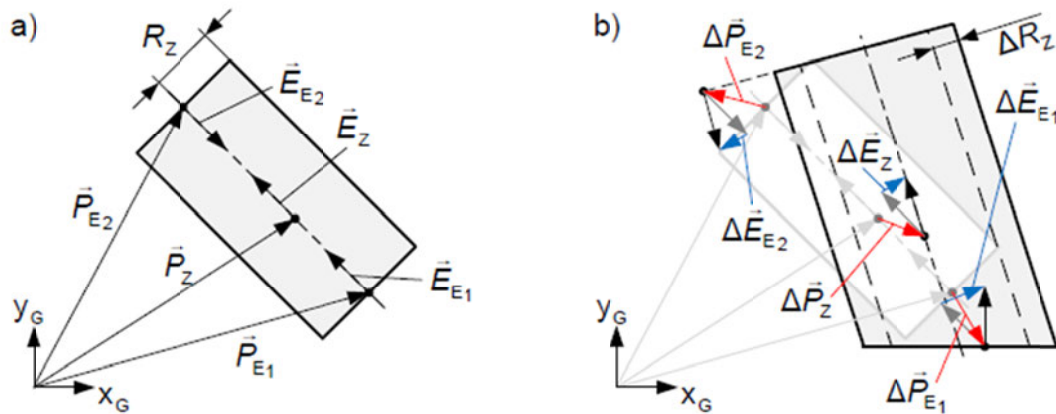
Figure 2. Nominal (a) and deviation effected cylinder (b) based on vectorial tolerance representation [Adam 2012]

By direct representation of mathematically evaluable tolerance parameters inside the CAD-model the designer is better motivated to define the necessary tolerances him-/herself and can analyse the impact of deviations along the tolerance chain (see Table 1). The analysis of the tolerance chain can now be done using standard geometry modelling and evaluating methods in the CAD-system itself. This means that for analysis and visualisation the deviation-affected surfaces can be re-parameterised ("moved") by CAD-API (application programming interface) methods within the limits defined by the tolerances (see Figure 2). In the case of SolidWorks this function is "InsertMoveFace2". This function moves the desired face for specific translation and rotation parameters. In order to ensure a consistent geometry, the CAD-system updates the B-Rep-models automatically as long as the topology remains unchanged. Also the alignment of the parts can be updated automatically as long as the defined mates (e.g. coincident) are still valid (see Figure 3) and the tolerance chain is open (details are explained in the following section). Consequently, it also becomes possible to analyse a tolerance chain across several components. Since all deviation-affected surfaces are described by displacements of ideal-geometric replacement elements, currently form tolerances cannot be represented using the concept of vectorial tolerancing. There exist several approaches in literature to handle form tolerances [Martinsen 1993], which will be addressed in the further work.

Table 1. Comparison of standardised and vectorial tolerance representation

| point of view	criterion	standardised tolerances	vectorial tolerances
mechanical design	familiarity of representation	simple and well known	complex and new in practice
	CAD integration	interpretation and mathematical conversion is necessary	direct
	further use	-	simulation
manufacturing	familiarity of representation	semantic symbols are well known	complex and new in practice
	CAM integration	conversion is necessary	direct
	further use	-	feedback of measuring data
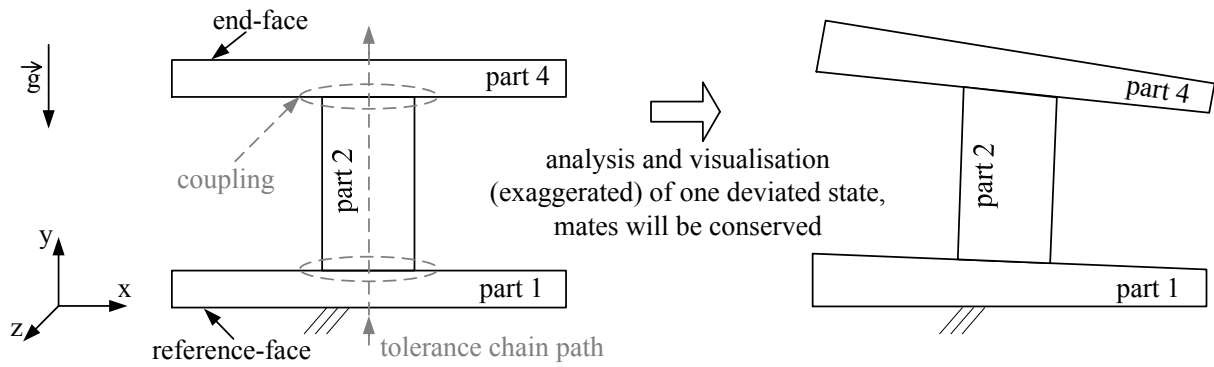
Figure 3. Updated geometry and alignment of parts (here without consideration of forces and friction at the couplings)

2.2 Partially closed tolerance loops

Each tolerance chain is defined by a start- and an end-point. The designer determines these points according to the functional chain he/she wants to analyse. The tolerance chain between these two points is – besides deviations of the individual parts – strongly determined by the couplings of the mated parts in the assembly. In the case, that all couplings are in series, the tolerance chain is considered as "open" (in analogy to the series connection of rigidities in engineering mechanics).

More formal, an open tolerance chain can be defined as:

- Open tolerance chains exist in technical products if each component is coupled with a neighbouring component via one mating surface pair only and, starting from an arbitrary component, this component cannot be reached via an alternative chain of couplings.

An open tolerance chain is exemplified in Figure 3. Start- and end-point of the tolerance chain are arbitrary points of the marked faces. It is clear that the orientations of the individual parts are mainly determined by the change in position of the coupling faces – their influence is much stronger than the influence of the position and orientation deviations of the faces of each individual part. The mates at the couplings can be retained.
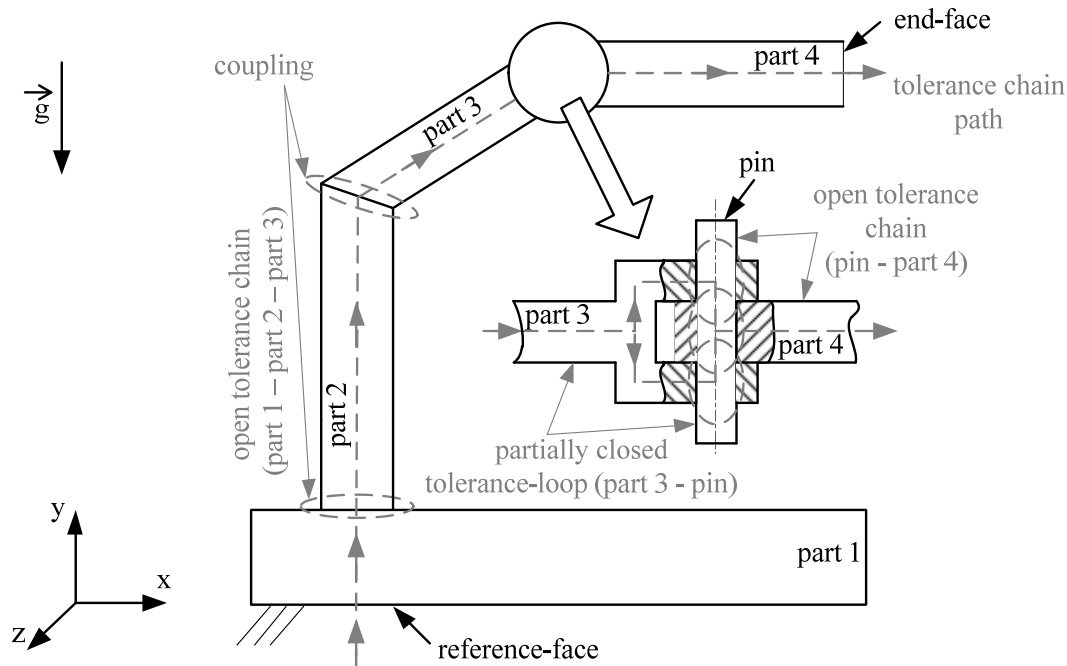

Figure 4. Total tolerance chain with a partially closed tolerance loop within a joint

In reality, most tolerance chains are not fully "open", but result from a concatenation of several open tolerance chains *rwu* some partially closed tolerance loops. A partially closed tolerance loop exists when the tolerance chain path is branched at the couplings, i.e. two or more individual tolerance chains exist in parallel (in analogy to the parallel connection of rigidities in engineering mechanics). More formal, a partially closed tolerance chain can be defined as:

- Partially closed tolerance loops exist in technical products if components have two or more couplings to the same neighbouring component and, starting from an arbitrary component, this component may be reached by a chain of couplings again.

Figure 4 shows a simplified example of a tolerance chain, which is composed of open tolerance chains as well as a partially closed tolerance loop.

For tolerance analysis of open tolerance chains commercially available CAT tools can already be used. An analysis of partially closed tolerance loops is mostly not possible with these tools. Such a partially closed tolerance loop is shown in Figure 5. For ease of illustration in this example, only some coupling faces especially of part 2 and part 3 (coupling to part 4) are provided with position deviations (exaggerated presentation). The difference to the consideration of an open tolerance chain is that the relative orientation of part 4 to part 2 and to part 3 results from the interaction of both position changes of the coupling surfaces of part 2 and part 3.
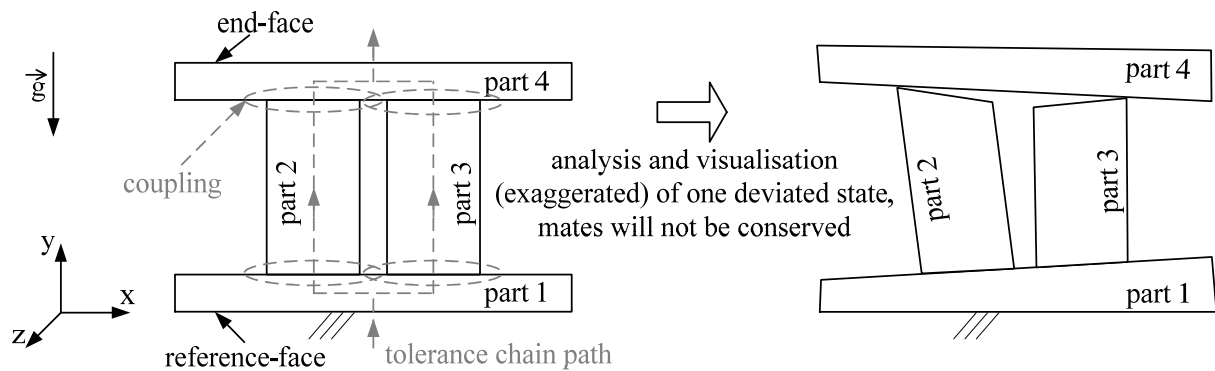

Figure 5. Orientation of the parts (updated geometry) in the case of a partially closed tolerance loop (here without consideration of forces and friction at the couplings)

The system according to Figure 5 is (kinematically) over-determined. Therefore, the mates at the couplings are no longer valid. Consequently, the orientation of part 4 must be re-determined based on the position-changes of both coupling faces of part 2 and part 3.

For further considerations some simplifications are introduced. For the investigation deviation-affected ideal rigid parts, that have faces without form tolerances, are assumed. Moreover, no additional external forces (e.g. caused by screw connections) act on the individual parts except gravity. The basic idea of the concept presented here is based on the statically determinate placement of an ideal rigid body on three points. These three points define a supporting triangle in such a way, that the toleranced part re-aligns itself.

In order to obtain a valid three-point pattern, in a first step the coupling faces are provided with deviations and then all possible patterns are deduced.

A large number of possible support triangles results from all these point-patterns. Now the best suitable support triangle for the re-alignment of the deviation-affected part has to be determined. An essential condition is to guarantee a safe stand of the part, which is fulfilled if the part's projected centre of gravity lies within the respective support triangle (see Figure 6, "intersection"). For assemblies, the combined centre of gravity must be taken into account. For real technical products that are loaded by additional forces and moments, the resulting force and moment must be taken into account.

At this point it is important to mention again, that this concept does not work on the original (geometrically ideal) component mates: Instead, new mates for the alignment of the deviation-affected

parts in a partially closed tolerance loop are constituted by point-to-face, point-to-line or point-to-point contacts.
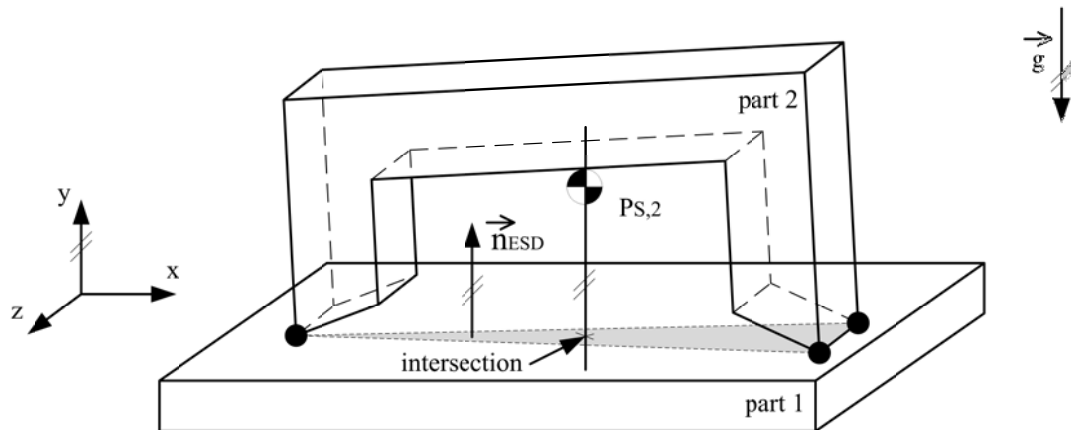

Figure 6. Orientation of a deviation-affected part on a support triangle in a partially closed tolerance loop

2.3 User interface and tolerance transformation

The mechanical engineer is accustomed to tolerance elements or assemblies according to the ISO-system. Tolerancing based on vectorial parameters is unfamiliar and difficult to imagine. Thus, the system presented here was provided with a user interface, which allows the input of tolerance parameters according to the well-known ISO-system. Afterwards, the tolerances are translated into vectorial tolerances automatically.

A translation of tolerances from ISO-system into vectorial tolerances is not unambiguous [Britten et al. 1999]. Therefore, certain assumptions have to be made and/or certain additional values have to be given by the engineer in order to clear ambiguity. This can be done by the user interface, so that the engineer can influence the translation process directly.

Dimension tolerances should be assigned before position tolerances. The dimension tolerances do not affect position tolerances themselves, because in the ISO-system the independence principle is valid (at least it is the default). But during the transformation of position tolerances from ISO-tolerances into vectorial tolerances the dimension tolerances have an effect on the face tolerance vectors. The position tolerance vector represents a combination of standardised dimension and position tolerances. In Figure 7 an abstract example of a cuboid solid is represented, where two opposing faces are toleranced by a parallelism tolerance; it serves to show the superposition of dimension and position tolerances when translating ISO-tolerances into tolerance vectors.
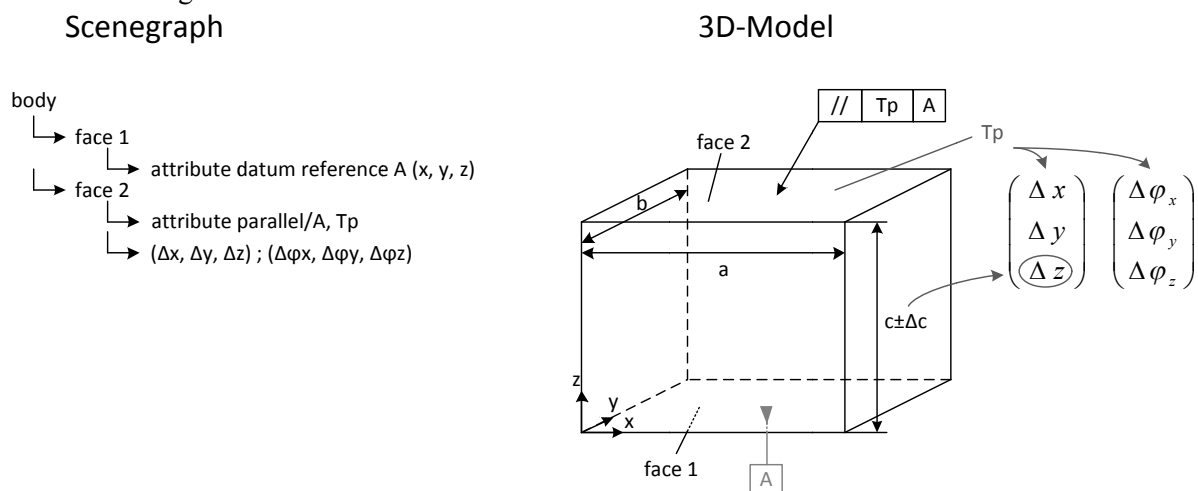
Scenegraph   3D-Model

Figure 7. Cuboid solid with parallelism tolerance (scenegraph and 3D-model)

In this example face 1 is the datum reference and face 2 is the toleranced surface. Based on the parallelism tolerance, the position and orientation of face 2 can deviate between two virtual planes parallel to a plane through surface 1 with a distance to surface 2 of $T_p/2$. This variation of surface 2 leads to a deviation of the surface tolerance vectors $(\Delta x, \Delta y, \Delta z)$ and $(\Delta\varphi_x, \Delta\varphi_y, \Delta\varphi_z)$. By means of these tolerance vectors the deviation of surface 2 can be described (Figure 7). Besides the position tolerances, the dimension tolerances of dimensions a, b and c influence the surface position tolerance vector $(\Delta x, \Delta y, \Delta z)$. In Figure 7 the dimensions a and b are assumed ideal, so there is no influence of these on the tolerance vector. Only the tolerance Δc affects the tolerance vector. It only affects the z-value of the tolerance vector, because dimension c is the length of the cuboid solid in z-direction.

So for the position tolerance vector $(\Delta x, \Delta y, \Delta z)$ there are two effects on the z-value, on one hand from the dimension tolerance Δc and on the other hand from the displacement of face 2 in z-direction (Δz) within the tolerance zone. This leads to the following position tolerance vector:

$$
\begin{pmatrix} \Delta x \\ \Delta y \\ \Delta z \end{pmatrix} = \begin{pmatrix} 0 \\ 0 \\ \Delta c + \Delta z \end{pmatrix}
\tag{1}
$$

The orientation tolerance vector can be calculated with the distance of the tolerance zone and the displacement of face 2 in z-direction (Δz). In order to finish the calculation, a parameter u has to be introduced that describes how the rotation is split between the x- and y-directions (e.g. u = 0,5 → 50 % rotation about the x-axis and 50 % about the y-axis). With these parameters the resulting orientation tolerance vector can be described as follows:

$$
\begin{pmatrix} \Delta\varphi_x \\ \Delta\varphi_y \\ \Delta\varphi_z \end{pmatrix} = \begin{pmatrix} \arctan\left((1-u)*\dfrac{T_p - 2*\Delta z}{b} \right) \\ \arctan\left(u*\dfrac{T_p - 2*\Delta z}{a} \right) \\ 0 \end{pmatrix}
\tag{2}
$$

The information about the tolerances should be attached to the surfaces [Stark 1994], so that the engineer can reconstruct the toleranced surfaces and the corresponding datum references later. This information is stored in attributes, which are attached directly to the surfaces. For the datum references the surface vector is stored and for the toleranced surfaces the tolerance type, the corresponding datum reference and the surface tolerance vectors (position and orientation) are stored (Figure 7).

3. Implementation

The explained concept was implemented as a prototype in the CAD-system SolidWorks using the CAD-API. In a first step the CAD-model is scanned. During this process surface-type specific attribute containers are generated for all surfaces (unless this has been performed earlier). These containers can be visualised to the CAD user, if required. The engineer can define dimension tolerances as usual directly together with the respective dimension definitions. For the definition of position tolerances a special user interface exists as explained in section 2.3. The datum reference(s) also can be defined using this interface.

Through user interaction the tolerance analysis can be started. During this analysis the deviation-affected surfaces are moved (translation and/or rotation) according to the tolerance vectors, using CAD-API functions. The result of the analysis can be seen directly for discrete deviations on the CAD-model.

4. Example application

Tolerances and tolerance chains exist in each geometrical model of a technical product. One focus of the current investigations lies on systems in the area of precision measuring and positioning. For such systems an error analysis has to be performed during the design process in order to minimise the measurement uncertainty. Important influencing factors are the expected form, position and dimension tolerances. Although precision machines as a whole are kinematically well constrained (i.e. no kinematic over-determination) there exist several closed tolerance loops inside the machines. One example is the metrological frame of the Nanopositioning and Nanomeasuring Machine (see Figure 8) [Manske et al. 2012]. This frame connects the measuring tip with the three-dimensional mechanism moving the probe below the (non-moving) tip. Therewith the tolerance chain of the frame influences the measuring accuracy. The tolerance chain of the metrological frame was modelled using the software tool described in this paper. Without the possibility to calculate partially closed tolerance loops the CAD-model has to be modified, so that all closed tolerance loops have to be opened. Considering partially closed tolerance loops and deleting the standard mates in the CAD-model a more realistic tolerance chain and its consequences for the measurement uncertainty of the machine can be calculated.
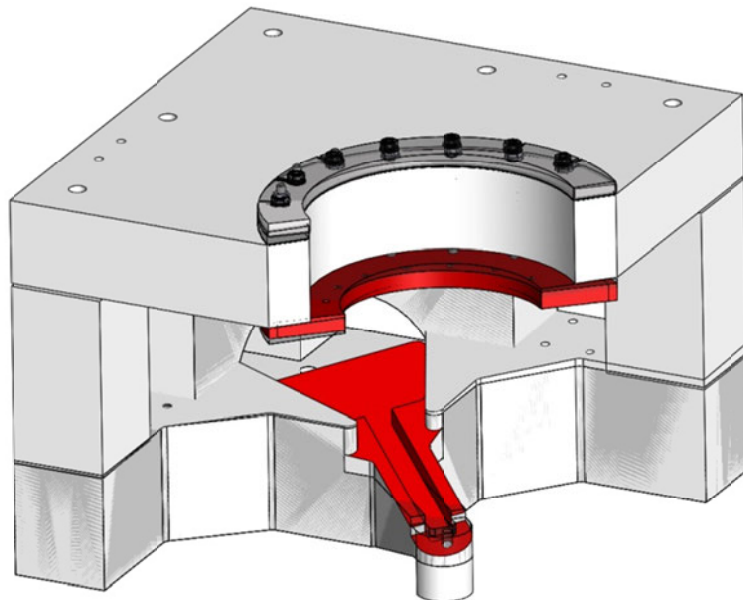

Figure 8. Metrological frame (model simplified) with start- and end-tolerance face (in red: deviated surfaces at the end of the relevant tolerance chain)

5. Summary and outlook

In this paper a method and a tool are presented, which enable tolerance representation and analysis directly in the CAD-system using vectorial tolerances. By direct representation of mathematically evaluable tolerances in the CAD-model the designer is better motivated to define the tolerances him-/herself and can analyse the impact of deviations along the tolerance chain. A major motivation for the integration in the CAD-model is the similarity of the vectorial tolerance representation to the B-Rep (Boundary representation) description in current CAD-systems. In the paper furthermore the user-input is discussed.

From a scientific point of view new in this paper is the possibility to analyse partially closed tolerance loops. When using existing CAT-tools only open tolerance chains can be considered. The developed approach is based on the statically determinate placement of a rigid body on three points, where the projected centre of gravity - even for several parts - lies within the triangle of the three contact points.

Assuming ideal rigid but deviation-affected components the new placement-triangle can be determined.

For the investigations on partially closed tolerance loops as presented in this paper a number of simplifications were made. In the ongoing research these points will be addressed, i.e. the simplifications will be dropped. The research will focus on the impact of additional external forces and moments as well as consideration of form tolerances. Parallel to this, the investigations concentrate on the combination of technological tolerances with thermally- and load-induced deformations.

Acknowledgement

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