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Use of vectorial tolerances for direct representation and analysis in CAD-systems

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Abstract

The paper deals with the representation of vectorial tolerances in the parametric CAD-model using surface-type specific attribute containers, which are attached to the desired nominal surfaces. Since CAD-systems can only evaluate linear, one-dimensional tolerance chains, currently additional CAx-systems are used for tolerance representation, analysis and synthesis. In this paper the representation of vectorial tolerance parameters in the CAD-model is explained. A major motivation for their integration into the CAD-model is the similarity of the vectorial tolerance representation to the B-Rep (Boundary representation) description in current CAD-systems. Furthermore, most engineers think in the standardized tolerance system. So for the user input, a user-controlled translation between the ISO and vectorial tolerances is necessary. The presented approach was realized in a software prototype using the CAD programming interface of SolidWorks.

A main focus of the paper is the handling of partially closed tolerance-loops. Most of the CAT-systems only handle open tolerance chains. In contrast, a real system, even if the overall tolerance chain can be considered as an open chain, consists of partially closed tolerance-loops just in the joints or base frames. The developed approach is based on the statically determined placement of a rigid body on three points, where the resulting line of action of the force - even for several parts – goes through the support triangle. Assuming ideal rigid but deviation-affected components the new placement-triangle can be determined. This is a departure from the classical component mates in 3D-CAD that assume geometrically ideal elements. The concept is demonstrated in the article by examples.

Keywords: vectorial tolerances; CAD-system; closed loops; boundary representation

1. Introduction

During the design process mechanical engineers have to define all product design parameters or characteristics (e.g. geometry, material, surface parameters) as well as have to think ahead and guarantee all properties that are influenced by the design parameters/ characteristics (e.g. functional behavior, assembly). An important fact is that for real products all design parameters/characteristics have deviations from their nominal values. Therefore,

the mechanical engineer also has to investigate the consequences of deviations of the design parameters/characteristics on the product properties, especially with regard to kinematic behavior and assembly [1]. Currently, the permitted deviations of geometry design parameters as well as of the positions and orientations of elements are described by tolerances in 2D-drawings and/or by adding semantic annotations to digital 3D-product-models. Base are usually standards of tolerancing (e.g. by ISO, ASME). Since CAD-systems in general can only evaluate linear, one-dimensional

tolerance chains, additional CAx-components (CAT – Computer-Aided Tolerancing systems, e.g. 3DCS, VisVSA, CETOL) are often necessary for advanced tolerance representation, analysis and synthesis as well as the evaluation [2]. Using different systems for modeling and analysis (CAD- and CAT-tools) means that mechanical engineers often do not represent the tolerances in the product model (not at all or not properly) and are not aware of the consequences of the defined tolerances.

The goal of the presented work is:

- immediate assignment of analyzable tolerances to the product model by the mechanical engineer
- tolerance analysis and evaluation by the mechanical engineer in person in the familiar working area

An interesting possibility for the representation and analysis 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. By direct representation of mathematically evaluable tolerances in the CAD-model the analysis of the impact of deviations along the tolerance chain during the design process is much easier for the mechanical engineer. The considered applications in this research focus on products consisting mostly of components with ruled geometries.

A main focus of the paper is the handling of partially closed tolerance loops. Most CAT-systems only handle open tolerance chains. In contrast, a real product often consists of partially closed tolerance loops. Closed tolerance loops exist either in kinematically closed chains (e.g. crank-rocker mechanisms) or in over-determined systems (e.g. joints, base frames) where a high rigidity is necessary. Even if the overall tolerance chain can be considered as an open chain, many technical products have partially closed tolerance loops.

2. Concept of tolerance representation

There exist several types of tolerance representations. The most known and standardized 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 standardized 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 misinterpretations with regard to 3D-consequences of tolerances (position and orientation of the tolerance zone) and combinations of tolerances [3]. Besides the standardized tolerances, science has investigated several different tolerance representations, often concentrated on proper mathematical evaluation [4], [5], [6], [7]. One concept uses vectorial tolerances [8]. This concept originally comes from coordinate measuring technology in the 1980s. First investigations using vectorial tolerances in 3D-CAD were done during the 90s. One difficulty of the investigation was that the CAD-systems were not ready for an implementation yet. In the meantime, the CAD technology is quite sophisticated, so a new and extended attempt of realization looks promising.

In contrast to standard tolerance specifications, vectorial tolerance representations address only the surfaces of components. Current investigations on tolerance representation and analysis focus on five standard surfaces (plane, cylinder, sphere, cone and torus – also see Fig. 1); although due to their wide-spread application torus-type surfaces play a secondary role.

A surface is described in the three-dimensional Euclidean space by a mathematical description for the geometry, a nominal position and orientation as well as the allowed deviations. So for each surface type up to two tolerance vectors exist: One for the position and one for the orientation of the surface; only a sphere does not need an orientation tolerance vector. Some surface types need an additional size tolerance parameter (see Fig. 1). The surfaces in the tolerance representation are boundless (except of sphere and torus, which are closed surfaces). The bounded faces of a solid body – its topology – 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 at the part level (see Fig. 2).
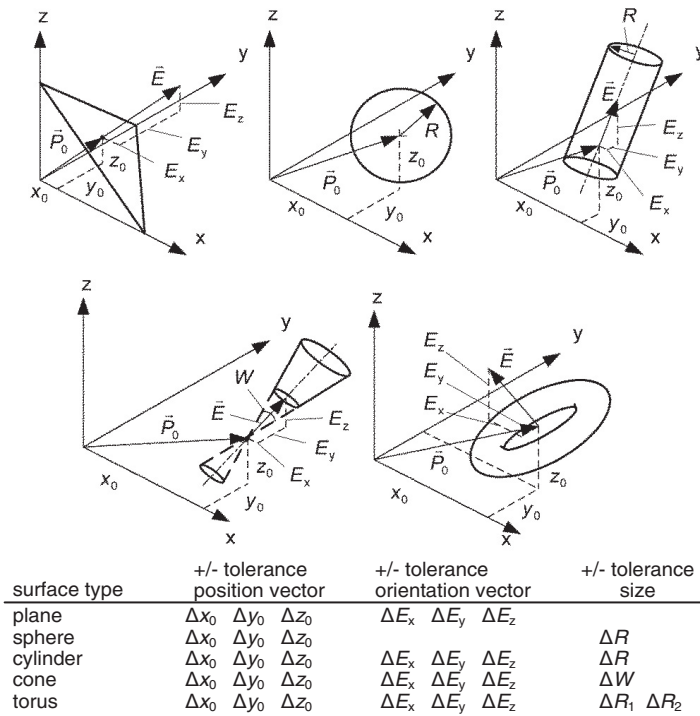
Fig. 1. Surface types and parameters for vectorial tolerances (according to [8] and [13])

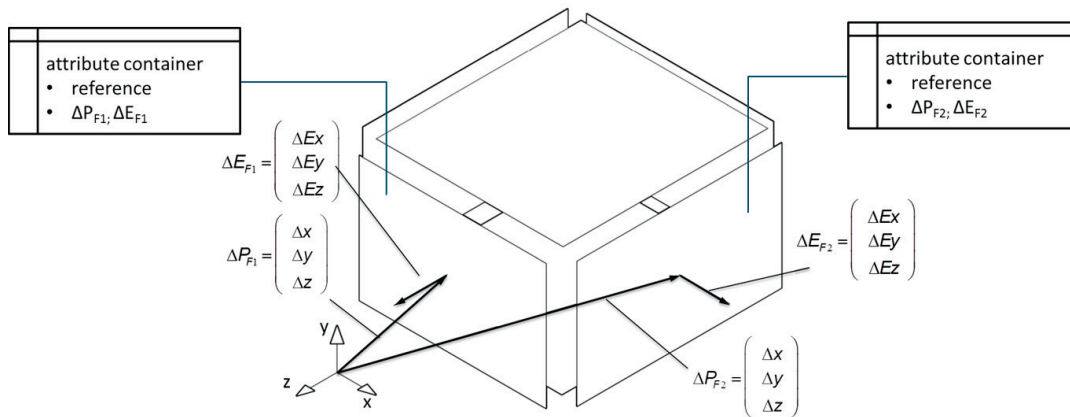| surface type	+/- tolerance position vector			+/- tolerance orientation vector			+/- tolerance size	
plane	Δx_0	Δy_0	Δz_0	ΔE_x	ΔE_y	ΔE_z		
sphere	Δx_0	Δy_0	Δz_0				ΔR	
cylinder	Δx_0	Δy_0	Δz_0	ΔE_x	ΔE_y	ΔE_z	ΔR	
cone	Δx_0	Δy_0	Δz_0	ΔE_x	ΔE_y	ΔE_z	ΔW	
torus	Δx_0	Δy_0	Δz_0	ΔE_x	ΔE_y	ΔE_z	ΔR_1	ΔR_2

Fig. 2. Faces with tolerance vectors and attached attribute containers

By direct representation of mathematically evaluable tolerance parameters inside the CAD-model the mechanical engineer is better motivated to define the tolerances him-/herself and can analyze the impact of deviations along the tolerance chain (see Table 1). The analysis of the tolerance chain can now be done using standard geometry modeling and evaluating methods in the CAD-system itself. This means that for analysis and visualization the deviation-affected surfaces can be re-parameterized ("moved" – displacement, rotation and scaling) by CAD-API (application programming interface) methods within the limits defined by the tolerances (see Fig. 3). In the case of SolidWorks this necessary API-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 of the parts automatically as long as the topology

Table 1. Comparison of standardized and vectorial tolerance representation

| point of view	criterion	standardized tolerances	vectorial tolerances
design and development	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 preparation and 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
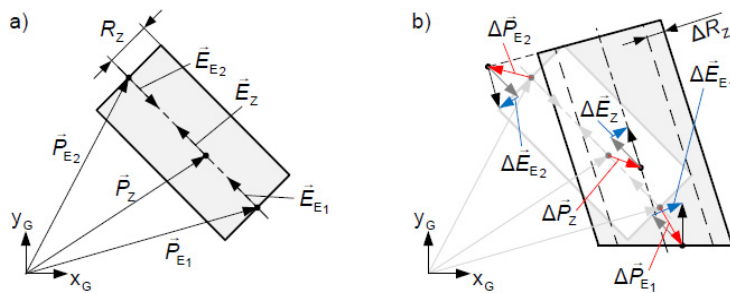
Fig. 3. Nominal (a) and deviation effected cylinder (b) based on vectorial tolerance representation [12]

remains unchanged. Usually, the deviations are very small. Therefore, for most parts the topology is maintained. Also the alignment of the parts can be updated automatically as long as the defined mates (e.g. "coincident") are still valid (see Fig. 4) and the tolerance chain is open (details are explained in the following section). Consequently, it also becomes possible to analyze 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 directly (only the limits by the ideal-geometric replacement elements) using the concept of vectorial tolerancing. There exist several approaches in literature to handle form tolerances [9] , which will be addressed in the further work.

3. Partially closed tolerance loops

3.1. Motivation

For tolerance analysis of open tolerance chains commercially available CAT tools can already be used. In contrast, a real product often consists of partially closed tolerance loops. Even if the overall tolerance chain can be considered as an open chain, many technical products have partially closed tolerance loops just in the joints or base frames. For such an analysis of partially closed tolerance loops these CAT tools cannot be used. So in the following part a concept for handling partially closed tolerance loops is presented.

3.2. Concept of analysis

Each tolerance chain is defined by a start- and an end-point. The mechanical engineer determines these points according to the functional chain he/she wants to analyze. The tolerance chain between these two points is
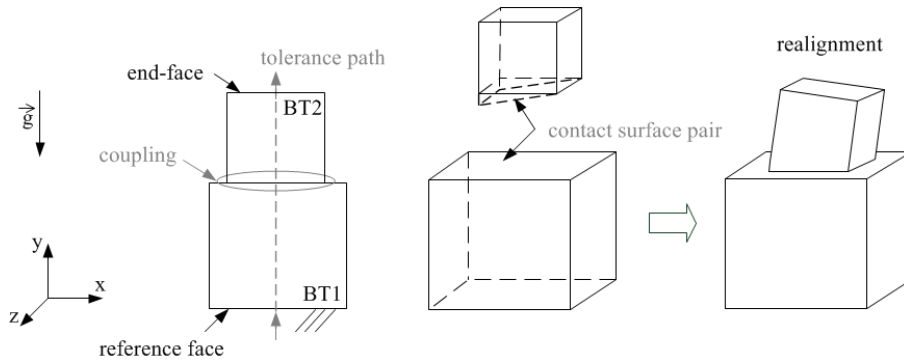
Fig. 4. Updated geometry and alignment of the parts (here without consideration of force and friction at the couplings)

– 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 neighboring 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 Fig. 4. 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.

In reality, most tolerance chains are not fully "open", but result from a concatenation of several open tolerance chains *plus* 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 (analogy to the parallel connection of rigidities in engineering mechanics).

More formal, a partially closed tolerance loop can be defined as:
• Partially closed tolerance loops exist in technical products if components have two or more couplings to the same neighboring component and, starting from an arbitrary component, this component may be reached by the chain of couplings again.

Such a partially closed tolerance loop is shown in Fig. 5. For ease of illustration, in this example only the 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 orientations of part 4 to part 2 and to part 3 result from the interaction of both position changes of the coupling surfaces of part 2 and part 3.

The system according to Fig. 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 the coupling faces from part 2 and part 3.
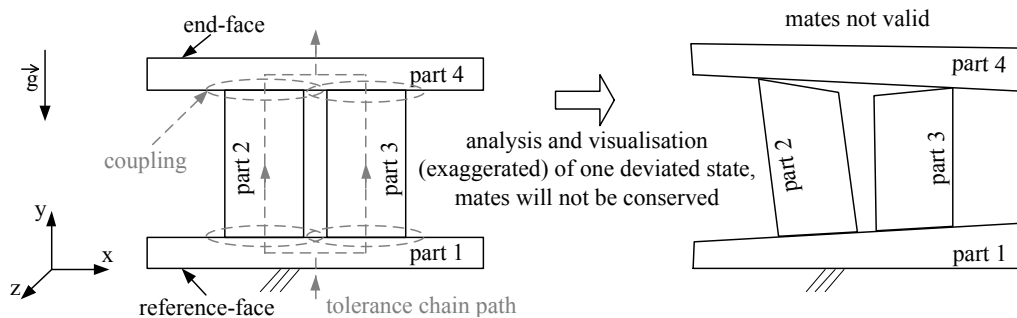

Fig. 5. Orientation of the parts (updated geometry) by existing of a partially closed tolerance loop (here without consideration forces and friction at the couplings)

In Fig. 6 a simplified model of a metrological frame of a Nanopositioning and Nanomeasuring Machine with closd tolerance loops can be seen.
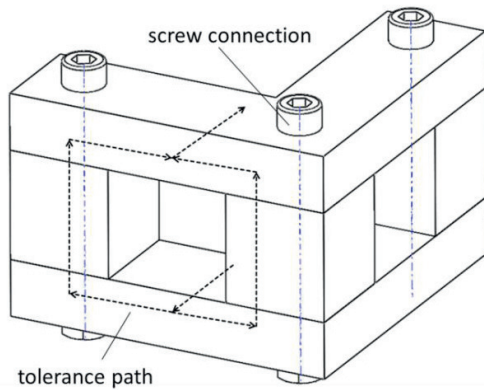

Fig. 6. Simplified metrological frame of a Nanopositioning and Nanomeasuring Machine with partially closed tolerance loops

The tolerance path shows a partially closed tolerance loop across the parts. In this metrological frame the contact of the different parts is held through screw connections, which put an external force to the assembly.

For further considerations some simplifications are introduced. For the investigation deviation-affected ideal rigid components, that have faces without form tolerances, are assumed. Moreover, no additional external forces (e.g. caused by screw connections) act on the individual components except gravity.

The basic idea of the concept presented here is based on the statically determinate placement of ideal rigid bodies on three points. These three points define a supporting triangle in such a way, that the toleranced components or assemblies (connection of components) re-align themselves.

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.
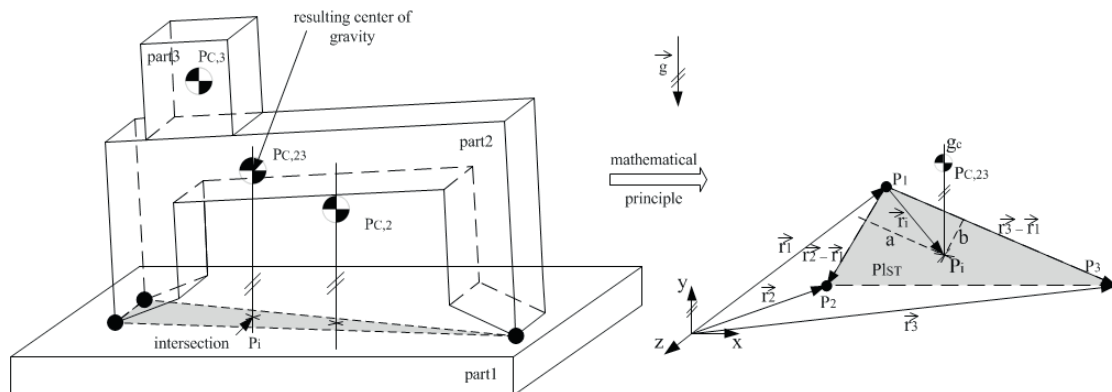
A large number of possible support triangles result from all these point-patterns. Now the best suitable support triangle for the re-alignment of the deviation-affected components has to be determined. An essential condition is to guarantee a safe stand of the components, which is fulfilled if the component's projected center of gravity lies within the respective support triangle. In assemblies, the combined center of gravity must be taken into account (see Fig. 7, "intersection"). For mathematical verification of this essential condition, a straight line, as a first step, is formed, which runs through the center of gravity P_C (either only for one component or an assembly) in the direction of gravity. Furthermore, a plane Pl_{ST} is derived by the three points of the support triangle. Then, the intersection point P_i between the straight line and the plane is determined. The position of this intersection point can be described by the vector $\vec{r_i}$, which is characterized by the following equation (1):

$$\vec{r_i} = a \cdot (\vec{r_3} - \vec{r_1}) + b \cdot (\vec{r_2} - \vec{r_1}) \qquad (1)$$

The sum of the parameters a and b determines, whether the intersection is located inside the supporting triangle or outside. For this, the following mathematical correlation (2) is used:

$$a + b < 1 \qquad [a, b \in R] \qquad (2)$$

Only if the sum is less than one, the intersection point is located inside the supporting triangle. For this case, all possible support triangles are analyzed. For further aptitude tests (e.g. volumetric intersections after re-alignment), only support triangles are considered where the intersection point lies within the support triangle and thus a safe stand for the deviation-affected components or assemblies is guaranteed.

At this point it is important to mention again, that this concept does not work on the original (geometrically

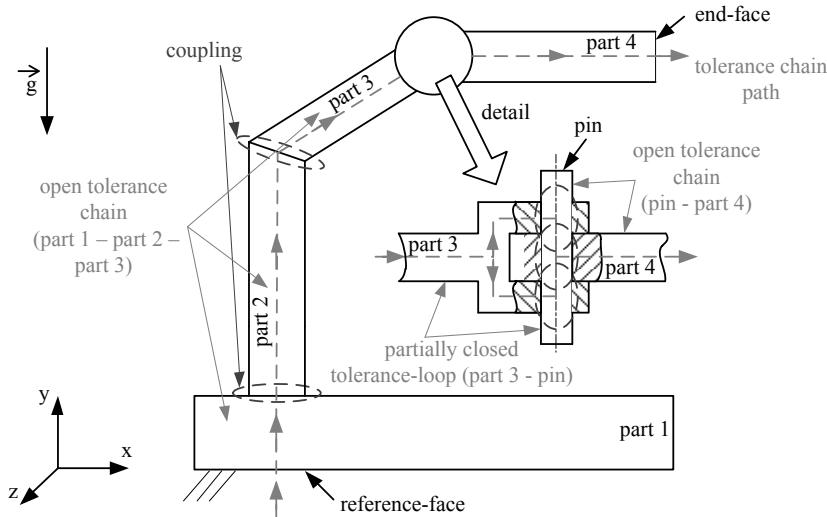Fig. 7. Orientation of a deviation-affected assembly on a support triangle in a partially closed tolerance loop

Fig. 8. Total tolerance chain with partially closed tolerance loop within a joint

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.

3.3. Example

Fig. 8 shows a simplified example of a tolerance chain which is composed of open tolerance chains as well as a partially closed tolerance loop. The open tolerance chain includes parts 1, 2, 3 and 4. And the partially closed tolerance loop is located in the joint between parts 3 and 4, which is shown in detail in the figure.

4. User interface and tolerance transformation

The mechanical engineer is accustomed to tolerance parameters 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. These translated tolerances can be visualized in the CAD-Model und thus the user can see how the different components are influenced by these tolerances and how they influence the whole assembly (scheme Fig. 9).

A translation of tolerances from ISO-system into vectorial tolerances is not unambiguous [3]. Therefore, certain assumptions have to be made and/or certain additional values have to be given by the engineer for clarifica-
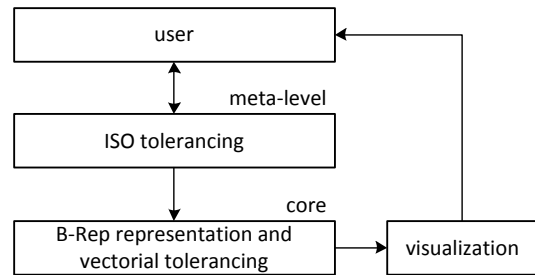

Fig. 9. Principle of the user interface and tolerance transformation

tion. This can be done by a properly designed 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 standardized dimension and position tolerances. In Fig. 10 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.

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
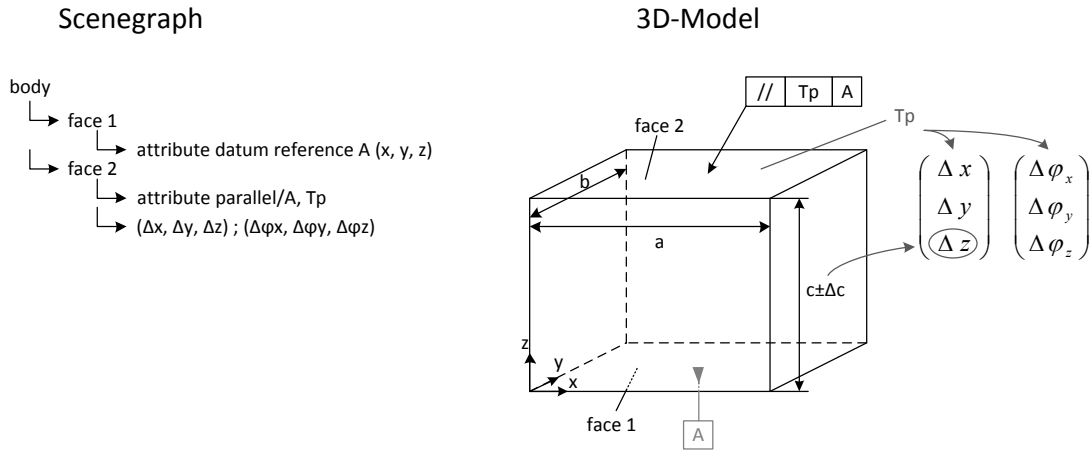
Scenegraph   3D-Model

Fig. 10. Cuboid solid with parallelism tolerance (scenegraph and 3D-model)

deviate between two virtual planes parallel to a plane through surface 1 with a distance to surface 2 of Tp/2.

The deviation can be caused by a displacement of surface 1 in z-direction and by a rotation of the surface about the x- and y-direction. This leads to a position tolerance vector $(\Delta x, \Delta y, \Delta z)$ and an orientation tolerance vector $(\Delta \varphi_x, \Delta \varphi_y, \Delta \varphi_z)$.

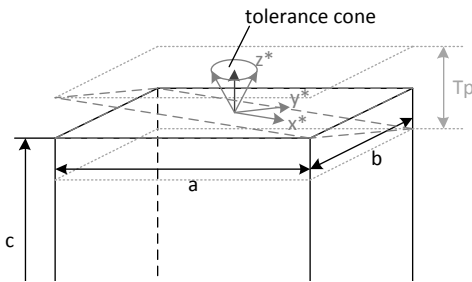By means of these tolerance vectors the deviation of surface 2 can be described (see Fig. 11).

Fig. 11. Cuboid solid with parallelism tolerance and tolerance cone

Besides the position tolerances, the dimensional tolerances of dimensions a, b and c influence the surface position tolerance vector $(\Delta x, \Delta y, \Delta z)$. In Fig. 10 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:

$$\begin{pmatrix} \Delta x \\ \Delta y \\ \Delta z \end{pmatrix} = \begin{pmatrix} 0 \\ 0 \\ \Delta c + \Delta z \end{pmatrix} \qquad (3)$$

The orientation tolerance vector is affected by the rotation of face 2 about the x- and y-axis. When the face is rotated, the orientation vector changes. The face can be rotated in different splittings between the two axes, e.g. 50 % about the x- und 50 % about the y-axis or 70 % about the x- and 30 % about the y-axis. A parameter u is introduced to describe the splitting between the rotation-axes (e.g. u = 0,5 → 50 % rotation about x-axis and 50 % about y-axis). Through the different possibilities of the orientation vector a tolerance cone can be created which limits the tolerance orientation vectors (see Fig. 11) [8].

The boundaries of the tolerance cone also show the maximum allowed rotational deviation of the orientation vector. The deviation limitation by this tolerance cone for the orientation vector is greater than the limitation by the tolerance pyramid which is defined by the ISO-tolerances. So the vectorial tolerances allow a smaller tolerance cone angle than tolerancing by the ISO-system [3]. The orientation tolerance vector can be calculated with the distance of the tolerance zone and the displacement of face 2 in z-direction (Δz). With these parameters the resulting orientation tolerance vector can be described as follows:

$$\begin{pmatrix} \Delta\varphi_x \\ \Delta\varphi_y \\ \Delta\varphi_z \end{pmatrix} = \begin{pmatrix} \arctan\left((1-u) * \dfrac{T_p - 2 * \Delta z}{b} \right) \\ \arctan\left(u * \dfrac{T_p - 2 * \Delta z}{a} \right) \\ 0 \end{pmatrix} \quad (4)$$

The information about the (vectorial) tolerances should be attached to the surfaces [8] 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 (Fig. 10).

5. Implementation

The explained concept was implemented as a prototype in the CAD-system SolidWorks using the system's API. In a first step the CAD-model is scanned component by component. During this process surface-type specific attribute containers are generated for all surfaces in the component (unless this has been performed earlier). These containers can be visualized to the CAD user, if required.

One of the major tasks is to identify the contact surfaces between the components during the assembly process and to determine the (relevant) contact parameters. The contact surfaces in the CAD-model result from the (nominal) geometry of components and their alignment (position, orientation). The contact surfaces are not described explicitly in the CAD-model. However, they can be detected by a face-to-face comparison. This enables a good-enough identification of potential contact surface pairs. The proposed face-to-face comparison is complex and time-consuming, especially because of the different surface types involved, their various combinations and various modeling strategies that may have been used to create the CAD-model. Using the constraints between the components in the CAD-model, a bounding-box comparison and a scaling procedure with a subsequent intersection test [10] in a pre-process helps to reduce the surface pairs which have to be tested for contact. Currently form and force connections can be interpreted for elementary contact surface pairs (see Table 2).

The developed prototype was equipped with a graphical user interface (GUI) that allows the definition of position tolerances as explained in section 2.3. The respective datum reference(s) also can be defined using this interface. The engineer can define dimension tole-

Table 2. Interpretable contact surface pairs

	cone	cylinder	plane	sphere	torus
cone	x				
cylinder		x	x	x	x
plane		x	x	x	
sphere		x	x	x	
torus		x			

rances as usual directly together with the respective dimension definitions via the standard CAD-GUI.

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.

6. 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 machinery for measuring and positioning purposes. For such systems an error analysis has to be performed during the design process in order to minimize 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 developed by TU Ilmenau (see Fig. 12) [11]. 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 modeled 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.
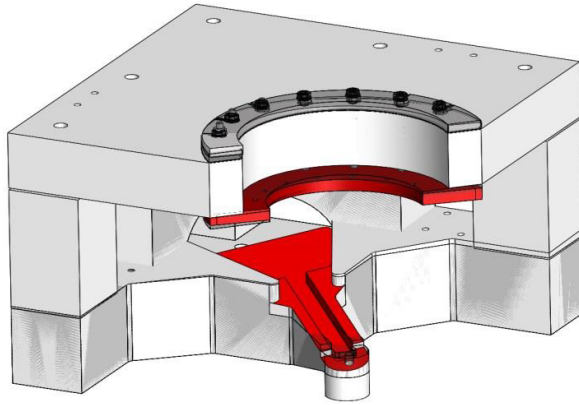
Fig. 12. Metrological frame (model simplified) with start- and end-tolerance face (in dark: deviated surfaces at the end of the relevant tolerance chain)

7. 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 mechanical engineer is better motivated to define the tolerances him-/herself, and he/she can directly analyze 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. Furthermore, in the paper the user-input is discussed – preferably done in standard ISO tolerances and then translated into vectorial parameters as required for more precise analyses.

New from a scientific point of view is the possibility to analyze partially closed tolerance loops. When using existing CAT-tools only open tolerance chains can be considered (or closed loops assumed open). The developed approach is based on the statically determinate placement of a rigid body on three points, where the resulting line of action of the force - even for several parts – goes through the support triangle. Assuming ideal rigid but deviation-affected components the new placement-triangle for the alignment can be determined.

For the investigations on partially closed tolerance loops as presented in this paper a number of simplifications were made in the first place. In the ongoing research these points will be addressed, i.e. the simplifications will be successively 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.

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