INTERNATIONAL CONFERENCE ON ENGINEERING DESIGN
ICED 05 MELBOURNE, AUGUST 15-18, 2005

QUALITY MANAGEMENT BY CONCURRENT COMPUTER AIDED TOLERANCE ANALYSIS AND FUNCTIONAL PROTOTYPING

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Keywords: Computer Aided Tolerancing, Multi Body Systems simulation, Quality Management

Abstract

Six Sigma philosophy is the dominating paradigm in mass production to minimise loss units. The geometric variation of a product is a source of malfunction of the product caused by the design, manufacturing and assembly processes. Simulation tools such as Computer Aided Tolerance (CAT) analysis and Multi-Body Systems (MBS) analysis provide designers with models that can be utilized for this purpose. The objective of this paper is to study the similarities of modelling components in CAT- and MBS-analysis. Based on the study, a general process model for concurrent utilisation of the tools in the design process is presented. The dimensional variation analysed by CAT predicts the tolerance level of Six Sigma limits. The variation of Six Sigma limits is utilised in the MBS analysis to predict the functional variation of a product. Finally, the behaviour of mass-produced products is predicted as a function of their tolerance limits.

Nomenclature

c_i = Constraint restricting i degrees of freedom.
$\Delta\overline{q}$ = Vector describing the clearance between two points belonging to two different mating parts.
Δq_k = kth clearance.
n_j = Relative variance in Gaussian distribution in jth d.o.f..
n_{tot} = Degree of freedom in a system.
n_k = Degree of restrictions of kinematic constraints.
n_{cl} = Degree of restrictions of flexible constraints.
n_{cons}= Total degree of constraints.
P = Total number of parts in assembly ground included.
q = Coordinate translation and rotation vector for transformation in part i HTM-matrix.
T = 4x4 Homogenous Transformation Matrix describing coordinate transformation.
r = 4x1 Vector describing the position of contact in given coordinate system.
x_j = Average clearance of kth d.o.f. obtained from CAT tolerance simulation.
$\overline{x}$ = Vector describing the geometric dimension and tolerance variation in a given part of assembly.
$\overline{y}$ = Vector describing the position and orientation of all parts respect to ground part in an assembly.
α = 1st Euler rotation angle around z-axis.
β = 2nd Euler rotation angle around x-axis.
γ = 3rd Euler rotation angle around new z-axis.
σ_j = Standard deviation of clearance kth d.o.f. obtained from CAT.
τ_k = Relative deviation of clearance at kth d.o.f.
Indices:
asi = assembly variation of part i.
cl = Degree of freedom restriction of flexible tolerance joints.
i = Part i in mating kinematic chain.
k = k’th clearance fit in joint system.
$kini$ = Kinematic displacement of joint i.
$toli$ = Part i internal tolerance deformation.
tot = Total coord. transf. from the kinematic, part and assemb. chain from "leave" part to ground part.
br = "Leave" parts coordinate system.

1 Introduction

1.1 Background

The management of the mass production of electro-mechanical products requires the management of manufacturing tolerances at part and assembly levels. In recent years, a new manufacturing paradigm, Six Sigma quality management, has changed manufacturing strategy and minimized the quota of scrap units. In order to minimise time consumption and determine costs, the management of tolerance analysis has been transferred to the design phase to precede manufacturing. During this phase, the manufacturing process is not defined and the costs of process changes are less. The development of Computer Aided Engineering (CAE) has enabled two tools to simulate the product functions in early design: Multi-Body Systems (MBS) simulation and Computer Aided Tolerancing (CAT).

Computer Aided Tolerancing is a tool that simulates the effects of dimensional variation of the manufacturing of parts and their assembly as completed products. CAT models often consider the parts ideally rigid kinematically, with no elasticity or friction to compute the mutual position change between interacting parts and their clearances. The rigid and kinematic model is a strong idealisation; physically more realistic results are obtained by Multi-Body Systems simulation with elastic joints. The designer should ensure that the interacting parts have contacts only over bearing elements.

Tolerance analysis consists of tolerance specification, variation modelling, and sensitivity analysis. In tolerance specification, the allowed variation in shapes and configurations are defined for the parts of the system. The specification can be parametric, geometric [1,5] or vector based [8]. Variation modelling produces mathematical models that map the tolerance specifications to assembly and function variations. Finally, with sensitivity analysis, the critical properties are studied with worst-case and statistical analyses.

1.2 Clearance effects

While, with CAT, the geometrical clearance can be analysed, MBS modelling is required to physically simulate the clearance function of a product. Slide bearings are often subjected to dynamic loading conditions, and therefore must be designed against fatigue failure. Accurate prediction of the dynamic behaviour of the bearing contact is therefore required. The clearance falls into three functional categories – negative clearance or compression, zero clearance and positive clearance – for every sliding bearing depending on the free distance between the shaft and the housing.

- In negative clearance, the shaft and the housing are compressed together so tightly that no free distance is available. The negative distance describes how much the parts would be compressed inside each other if the bodies were rigid rather than elastic. The bearing contact force is in the normal direction of the bearing functional plane, while its force magnitude is a function of the negative compression length.

- In zero-clearance, the parts are compressed against each other so that no free length exists between the shaft and housing. The compression force is very low or zero.

- In positive clearance, a loss of contact has occurred and there is a free gap between shaft and housing. When this happens, the bearing contact force is reduced to zero enabling the pin to move freely in the clearance gap. If this occurs, undesirable impact may result when contact is remade. To avoid such loss, a model, able to predict bearing forces accurately is required.

The objective of this paper is to present a method for the mutual utilisation of CAT and MBS simulation early in the product design phase to ensure the electro-mechanical products readiness for mass production. The method utilises the tolerance variation data from critical functional dimensions, i.e., the joint clearances. The variation of CAT clearances are utilised in MBS analysis to predict the behaviour of a slider mechanism as a function of clearance variation. This is the original contribution of the paper.

1.3 Review of literature

Tolerance design involves tolerance specification, tolerance analysis and tolerance synthesis. First, the tolerance system is specified; in analysis, the product has given dimensions and tolerances, while, in synthesis, these are changed to obtain better functionality and less scrap. Statistical tolerance analysis has been reviewed by Turner and Nigam [4] and tolerance synthesis methods by Voelcker [5], Juster [6], and Chase and Parkinson [7]. According to [11], kinematic tolerance analysis falls into three general categories: static (small displacement) analysis, kinematic (large displacement) analysis and kinematic analysis with contact changes. The one-dimensional approach is simplest, but very applicable to many engineering problems and based on tolerance stack analysis. Simple min-max analysis gives the worst case (WC) analysis by adding the minimal and maximal dimensions. Another deterministic analysis type is DOE, Design of Experiments, where parameter sensitivity is studied by systematic variation. Many other deterministic methods exist, like integer programming, non-linear programming and heuristics, reviewed by Kusiak and Feng [2]. In statistical variation, the probability of worst limits decreases near zero, and the tolerance limits increases the manufacturing costs. Assumptions for the statistical distribution must therefore be applied. The RSS (Root Sum Square) approach computes the distribution usually assuming normal distribution. The synthesis of tolerance limits is then selected to follow standard deviation with 6σ ($\pm 3\sigma$) standard deviation limits leading to 2.7 scrapped units per thousand. The Monte Carlo method is a simple and popular method of statistical analysis. It is suitable for stack, but more applicable to planar and spatial tolerance analysis. Random values for variations are generated according to statistical distribution; the method is therefore also applicable to distributions other than normal. It can be easily applied to linear and non-linear response functions, since the function values are computed by simulation. The major drawback of the method is that, in contrast to deterministic methods, intensive simulation is required to get accurate estimates. On the other hand, if the number of points is insufficient, the Monte Carlo analysis becomes inaccurate. The number of evaluations is considerably reduced in Taguchi method. This reduces considerably the computational effort, apparently without compromising the reliability of the results.

In this paper, statistical tolerance analysis is used. Commercial CAT and MBS software are utilised. 3D solid geometry models are imported from CAD geometry. The modelling and solving of kinematic (small displacement) tolerance equations are performed with VisVSA and the dynamic (large displacement) motion analysis is performed with ADAMS.

2 Method for integration of MBS and CAT-model

2.1 Computer Aided Tolerance modelling

Computer Aided Tolerance analysis enables the computation of the mutual part clearance vector. It depends on the geometry (dimension) variation vector and assembly variation vector, which describes the position and orientation of all parts relative to the origin of the ground part. CAT analysis then solves the clearances in the mechanism system by computing the clearance vector several times, with appropriate variations on part and assembly parameters, the Monte Carlo method, for instance.

$$\Delta\overline{q} = \Delta\overline{q}(\overline{x}, \overline{y}) \tag{1}$$

It is possible by this method to obtain the statistical variation of the clearance vector. The clearance vector describes the clearance between two mating parts in some certain designer selected points. To be re-utilised in multi-body analysis, the clearance points must be identical in CAT and MBS analysis. In MBS analysis, the joint clearances are described with position feedback force functions modelling the contact phenomena at joints.
(Figure 1. and 4.).

In this paper, the numerical analysis of CAD geometry clearance and multi-body dynamics is performed with commercial software. The mechanical assembly vector can also be computed using the coordinate transformation method. For each part in the assembly, its position relative to origin is assumed to depend on its location in the chain of mated parts. The position chain is dependent on the variations in the mating chain. Each mated part varies the chain by three coordinate system transformations [9]:

1. The large displacement of part i due to kinematic displacement of joint $T(q_{kini})$ part i's small internal deformation due to tolerance variation $T(q_{toli})$.
2. Small assembly variation due to changes in part i's assembly tolerance $T(q_{asi})$.

The coordinate system transforms from kinematic to tolerance and assembly variation are shown in figure 1.

The total chain of displacement for part n in the chain is thus

$$T_{tot} = \prod_{i=1}^{n} T(q_{kini})T(q_{toli})T(q_{asi}) \tag{2}$$

The total transformation with part geometry variation, kinematic movement, and assembly variation of the part mating chain is thus:

$$r_{tot} = \underset{4x4}{T}{}_{tot}\, r_{br} \tag{3}$$

where the dimension vectors depending on the geometry are expressed by the GD&T method (Geometric Dimensioning and Tolerancing). The variation chain method described by eq. (2) is independent of the way tolerances are described; it can therefore be applied also to the Vectorial Dimensioning and Tolerance Concept originally described by Humienny [8].

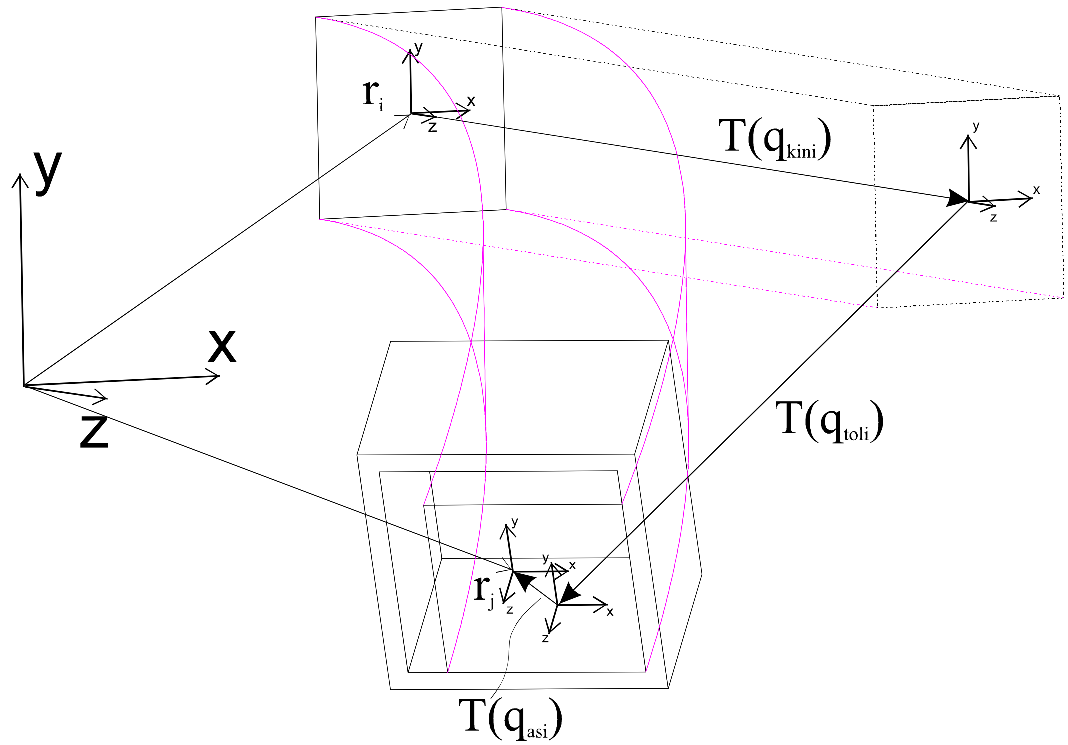

Figure 1.The coordinate transforms from part i to j with prismatic joint.

The coordinate transformations are expressed by 4x4 homogenous transformation matrices (HTM-matrix), where the co-ordinate system is first rotated with a 3x3 submatrix by Euler angels or some other rotation specification method, and then translated with a 4x1 translation vector.

$$\underset{4x4}{T}(\alpha,\beta,\gamma,r_x,r_y,r_z) = \begin{bmatrix} \underset{3x3}{R(\alpha,\beta,\gamma)} & \begin{matrix} r_x \\ r_y \\ r_z \end{matrix} \\ \underset{1x3}{0} & 1 \end{bmatrix} \tag{4}$$

2.2 Multi Body Systems modelling

The concept of integration of computer aided tolerance (CAT) analysis to multi body systems (MBS) analysis is based on the approach, that both models have identical setup of parameters $\Delta\bar{q},\bar{x}$. and $\bar{y}$. The tolerance vector $\bar{x}$ is varied by statistical dimensional variation method for each simulation case by Monte Carlo simulation. It is based on Gauss-distribution, but even other methods can be utilised.

The dimension of the assembly constraint vector is defined in MBS-analysis. The total degrees of freedom of the mechanism assembly is

$$n_{cons} = n_k + n_{cl} \tag{5}$$

It is divided into two kinds of constraints, where the total number of constraints is n_{cons}: ideally kinematic constraints nk and flexible tolerance modelling constraints n_{cl}.

$$n_{cons} = n_k + n_{cl} \tag{6}$$

where n_k is the number of constrained degrees of freedom by kinematic joints given by the equation

$$n_k = \sum_{i=1}^{5} i \cdot c_i \tag{7}$$

where i is the number of joints constraining c_i degrees of freedom. Thus, the large unconstrained displacement degree of freedom is n

$$n = n_{tot} - n_{cons} = n_{tot} - n_k - n_{cl} = 6(N-1) - \sum_{i=1}^{5} i \cdot c_i - n_{cl} \tag{8}$$

The clearance gap variation of each component of the $\Delta\overline{q}$ vector describes the clearance variation in flexible n_{cl}-restricted joint dimensions. In each dimension, the variation is computed with CAT Variational Systems Analysis software using a Monte Carlo analysis with a given tolerance specification. The variation results in a Gauss distributed variation of clearance. For each dimension j, the average value of clearance x_j and standard deviation σ_j is computed. These values are utilised in MBS-analysis, where the minimum clearance can be varied using an equation for each gap q_j. If n_j=3.0, 99.87% of products will have a greater clearance than the analysed mechanism conforming 1300 scrapped units per million.

$$q_j = x_j - n_j \sigma_j \tag{9}$$

The clearance gaps described by the clearance vector Δq are modelled by force restrictions in each dimension by position-dependent force functions in ADAMS, see Figure 2. These forces are acting in the normal direction of the plane. In the direction of the plane, Coulomb friction is assumed.

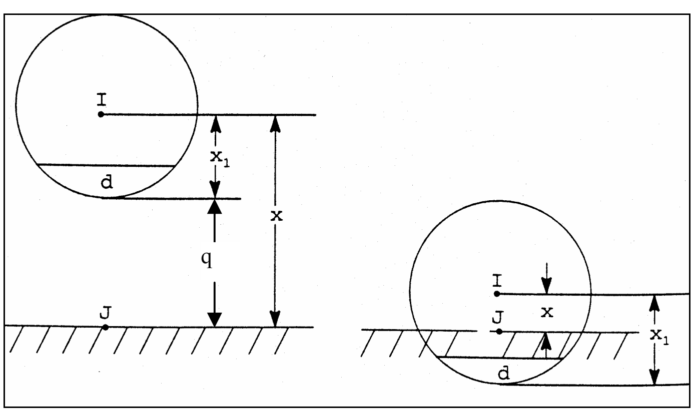

Figure 2. Impact-function's contact geometry.

The contact force function is given by equation

$$
\begin{aligned}
&F(x) = f_k(x - x_1)(x - x_1)^e + f_c((x - x_1)/d)\dot{x}\\
&f_k(x) = k(x - |x|)/(2x); f_k(0) = 0 \qquad ; \qquad (10)\\
&f_c(x) = c(x - |x|)/(2x)(2x^3 + 3x^2)(-x - 1 - |-x - 1|)/(2(-x - 1)); f_c(0) = 0
\end{aligned}
$$

Where

k = contact stiffness
x_1= free length of x. If x is less then x_1, the contact is established
e = force exponent of deformation characteristic.
For stiffening contact $e > 1.0$
c = maximum damping coefficient
d = positive variable defining the penetration depth when maximum damping is applied.

The free clearance distance $q=x-x_1$ varies according the free clearance Δq_i for each dimension in each contact. ADAMS/Solver has a function similar to the above one, IMPACT and BISTOP. The Bistop-function is similar to the Impact-function, but here the clearance is limited from both sides from the maximum and minimum dimension, thus there are two equations (9) for each dimension q_i.

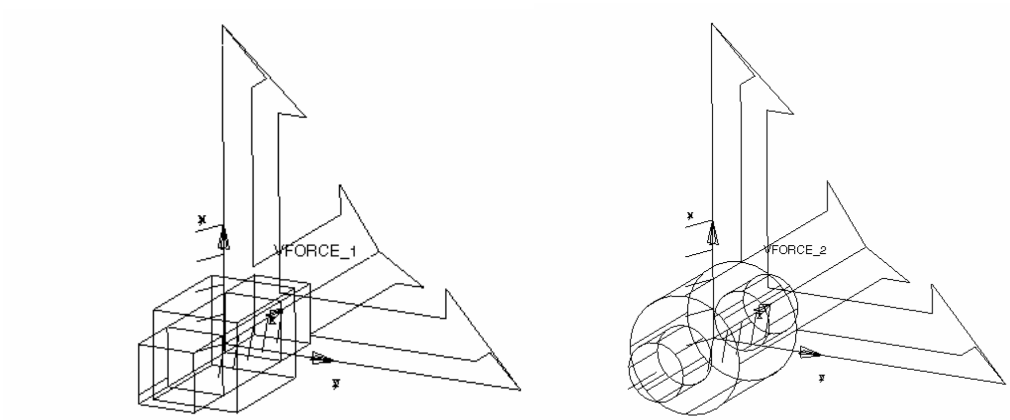

Figure 3. a) Scheme of translational MBS-clearance joint b) cylindrical MBS-clearance-joint

The clearance is described in each clearance coordinate as minimum and maximum limits for allowed free motion. The joints are then implemented with three dimensional position-force relations describing the contact mechanical motion restriction in each kinematical joint. Figure 3. shows a scheme of the motion restriction which implements a in force relation in a kinematic pair (a joint). The kinematic restriction is substituted with vector force relation, where the vector force components in main directions (x,y,z), is a function of the joints part j:s marker from i-parts marker, thus

$$
\begin{aligned}
F_x &= F_x(x_j - x_i)\\
F_y &= F_y(y_j - y_i) \qquad (11)\\
F_z &= F_z(z_j - z_i)
\end{aligned}
$$

In each component, the coordinate restriction can be as in equation 10, depending of the joint's geometry. An example of the joint geometry in the model is given in figure 6.

3 Example: CD-ROM Drive

3.1 Tolerance analysis

A CD-ROM drive slide frictional behaviour variation was analysed with three programs. The geometry was modelled with Pro/Engineer, the tolerance with VisVSA tolerance analysis software and the function was simulated with ADAMS Multi-Body analysis program.

Table 1. Mechanism data

Mutual joint width	124 mm
Mutual joint distance	36 mm
Slider mass	81.8 g
Nominal height clearance	0.4 mm
Nominal width clearance	0.4 mm
Joint stiffness	100 N/mm
Joint damping	1 Ns/mm
Maximum damping depth	0.1 mm
Joint progressiveness exponent	1.5
Coulomb coefficient of friction	0.3

The tolerance analysis was utilised to compute the clearance mean and standard deviation variation of the slider joints. The CD-ROM slider mechanism contained four linear glide joints, allowing the translational movement to have one degree of freedom.

The model of the CD-ROM drive consisted of 29 plane- and 18 point-features. Since there was no statistical data for tolerance measures for the demonstration example of this method, tolerances according to the ISO 2768H for flatness were chosen as the default.

3.2 Tolerances in the frame

For the clearance analysis between the mating elements of the slide and the frame, some features of the parts have to be tolerated. Surfaces, that contain no features are not tolerated and therefore have no effect in the analysis process in VisVSA.

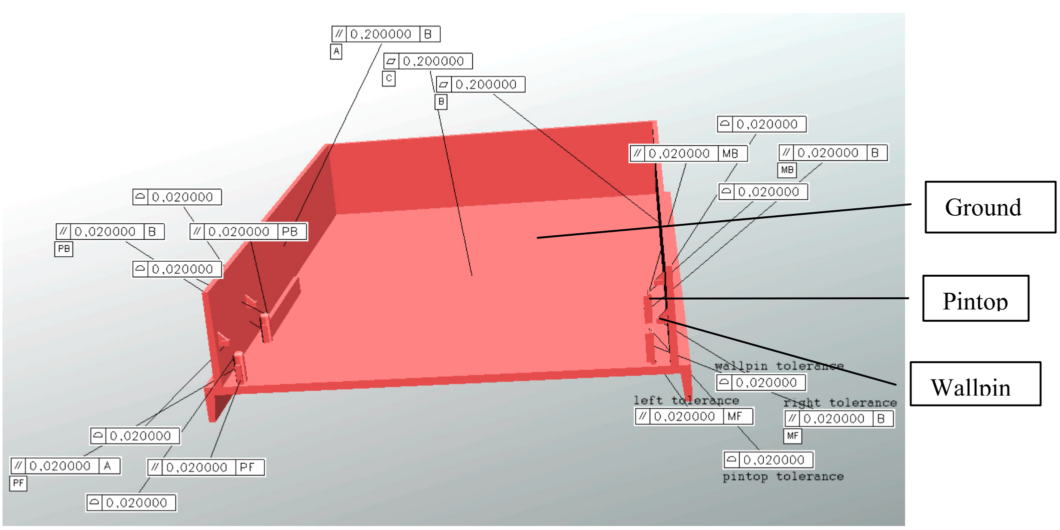

Figure 4. Defining tolerances (ISO) and references in the frame.

3.3 Tolerances in the slide

The slide contains the rails for the guided motion of the mechanism. Important tolerances for the clearance analysis are all in the tolerance library:

Slide/flat contains a flatness tolerance for rail planes, that serve as datum references
Slide/parallel contains a parallelism tolerance for the rail planes.

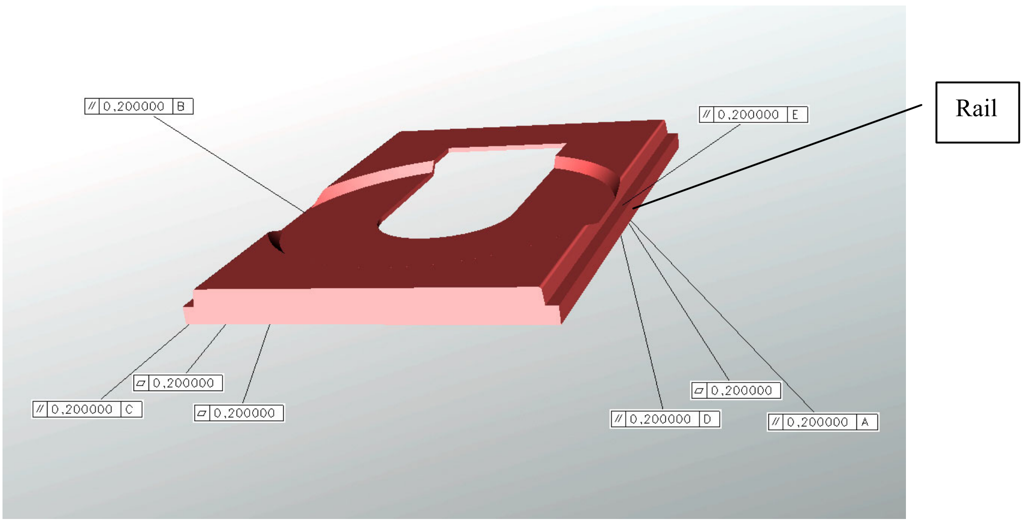

Figure 5. Defining tolerances (ISO) and references in the slide.

3.4 Measurement Operations and Clearance analysis

In the model of the CD-ROM-drive, clearances are critical dimensions in the mechanism. They are computed as *measures,* critical dimensions as function of defining tolerances. Each pin-rail-wallpin-setup needs four measurement operations to be fully described, see Figure 5. Also, two measurements are necessary to determine the distance of the two rails in the front and in the back of the slide. The model contains 18 measurement operations in total.

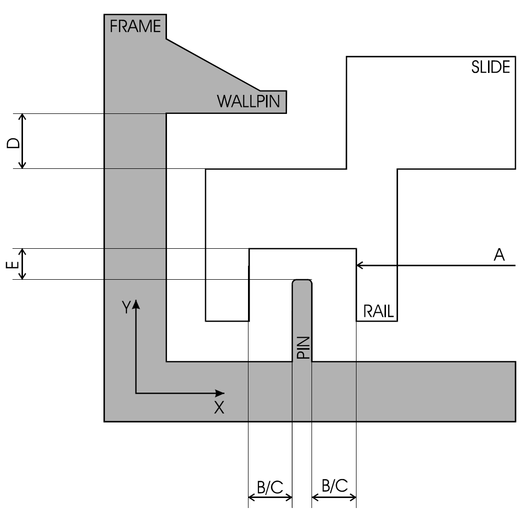

Figure 6. Clearances according to the measurement operations.

All the measurements are points from a plane. The result, calculated by VisVSA, is always the shortest distance between the plane and the definite point on the other surface.

In order to determine the overall clearance in the X- and Y-direction, the single clearances have to be added.

When running the analysis, VisVSA calculates the parameters of each measurement operation. Since the tolerances are normally distributed, the results are Gaussian-curves. Table 2 shows the parameters of all the measurements. 6 shows the results of VisVSA.

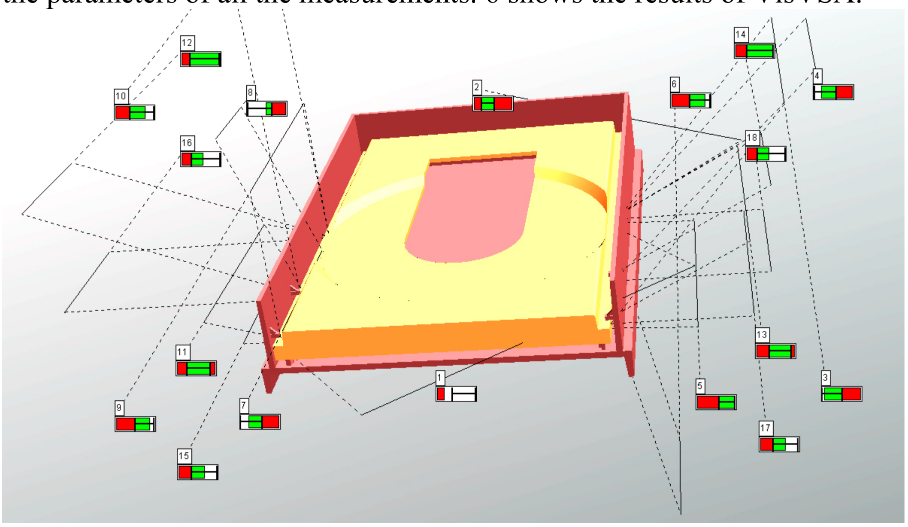

Figure 7. Measuring points to be computed in result-screen in VisVSA.

The results are summarised in Table 2. The table describes the clearance variation in the four clearance joint pins. In each pin, four clearances are measured as shown in Figure 5. In each analysis, the mean value and standard deviation are computed according 5000 Monte Carlo simulations using the ISO 2768H tolerances. According to these values, the relative position of each pin position relative to the journal is computed in two directions: left-right position and bottom-top direction (intermediate results not shown). The results are shown in column n., computed according to eq. 9. The free clearance and relative mean value position in the clearance gap is computed using the relative variation n=3.43, which means that 0.031 % of the production might have tighter clearances than this example. In the last column, the toe-outs of the rails are computed in the main directions.

Table 2. Results from tolerance analysis.

LEFT BACK PIN				**n**	3.43		
clearance	Nr.	mean value	std dev			Toe-out	
left of pin	10	0.0064	0.0404	Mean	0.2368	Back toe-out	0.0650
right of pin	8	0.4327	0.0266	Clearance	0.2093		
from pin to rail	16	-0.0003	0.0145	Mean	0.2331	Back top toe	-0.0029
from wallpin to slide	12	0.5757	0.0465	Clearance	0.3662		
RIGHT BACK PIN							
clearance	Nr.	mean value	std dev				
left of pin	6	0.2049	0.0573	Mean	0.3018	Left toe-out	-0.0005
right of pin	4	0.776	0.0478	Clearance	0.6204		
from pin to rail	18	0.0002	0.0148	Mean	0.2302	Left top toe	-0.0147
from wallpin to slide	14	0.5779	0.049	Clearance	0.3593		

LEFT FRONT PIN							
clearance	Nr.	mean value	std dev				
left of pin	9	-0.0223	0.0624	Mean	0.2363	Front toe-out	0.0735
right of pin	7	0.4037	0.0488	Clearance	0.0000		
from pin to rail	15	-0.0003	0.0176	Mean	0.2184	Front top toe	0.0015
from wallpin to slide	11	0.5754	0.0581	Clearance	0.3154		
RIGHT FRONT PIN							
clearance	Nr.	mean value	std dev				
left of pin	5	0.204	0.0767	Mean	0.3098	Right toe-out	0.0079
right of pin	3	0.7776	0.0633	Clearance	0.5014		
from pin to rail	17	0.0002	0.0177	Mean	0.2199	Right top toe	-0.0103
from wallpin to slide	13	0.5778	0.0579	Clearance	0.3187		
						Front-Back toe	0.008432
						Right-Left toe	0.008432

3.5 Multi-Body simulation

A spatial 3D six degree of freedom model was created of the slide with ADAMS 12.0 software. The clearances in each joint were modelled having parametrically variable clearances. The total clearance in each gap was a function of the relative variance *n*, changing the minimum clearance gap in each joint as a function of relative variance. Since the minimum clearance is crucial for the clearance behaviour, the method enables to simulate the function of the products that have clearances smaller than the minimum clearance according to eq. 9. The slide carrier opening operation was simulated from the first 200 ms of the opening. The relative clearance variation was from n=0..2.5 (50-0.62%) of lost production because of too-tight tolerances. The carrier speed opening was velocity force controlled according to the ADAMS force function, see [10].

$$FZ = 0.04*(STEP(TIME,0.05,0,1.0,100)-VZ) \qquad (12)$$

The STEP-function is defined in the ADAMS/Solver Handbook [10]. The force function accelerates the velocity to a maximum of 100 mm/s according to the Step-function, with P-control gain 0.04 Ns/mm. VZ is the actual momentary speed of the slide.

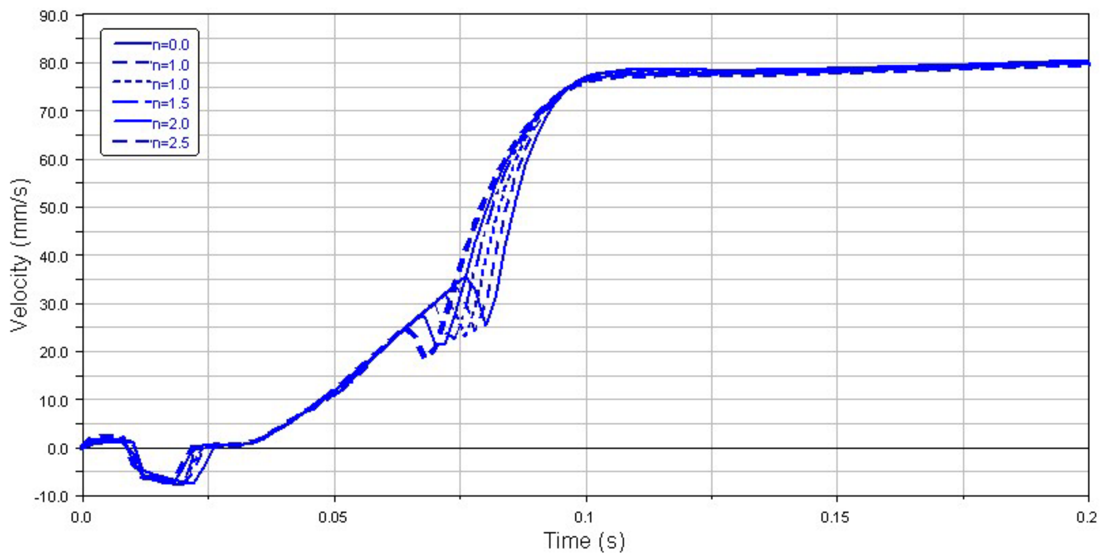

Figure 8. Carrier velocity as a function of relative variance.

[1] The force required to move the carrier is shown in Figure 8 as a function of time and relative variance *n* in eq. 9.

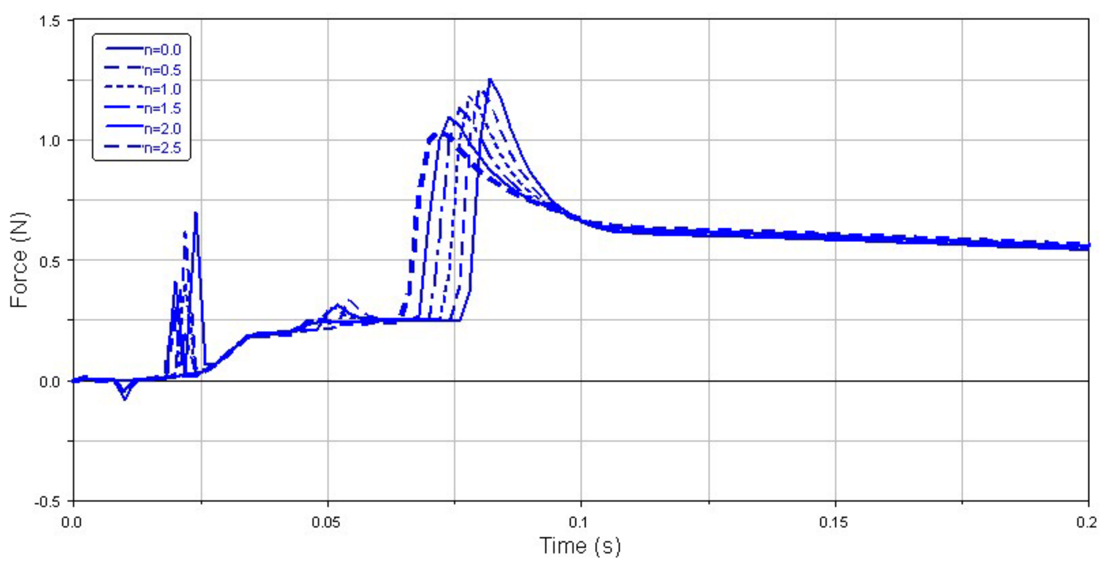

Figure 9. The required driving force, as a function of clearance variance *n*.

4 Conclusion

Computer Aided Tolerancing is a methodology, where a kinematic model is created of a product. The dimensions of the mechanism are divided to to classes; to micro-level variational dimensions which are alternated according manufacturing requirements and to macro-level kinematic dimensions, which are alternated to change the mechanisms configuration. The output of the analysis the clearance variation in mechanism joints.

In the analysis methodology presented in this paper, Multi Body Systems simulation utilised, where the micro-level clearance dimensions and macro-level configuration dimensions are identical to those of CAT analysis. The statistical clearance variation is utilised as input to the the critical clearance dimensions in Multi Body Systems simulation. Since the functional reliability in MBS greatly depends on minimum clearances of the mechanism, the subsequent use of both CAT and MBS results to predictions, how those products having the most or least compressed joints can behave in MBS analysis. Therefore, a vision of the most varied products malfunctions in a long production serie can be predicted before manufacturing is started.

Based on this analysis, tolerance synthesis is then needed to change the tolerances so, that all of the products can pass the functional requirements. This can be done by several ways, which Six Sigma is prehaps the most known. The utilisation of Six Sigma is not main goal of this paper.

The Six Sigma method has become a dominating paradigm in mass production industries. This requires a valid design synthesis method for ensuring the production remains inside tolerance limits. Tolerance synthesis is impossible without a valid tolerance analysis method. Geometry-based tolerance analysis does not consider the functional effects of tolerance variation. Therefore, it is essential to couple multi-body simulation to tolerance analysis if functional variation is to be analysed. Tolerance analysis in this study is based on the Monte Carlo analysis assuming Gaussian distribution. In large production series, this assumption is valid. The multi-body analysis is based on the variation of clearances. Conventionally, the increase of clearance usually results in a smoother operation due to the elimination of compressive fits in bearings. This can be seen in Figure 8, where increasing clearance results in a diminishing actuator force requirement. Variation between compressive and non-compressive fits in glide bearings results in a very variable friction coefficient, which is undesirable.

This paper demonstrates the methodology of subsequent tolerance and function simulations; considering the new approach of parallel use of CAT and MBS. The goal of this paper is to present the developed concept and methodology. The proposed method shows efficiently by simulation possible critical dimensions and simplifises the prediction of modification’s effects.

The validation of the results by proposed method requires still more research; an essential future question is the verification of the hypothesis, that tightes and loosest combinations of tolerances are the most probable to malfunction. This must be tested long series of MBS analysis where combinations are varied according the whole set of CAT analysed product variants. This is the next stage of this research.

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