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Assembly Sequence Influence on Geometric Deviations Propagation of Compliant Parts

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This paper presents a non-rigid part variation simulation method for fulfilling functional requirements on compliant assemblies. This method is based on the propagation of different geometrical deviations (manufacturing and assembly process defects) using the Method of Influence Coefficient. Tolerance analysis of compliant assemblies is also achieved very early in the design stage. As a consequence, designers and manufacturing engineers can efficiently analyse the assembly design principles both in terms of installed stresses and geometric variation clearance. They can also set optimised sequences that enable to get rid of geometric variations.

Keywords: Tolerancing Analysis, compliant parts, assembly sequencing

Introduction

To reduce energy consumption, automotive and aeronautics industry are concerned with decreasing the weight of their assemblies while using more and more systems to improve their products -avionics, electronics, hydraulics or mechanical systems, for instance. The important developments in computer engineering in the last three decades allow these industries to model more and more components during the design stage. The complexity may be defined in terms of material or mechanical behaviour, geometry. It also involves manufacturing overconstrained assemblies taking into account part compliance.

The reduction in time development leads these industries to simulate the behaviour of their assembly as accurately as possible and at the earliest stage, in order to minimize the need for physical tests and to improve their producibility. A way to achieve this aim is the use of variation simulation tools as soon as possible.

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2
3
4 In early design, the nominal dimensions of parts are decided and tolerances
5
6 and locating schemes are assigned. Assemblies with such parts have to respect some
7
8 functional requirements like geometric clearance or installed stresses in the
9
10 assemblies' joints. At any time, it is important to make sure that the functional
11
12 requirements of a complex assembly are satisfied. Söderberg [1] explains that
13
14 variation in a geometrical key characteristic of an assembly depends on three different
15
16 sources: component variation, assembly variation and design concept as shown in
17
18 Figure 1. Mechanisms in automotive or aircraft industries are composed of compliant
19
20 parts with deviations from nominal shapes. Moreover such parts may be jointed
21
22 together with an important number of joints which overconstrain the considered
23
24 assembly. In this context, the industrial need for variation simulation tools used in the
25
26 design stage increases quickly, while the development and pre-production time
27
28 decreases.
29
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34
35 Please Insert Figure 1
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38
39 There are different ways to manufacture an over-constrained assembly of
40
41 components whose geometry has defects:
42

- 43 • Components should have restrictive tolerances. This implies important costs,
44 which makes the industry less competitive.
- 45 • Some kinematic adjustments between parts may be used to compensate for
46 assembly variation. Functional requirements may not be expected.
- 47 • Components may be compliant to compensate for parts' defects.
- 48 • Finding the best equilibrium between the last two points.

49
50 To help designers analyze their products, it is thus important to provide them
51
52 with tools to perform the best and fastest choices; more particularly to predict how the
53
54 design and set of tolerances they assign to assembly compliant parts affect the
55
56 product's functional requirements.
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Scope of the paper

This paper presents a method based on different existing approaches to analyze an assembly composed of compliant parts. Used in the design stage, the proposed method allows engineers to verify if functional requirements of final assembly are fulfilled. The proposed method relies on two major axes which are tolerance analysis and compliant assembly variation.

After a description of recent advances, the simulation tools that are used are developed. To illustrate this theory, an aeronautic hydraulic system assembly is presented in section four as a case study. The last section discusses the results and concludes the paper.

Literature review

Over the past years, many different approaches to Tolerance analysis and Compliant Assembly Variation have been developed.

Many researchers have worked on tolerance analysis and developed different models, Wirtz (1998) studies, for example, vectorial tolerancing whereas Pillet *et al.* (2005) worked on inertial tolerancing. Torsor and Jacobian approaches were respectively developed by Ballot *et al.* (1995) and Laperriere *et al.* (2002), just to name a few. Laperriere and Kabore (2001) have already worked on tolerance synthesis with Monte Carlo simulation, but most of these works deal with tolerance analysis of rigid-body components, which is far from the exposed context.

Tolerance analysis for compliant parts consists in predicting the amount of misalignment that will occur between parts, but also enables to predict the amount of deformation of parts and stresses in joints. Many research works have focused on predicting dimensional variation on sheet metal assembly. Most of the developed methods rely on Finite Element Analysis (FEA). Chang and Gossard (1997) simulated

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2
3 the assembly and measurement processes with part stiffness matrix to find the final
4
5 variation of an assembly. Liu and Hu (1997) proposed a model to analyze the effect of
6
7 part geometrical deviations and assembly springback on assembly variation. The
8
9 method based on sensitivity matrix, called method of influence coefficient, combines
10
11 FEA and Monte Carlo simulation to reduce computational costs. Ceglarek and Shi
12
13 (1997) presented a Beam-based model for tolerance analysis for sheet metal
14
15 assemblies. Merkley (1998) proposed a tolerance analysis of compliant parts which
16
17 accounts for the part covariance in conjunction with the part stiffness matrix. Cid *et*
18
19 *al.* (2005) established an extension of 3D link chains to provide symbolic equations of
20
21 the assembly by integration of compliance and actual deviation from the nominal
22
23 shape of the considered part. Breteau *et al.* (2007) developed this approach for the
24
25 measurement of compliant parts. Most of these works, based on the study of one
26
27 compliant part, have been extended to mechanisms composed of several parts and/or
28
29 to the integration of different parameters influencing assembly variation.
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32
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34

35
36 Hu *et al.* (2006) summarized recent developments in simulation models for
37
38 compliant assemblies and presented different applications. Based on assembly state
39
40 space models and multi-station levels for assembly variation analysis, Camelio *et al.*
41
42 (2003) developed these applications for variation analysis, robust design, tolerance
43
44 allocation and variation reduction. The state space model has been extended by Loose
45
46 *et al.* (2009) with the development of an analytical derivation to describe features
47
48 deviations controlled by GD&T characteristics. For tolerance analysis some aspects
49
50 such as assembly sequence, contact modelling, we refer to Dahlström *et al.* (2005),
51
52 and fixture layout, referring to Huang *et al.* (2007), need to be incorporated to find a
53
54 correlation between simulated variations and actual ones. Söderberg *et al.* (2007)
55
56 presented a tool for non-rigid variation simulation and visualization based on
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1
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3 sensitivity matrix and more particularly on the unit influence coefficient method. This
4
5 paper proposes to extend this approach by taking into account relevant aspects
6
7 exposed before such as assembly sequence, fixture defects and compliance,
8
9 manufacturing variations in order to check if functional requirements of a compliant
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11 hydraulic assembly, we refer to Mounaud *et al.* (2007), are satisfied. Wei (2001) used
12
13 a similar approach to the one proposed to predict the fulfilment of functional
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15 requirements of such assemblies by taking into account geometrical variations in
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17 tubes and in structures in order to evaluate the cost of different designs. But the
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19 proposed models are restricted to tube compliance with no influence of the assembly
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21 sequence.
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27

28 **Description of the proposed method**

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30 Part compliance enables assembly but it also causes considerable variations that
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32 influence the final geometry of the assembly. As compliant parts may be linked to
33
34 others parts (compliant or not) with important number of joints it over-constrains the
35
36 considered mechanism.
37
38

39
40 To be functional, assemblies of compliant parts need to fulfil some functional
41
42 requirements (see equations 1 and 2) which are usually residual stresses “ F_i ” of the
43
44 assembly in each joint “ i ” and geometrical requirements “ U_i ” on specific points of the
45
46 assembly.
47
48

$$49 \quad F_i < F_{max} \quad (1)$$

$$50 \quad U_j < U_{max} \quad (2)$$

51
52 To predict the fulfilment of the above functional requirements the proposed
53
54 method is carried out according to the seven following steps detailed in the current
55
56 section:
57
58

- 59 (1) Definition of each part's deviations.
- 60 (2) Definition of the mechanical behaviour of each part.

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- 4 (3) Definition of the mechanical behaviour of the assembly
- 5 (4) Calculation of the Influence matrix based on the unit displacement method and
- 6 part stiffness matrix from Finite Element Analysis
- 7 (5) Identification of the most robust fixturing scheme for part localization.
- 8 (6) Simulation of an assembly sequence
- 9 (7) Results
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Actual Geometry of components

Considering Tolerance Analysis and assembly simulation, the actual geometry of parts is supposed to be known thanks to admissible geometries described through geometric specifications or indirectly by the way of manufacturing parameters. Whatever the definition of admissible geometry, a representative set of geometry instances is supposed to be known for each part of the assembly.

Let us suppose a frame is attached to each component of the assembly. For each point C_i of the nominal component, the manufacturing deviation $d_M(C_i)$ from the nominal model to the component is known and expressed relatively to the component frame. The actual position of a point \bar{C}_i of the component is known by the equation 3:

$$\bar{C}_i = C_i + d_M(C_i) \quad (3)$$

Definition of mechanical behaviour of each component

Tolerance analysis is applied on assemblies that mix rigid components and compliant components.

- Considering compliant components, geometric and mechanical behaviour are linked through a rigidity matrix since the following hypotheses are used:
 - the deformation of the component remains linear,
 - material behaviour is isotropic,
 - small displacements are considered.

Let the deformation of the component be expressed relatively to the associated frame. Let C_i be a nominal point of a component C , the position of the point that results in a small displacement of the component frame and of the component deformation is given by the equation 4.

$$C_i^* = \begin{bmatrix} \underline{R} & T \\ 0 & 0 & 0 & 1 \end{bmatrix} [C_i] + U(C_i) \quad (4)$$

where \underline{R} represents the small rotation of the frame, T represents the translation of the origin of the frame and U represents the displacement of the C_i point due to the component deformation (as shown in figure 2).

Please insert Figure 2

The displacement of the C_i point due to the component deformation is implicitly known by the linear relations between exterior mechanical solicitations F and displacements U:

$$F = \underline{K}.U \quad (5)$$

where \underline{K} is the matrix rigidity of the component.

- Considering rigid components, the geometric behaviour is defined through screw theory as far as geometry is concerned and static as far as mechanical behaviour is concerned.

Let C_i be a nominal point of a component C, the position of the point that results in a small displacement C_i^* of the component frame is given by the equation 6

$$C_i^* = \begin{bmatrix} \underline{R} & T \\ 0 & 0 & 0 & 1 \end{bmatrix} [C_i] \quad (6)$$

where \underline{R} represents the small rotation of the frame and T represents the translation of the origin of the frame.

Definition of mechanical behaviour of the assembly

The behaviour of the assembly is known through the behaviour of each component and the links between components. Using a finite element method, the behaviour of links between components is known as the boundary conditions.

A link behaviour is defined by equations that represent the interface between two components expressed in terms of transmissible mechanical solicitation (contact

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2
3 is insured) and degrees of freedom (mechanical solicitation is null) as it is explained
4
5
6 by Mounaud *et al.* (2007).

7
8 For each interface point, the equations result from:

- 9
10
11
- 12 • The mechanical equilibrium of the interface.
 - 13 • The continuity of the displacement when contact is effective.
 - 14 • The absence of mechanical solicitation when a degree of freedom exists.
 - 15 • The stiffness of the interface when the interface is elastic.
- 16

17 As the values of the mechanical deviations are supposed to be known, the
18
19 solution to the set of equations provides the values of the unknown in terms of
20
21 displacements and mechanical solicitations.
22

23
24
25 ***Calculation of Influence matrix based on the unit displacement method and part***
26 ***stiffness matrix from Finite Element Analysis***
27

28 A way of performing the simulation faster than solving the set of equations for each
29
30 instance of geometry is to establish a linear relationship between part deviations and
31
32 assembly spring-back deviations by using the Method of Influence Coefficient
33
34 developed by Liu and Hu (1997). This method can be applied as part deviations from
35
36 the nominal geometry are considered small and the material properties remain in the
37
38 linear range.
39
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41

42 In this method, a unit displacement is applied at the i -th source of variation ,
43
44 $i=1$ to $6 \cdot p$, where p is the number of links. Then the set of equations is used to
45
46 calculate the response f_i under that unit displacement. f_i represents a vector of
47
48 influence coefficients associated to the i -th source of variation. If the response is
49
50 recorded for all the sources of variation, the matrix of influence coefficients \underline{S} is
51
52 obtained. As the mechanistic problem is considered a linear one, using the property of
53
54 superposition of linear systems, the total imposed displacements u_d of the tube can be
55
56 expressed as a linear combination of unit displacements used for establishing matrix
57
58
59
60
 \underline{S} :

$$f_{tot} = \underline{S}^* u_d \quad (7)$$

Consequently, all the installed forces f_{tot} (at each joint of the tube) can be calculated for all the configurations of the actual pipe geometry obtained with Monte Carlo simulations. As the assemblies studied are over-constrained, it is important to find an optimal datum choice to localize the actual tube's geometry.

Identification of robust fixturing scheme

To simulate the influence of the assembly sequence on the functional requirements the assembly has to satisfy, the first step is to find the most robust fixturing scheme. That consists in finding the best set of joints to minimize the geometrical deviations of the assembly. Each joint of a compliant part with others can be described as a combination of elementary joints i.e. point contact locator characterized by both its own point contact "P_i" and its outward unit normal vector at contact point n_{ij} . 'j' represents the elementary joint composing the *i*-th joint in case of same point contact.

To illustrate this, if a joint is considered a planar joint, it will be defined as a combination of 3 point locators in the same plane. For each point locator it is possible to define plückerian coordinates with equation 8:

$$\{T_{pi}\}_j = \left\{ \begin{array}{l} n_{ij} \\ OP_i \times n_{ij} \end{array} \right\}_{P_i} \quad (8)$$

where O is a fixed point.

Deterministic localization is a fundamental requirement indicating that the part cannot make infinitesimal motion without losing contact with at least one locator.

Some authors like Zhaoqing *et al.* (2008) use genetic algorithms to find the optimal fixture layout which sensitivity of product variation to fixture errors is minimized.

The method proposed by Wang *et al.* (2001) allows to achieve this requirement only if a matrix called locator matrix, \underline{M}_{loc} defined by equation 9, has its full rank of 6.

The proposed locator matrix in this paper is composed of the plückerian coordinates of 6 elementary joints among all the possible elementary joints.

Please insert Figure 3

A classical 3-2-1 fixture scheme, represented on Figure 3, has its locator matrix with the following form:

$$M_{loc} = (\{T_{P1}\}_1 \quad \{T_{P2}\}_1 \quad \{T_{P3}\}_1 \quad \{T_{P4}\}_1 \quad \{T_{P5}\}_1 \quad \{T_{P6}\}_1) \quad (9)$$

If there is a choice of 6 locators among N , at most $C_N^6 = \frac{N!}{6!(N-6)!}$ possible combinations to achieve this scheme exist. Maximizing the determinant of the locator matrix is a suitable criterion for accurate localization as shown by Wang and Pelinescu (2001) i.e. $\max(|\det M_{loc}|)$. As the method may be very time-consuming, a reduction of the combination can be achieved by avoiding wrong cases of localization. For example, if a fourth locator exists on the principal plan of figure 3, there is a choice of 6 among 7 locators to make. Thus, solutions with 4 locators on the principal plan which over-constrain the layout of the part can be ruled out before the combinatory method.

Simulation of assembly sequence

A simulation of assembly sequence predicts more correlated results with physical tests than the FEA, based on the minimization energy principle detailed in section 3.2. Camelio *et al.* (2003) considered the assembly sequence as a multi-station assembly if it is assumed that a step of the assembly is represented as a station.

At each stage of the assembly, the equation (10) is equivalent to the following one:

$$[K \quad -I] * \begin{bmatrix} u \\ f \end{bmatrix} = [0] \quad (10)$$

Formally, force or displacement values can be imposed on every point leading to the boundary conditions. Writing all the boundary conditions leads to the equation 11:

$$\begin{bmatrix} K & -I \\ \text{BoundaryConditions} \end{bmatrix} * \begin{bmatrix} u \\ f \end{bmatrix} = \begin{bmatrix} 0 \\ S_{BC} \end{bmatrix} \quad (11)$$

Considering that geometrical deviations are small and do not affect stiffness matrix coefficients, only the boundary conditions change while the assembly sequence proceeds. The propagation model of manufacturing variations, detailed in section 3.1, describes the gaps that must be compensated for when assembling the actual part on its joints. Only joints concerned at the current step of the assembly sequence are assembled. At each stage of the assembly sequence, boundary conditions have to meet the requirement set by the distance function defined below:

$$d = d_0 + d_{MP} + d_{Mf} + d_{efp} + d_{eff} + d_{RBM} \quad (12)$$

Where the different terms of the above equation are:

- d : actual distance between part and fixture
- d_0 : nominal distance between part and fixture (=0 in case of assembly)
- d_{MP} : Manufacturing deviation of the part
- d_{Mf} : Manufacturing deviation of the fixture
- d_{efp} : deformation of the part
- d_{eff} : deformation of the fixture
- d_{RBM} : rigid body motion if exists

The different distances can also be illustrated in the following Figure 4:

Please insert Figure 4

Although illustrations have been made in 1-D, as distances are defined as spatial vectors, 3-D problems can be solved thanks to this representation. As the actual distance between the part and the fixture has to be null to make an assembly, the boundary conditions can be solved. At a step of the assembly sequence, gaps of the considered joints thus constitute the boundary conditions the mechanistic problem

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3 has to take into account. Consequently, this displaces all the part geometry of the part
4
5 at the beginning while the mechanism can have a rigid body motion. Once the part is
6
7 totally constrained, from a kinematical point of view, its geometry deforms only under
8
9 the applied displacements. The solution found at the i -th stage represents the gaps that
10
11 need to be closed at the $i+1$ -th stage. The method is iteratively going on until the end
12
13 of the assembly sequence.
14
15

16 17 18 *Selection and representation of results*

19
20 The simulation of an assembly sequence is performed for instances of manufacturing
21
22 deviations. The Monte Carlo method is used so that the results are representative of
23
24 the real behaviour of the assembly. Synthetic representation has been chosen because
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26 the amount of available data is too important. Three representations of results are
27
28 proposed. The first one provides an estimation of how the characteristics are
29
30 respected, as shown in table 1.
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32
33
34

35
36 Please Insert table 1
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40 The global estimation of the conformity of the assemblies allows designers to
41
42 validate the design or to modify the design if the rate of conformity is not acceptable.
43
44 When this situation occurs, a two-dimensional or three-dimensional representation of
45
46 the results of simulations is proposed.
47
48

49 Figure 5a illustrates a two dimensional representation of force distribution, the
50
51 choice of the local projection plane is free. On the figure, the circle represents the
52
53 admissible limit in term of force. Each point corresponds to the projection of a
54
55 simulation result in the plane.
56
57

58
59 Please insert Figure 5a and Figure 5b
60

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4 As a complement to the previous representation, the distribution of any
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6
7 evaluated data is proposed and illustrated using a histogram (Figure 5b).
8
9

10 **Case Study**

11
12 The proposed method is illustrated through an aeronautic example which
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14 consists in assembling a hydraulic tube on its brackets as shown on figure 6. The
15
16 metallic tube must satisfy the functional requirements in terms of geometrical
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18 deviations and limited effort in each joint. It is assumed that tube and brackets are
19
20 compliant. The tube is fixed with 6 brackets on its route and 2 brackets on its
21
22 extremities. Moreover, the joints are called Controlled Check Points (CCP). To
23
24 position the tube, joints are thus considered as rotational joints for both ends and as
25
26 sphere-cylinders in other cases. The main geometrical characteristics of the tube and
27
28 its material properties are detailed in the following table 2:
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30
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34

35 Please Insert table 2

36
37
38 Please Insert Figure 6 and table 3 (if possible)

39
40
41 The objective of the study is to simulate the assembly of the tube on a
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43 structure. The deviations on structure have already been evaluated and the machining
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45 process of the tube is known. At this stage of the design, the designer has to choose the
46
47 technology of the brackets, to impose the assembly planning and to verify that the
48
49 choices permit to respect the functional requirements on the assembly.
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53 For a given choice of bracket technology, we propose to simulate the assembly
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55 of the pipe on the structure without taking into account the influence of assembly
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57 planning on specification fulfilments, This simulation is not time consuming and
58
59 provides pertinent indicators about the assembly feasibility. Once the assembly
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3 feasibility is assumed, a more detailed study is performed to verify that functional
4 requirements are respected while taking into account the influence of assembly
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6
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8 planning.
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10 We now propose to detail the seven steps of the method applied to the case
11
12 study.
13

14 15 16 ***Definition of each part deviations***

17 As mentioned above, deviations on the structure have already been evaluated. This
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19 previous study provides the dispersions of the positions of locating holes of the
20
21 brackets relative to their nominal position. Locating holes appear to be centred on
22
23 nominal position and the deviations respect a normal distribution. The structure is the
24
25 base of the assembly, its frame (O,X,Y,Z) is the reference of the assembly.
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27
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29

30 Please insert table 4
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33

34 In order to minimize both the mass and the machining cost of the six brackets,
35
36 a unique technology has been chosen. The deviation of the position of the bracket
37
38 centre, relative to its base, is defined in the frame associated to the bracket (refer to
39
40 table 5). Bracket deviations can thus be expressed in the frame of the structure.
41
42
43

44 Please Insert Figure 7 and table 5
45
46
47

48 Concerning the pipe, the manufacturing process (bending process) is known,
49
50 and the capability of the bending process is identified. As shown in Mounaud *et al.*
51
52 (2007), pipe geometry is classically represented as a sequence of elementary bending
53
54 operations made with a forming die which radius is called R_f . These operations are
55
56 characterized by L-R-A manufacturing coordinates, where L,R and A stand for
57
58
59
60

Length between bends, Rotation between two successive bends and bent Angle, respectively, as shown in Figure 8.

Please Insert Figure 8

As a tube is made from a sequence of elementary operations, it is assumed that the geometry can be considered as a sequence of successive straight and circular elements. So local frames $R_i (P_i, X_i, Y_i, Z_i)$ can be associated with elementary elements i . Due to the manufacturing process, these coordinates (L_i , R_i and A_i) are made with deviations (dL_i , dR_i and dA_i respectively) limited by the process capabilities dL , dR and dA represented by the following constraint relations (equation 13, 14 and 15) for each actual elementary bending operation $\tilde{L}_i (\tilde{L}_i = L_i + dL_i)$, $\tilde{R}_i (\tilde{R}_i = R_i + dR_i)$ and $\tilde{A}_i (\tilde{A}_i = A_i + dA_i)$:

$$L_i - \frac{dL}{2} \leq \tilde{L}_i \leq L_i + \frac{dL}{2} \quad (13)$$

$$R_i - \frac{dR}{2} \leq \tilde{R}_i \leq R_i + \frac{dR}{2} \quad (14)$$

$$A_i - \frac{dA}{2} \leq \tilde{A}_i \leq A_i + \frac{dA}{2} \quad (15)$$

Figure 9 represents the distribution of the dL process capability. dR and dA are assumed having similar representations.

Please Insert Figure 9

The actual geometry of a pipe can easily be obtained by representing each elementary operation and its own defect with the Homogenous Transformation Matrix for L , R and A respectively:

$$\begin{aligned}
 \mathbf{M}_{L_i} &= \begin{pmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & \tilde{L}_i \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{pmatrix}_{R_i} \\
 \mathbf{M}_{R_i} &= \begin{pmatrix} \cos(\tilde{R}_i) & 0 & \sin(\tilde{R}_i) & 0 \\ 0 & 1 & 0 & 0 \\ -\sin(\tilde{R}_i) & 0 & \cos(\tilde{R}_i) & 0 \\ 0 & 0 & 0 & 1 \end{pmatrix}_{R_i} \\
 \mathbf{M}_{A_i} &= \begin{pmatrix} \cos(\tilde{A}_i) & -\sin(\tilde{A}_i) & 0 & R_{fi} * [1 - \cos(\tilde{A}_i)] \\ \sin(\tilde{A}_i) & \cos(\tilde{A}_i) & 0 & R_{fi} * \sin(\tilde{A}_i) \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{pmatrix}_{R_i}
 \end{aligned}$$

R_{fi} is the actual radius which integrates spring-back phenomenon as indicated in Lou and Stelson (2001). Thus, from the bend plan, the geometry of the actual tube may be generated in 3D-space by combining the effect of multiple bends through a series of matrix multiplications. As a consequence, in a global Cartesian coordinate system, the relative position (P_i) of the different specific points (P_{iloc} in the local coordinate system of the elementary element i) of the tube is obtained with the following equation 16:

$$P_i = \left(\prod_{k=1}^{k=i} M_{L_k} * M_{R_k} * M_{A_k} \right) * P_{iloc} \quad (16)$$

The bending process is characterized by the following values: 1mm, 0.1° and 1° for dL , dR and dA capabilities, respectively.

Definition of the mechanical behaviour of each part

The structure of the assembly is supposed to be rigid and the manufacturing deviations are known. As the structure is the base of the assembly, its frame (O, X, Y, Z) is the reference of the assembly (cf. Figure 7).

The brackets 2 to 7 are considered as compliant with a linear model of deformation (5N.mm along axis in translation and 5Nm.rad along axis in rotation).

The elementary stiffness matrix of a bracket is given equation 17 and 18:

$$\underline{\underline{K}}^j = \begin{pmatrix} 5 & 0 & 0 \\ 0 & 5 & 0 \\ 0 & 0 & 5 \end{pmatrix}_{CCP_i, X_i, Y_i, Z_i} \quad (17)$$

$$\underline{\underline{C}}^j = \begin{pmatrix} 5 & 0 & 0 \\ 0 & 5 & 0 \\ 0 & 0 & 5 \end{pmatrix}_{CCP_i, X_i, Y_i, Z_i} \quad (18)$$

Moreover, it is assumed that extremities are considered as rigid CCP compared to brackets. So they are modelled with the elementary matrices in equation 17 and equation 18 where values of stiffness are respectively equal to 100 N.mm and 100 Nm.Rad. Considering the tube, the Finite Element Modelling used relies on a classical strength of material approach based on Bresses' equations. The model used is detailed explicitly in Mounaud *et al.* (2007). The simulation assumes that the tube deformation remains linear, that material behaviour is isotropic and that only small displacements are considered. The result of the Finite Element Modelling is the stiffness matrix of the tube.

Definition of mechanical behaviour of the assembly

The boundary conditions that permit to link the local behaviours of the components are now presented.

- For each interface between the structure and the brackets, the boundary condition is the continuity of displacement since contact is assured. Let S_i be the point of the structure and B_i be the corresponding point of the bracket, then:

$$Position(B_i) = Position(S_i) \quad (19)$$

These equations 19 allow us to calculate the small displacement of the considered bracket frame since the positioning of the bracket is equally constrained.

- For each interface point between the bracket and the pipe, the position of the actual bracket point is known since the small displacement of the bracket frame and the manufacturing deviation are known. The geometry of the tube is also

1
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3 available and then it is possible to calculate the value of the distance function
4 presented in 3.6.

5 The technology of the bracket allows us to consider the link between the tube
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7
8 and the bracket as a sphere-cylinder link while the bracket is not clamped, and as a
9
10 complete link when the bracket is clamped.
11

12 The boundary conditions associated to a complete link are the continuity of the
13 displacement in all directions.
14

15 The boundary conditions associated to a sphere-cylinder link are:
16

- 17 • The continuity of the displacement along Y_i and Z_i directions.
- 18 • The absence of mechanical solicitation along axis where degrees of freedom exist
19 (Xi direction in our case).

20 For each interface point, the remaining equations result from the mechanical
21 equilibrium of the interface.
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30 ***Calculation of Influence matrix based on the unit displacement method and part*** 31 ***stiffness matrix from Finite Element Analysis***

32 As the values of the mechanical deviations are supposed to be known, the solution to
33 the whole set of equations provides the values of the unknown in terms of
34 displacements and mechanical solicitations.
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38 Using the method of the influence matrix, detailed in section 3.4, current assembly
39 positions can be assessed and then the joint forces can be calculated at each step of the
40 assembly sequence.
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48 ***Identification of the most robust fixturing scheme for part localization.***

49 The only component, for which the identification of the most robust fixturing scheme
50 is needed, is the tube. The application of the method developed by Wang and
51 Pelinescu (2001) to find the best fixturing scheme indicates that the tube is positioned
52 with CCP 3, 6 and 7. The latter are called “positioning CCP” and the others,
53
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58
59 “clamping CCP”.
60

Simulation of an assembly sequence

This step of the method only depends on the assembly sequences of clamping. For this case study, three sequences are studied as shown in table 6.

Please insert table 6

Results

For each sequence, 5000 simulations are performed using Monte Carlo method.

The indicators that result from sequence 1 simulations are presented in the table and indicate a global satisfaction of functional requirements, except for force criterion at point 1 and 8 and for displacements at point 2.

Please Insert Table 7

Local representations of the forces and displacements at problematic points are presented in figure 10.

Please insert figure 10

From the local representations, we can conclude the global orientations of forces and displacements concerning points 1 and 2 and a homogeneous repartition concerning point 8.

As the results of simulations are simplified, a more detailed analysis is proposed in order to quantify the influence of assembly planning. Two alternative assembly sequences are studied and the global results are presented in table 8 and 9.

Please Insert table 8

Please Insert table 9

The global results in sequence 2 (Clamping Sequence CCP 2 6 7 5 4 3) show a potential improvement in the fulfilment of functional requirements in terms of forces.

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3 Local results concerning efforts at point 1 and 8 and displacements at point 2
4
5 are proposed for this sequence analysis in figure 11.
6
7

8
9 Please insert Figure 11
10

11
12 The yield rate for the force requirement criterion is lower, so this assembly
13 could be accepted concerning forces. The remaining problem concerns the
14
15 displacements at point 2. The influence of assembly planning is negligible and the
16
17 yield rate is not acceptable.
18
19

20
21 At this stage of the design, bracket technology may still change. As the
22
23 designer has effort and displacement repartitions at his disposal, he can take
24
25 advantage of them to suggest a modification. Concerning point 2 where the problem
26
27 occurs, we note in figure 12 that the range of displacement along UZ_2 axis exceeds the
28
29 limit, while the range of force along UZ_2 axis does not reach the limit.
30
31

32
33 Please insert Figure 12
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35

36
37 An increase in the stiffness of the bracket along Z_2 direction may increase
38
39 forces and reduce displacements to produce an acceptable assembly design.
40
41

42 43 44 **Conclusion and future work**

45 Taking into account the assembly sequence of components in an aeronautic assembly
46
47 contributes to the anticipation of the geometrical part deviations to verify that the
48
49 deformed geometry meets functional requirements. The purpose of this paper is to
50
51 present a model for the influence of assembly sequence which integrates
52
53 manufacturing deviations for the fulfilment of product key characteristics. Part
54
55 manufacturing deviations have been taken into account. Due to the assembly sequence
56
57 of the structure parts, CCP positions present different variations along the different
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59
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1
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3 axes of the global frame. Moreover, the value of the variation can vary significantly
4
5 from one CCP to another.
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8 The proposed method is applied to piping assembly. As shown in section 4.1,
9
10 the actual pipe geometry depends on bending process capabilities, so the impact of
11
12 manufacturing deviations on tube geometry is easier simulated than with machining
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14 process for instance. Nevertheless, the method can be displayed on other instances of
15
16 assembly since the inputs such as Stiffness Matrix, Key characteristics, process
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18 capabilities, joints scheme and models of joints behaviour are defined. Moreover,
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20 statistical tolerances with direct Monte Carlo simulation are employed, so
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22 computational time directly depends on the number of elements used in the
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24 simulations of manufacturing deviations.
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29 Future extensions of this work include possibilities for adjustment solutions to
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31 be implemented in the early design stage. The proposed method can be used to
32
33 compare different scenarii when multiple deviations occur. Last, the interaction of
34
35 different technologic choices at the design stage can be investigated.
36
37
38

39 **Acknowledgements**

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41
42 INNO'CAMPUS programme in partnership with EADS-IW.
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		Points of interest							
		CCP 1	CCP 2	CCP 3	CCP 4	CCP 5	CCP 6	CCP 7	CCP 8
Forces	Mean μ	6.13 N	3.95 N	3.23 N	2.18 N	2.32 N	3.59 N	5.43 N	11.04 N
	Standard deviation σ	3.98 N	2.13 N	2.06 N	1.25 N	1.25 N	1.99 N	2.91 N	6.03 N
	Yield Rate	83.14 %	99.02 %	99.37 %	100 %	100 %	99.48 %	92.47 %	77.86 %
Displacements	Mean μ	0.279 mm	0.542 mm	0.356 mm	0.357 mm	0.357 mm	0.357 mm	0.354 mm	0.330 mm
	Standard deviation σ	0.145 mm	0.314 mm	0.186 mm	0.186 mm	0.186 mm	0.186 mm	0.184 mm	0.187 mm
	Yield Rate	99.99 %	91 %	99.8 %	99.81 %	99.83 %	99.74 %	99.8 %	99.64 %

Table 1: Example of Global results of assembly sequence simulation

Outer diameter	15.87 mm
Thickness	0.81 mm
Material	Titan alloy
Forming die	50 mm

Table 2: Geometrical and mechanical characteristics of case study

L (mm)	R (°)	A (°)
54.85	0	90
58.64	-5.01	13.62
164.93	180	13.62
182.57	5.01	40.67
87.14	0	49.33
1387.80	26.41	50.14
295.07	-37.77	46.56
72.75	-25.46	48.28
134.48	-107.18	25.05
100.51	-15.64	35.61
71.97	0	0

Table 3: associated LRA manufacturing coordinates to case study

	CCP1	CCP2	CCP3	CCP4	CCP5	CCP6	CCP7	CCP8
X deviation	0.5 mm	0.5 mm	0.25 mm	0.25 mm	0.25 mm	0.25 mm	0.25 mm	0.25 mm
Y deviation	0.4 mm	0.4 mm	0.25 mm	0.25 mm	0.25 mm	0.25 mm	0.4 mm	0.4 mm
Z deviation	0.25 mm	0.25 mm	0.4 mm	0.4 mm	0.4 mm	0.4 mm	0.25 mm	0.25 mm

Table 4: Structure deviations on each CCP

Xi direction	0.11 mm
Yi direction	0.11 mm
Zi direction	0.25 mm

Table 5: Deviation of CCP position in the bracket frame

Step	Sequence 1	Sequence 2	Sequence 3
1	Full tightening of CCP	Installation of positioning CCP (i.e. 3, 6 & 7)	
2	Installation of all CCP	Installation of extremities	
3		Tightening of extremities	
4		Sequential positioning and tightening of remaining CCP following the sequence	
		CCP 3 6 7 4 5 2	CCP 2 7 6 5 4 3

Table 6: Assembly sequences simulated

		Points of interest							
		CCP 1	CCP 2	CCP 3	CCP 4	CCP 5	CCP 6	CCP 7	CCP 8
Forces	Mean μ	6.13 N	3.95 N	3.23 N	2.18 N	2.32 N	3.59 N	5.43 N	11.04 N
	Standard deviation σ	3.98 N	2.13 N	2.06 N	1.25 N	1.25 N	1.99 N	2.91 N	6.03 N
	Yield Rate	83.14 %	99.02 %	99.37 %	100 %	100 %	99.48 %	92.47 %	77.86 %
Displacements	Mean μ	0.279 mm	0.542 mm	0.356 mm	0.357 mm	0.357 mm	0.357 mm	0.354 mm	0.330 mm
	Standard deviation σ	0.145 mm	0.314 mm	0.186 mm	0.186 mm	0.186 mm	0.186 mm	0.184 mm	0.187 mm
	Yield Rate	99.99 %	91 %	99.8 %	99.81 %	99.83 %	99.74 %	99.8 %	99.64 %

Table 7: Global results of tolerance analysis with no assembly sequence simulation

		Points of interest							
		CCP 1	CCP 2	CCP 3	CCP 4	CCP 5	CCP 6	CCP 7	CCP 8
Forces	Mean μ	7.00 N	4.53 N	3.28 N	2.24 N	2.42 N	3.84 N	5.38 N	9.78 N
	Standard deviation σ	4.68 N	2.55 N	2.13 N	1.27 N	1.31 N	2.12 N	2.88 N	5.53 N
	Rate of conformity	77.84 %	96.56 %	99.06 %	100 %	100 %	99.06 %	91.38 %	79.25 %
Displacements	Mean μ	0.281 mm	0.544 mm	0.351 mm	0.357 mm	0.358 mm	0.357 mm	0.353 mm	0.334 mm
	Standard deviation σ	0.146 mm	0.321 mm	0.183 mm	0.188 mm	0.188 mm	0.184 mm	0.183 mm	0.190 mm
	Yield Rate	100 %	90.56 %	99.8 %	99.78 %	99.82 %	99.82 %	99.84 %	99.68 %

Table 8: Global results of tolerance analysis of assembly sequence 3 6 7 5 4 2

		Points of interest							
		CCP 1	CCP 2	CCP 3	CCP 4	CCP 5	CCP 6	CCP 7	CCP 8
Forces	Mean μ	5.81 N	3.95 N	3.36 N	2.25 N	2.42 N	3.82 N	5.38 N	10.60 N
	Standard deviation σ	3.66 N	2.16 N	2.14 N	1.3 N	1.31 N	2.09 N	2.84 N	6.00 N
	Yield Rate	87.66 %	99.06 %	99.1 %	100 %	100 %	99.1 %	92.9 %	83 %
Displacements	Mean μ	0.282 mm	0.546 mm	0.357 mm	0.356 mm	0.355 mm	0.354 mm	0.355 mm	0.333 mm
	Standard deviation σ	0.147 mm	0.320 mm	0.187 mm	0.185 mm	0.186 mm	0.184 mm	0.183 mm	0.191 mm
	Yield Rate	100 %	90.3 %	99.78 %	99.76 %	99.78 %	99.82 %	99.88 %	99.52 %

Table 9: Global results of tolerance analysis of assembly sequence 2 6 7 5 4 3

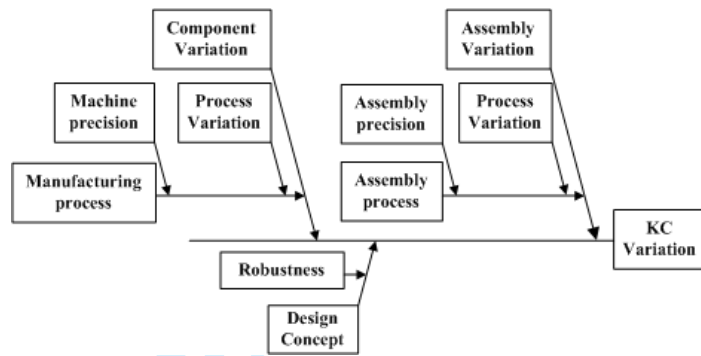


Figure 1: Geometrical KC variation contributors (Söderberg (1998))

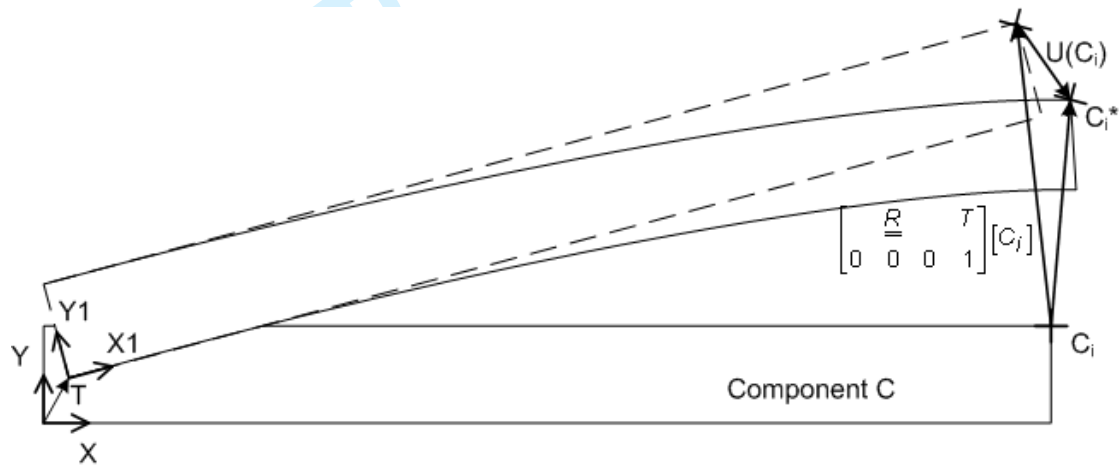


Figure 2: Displacement representation of a compliant component

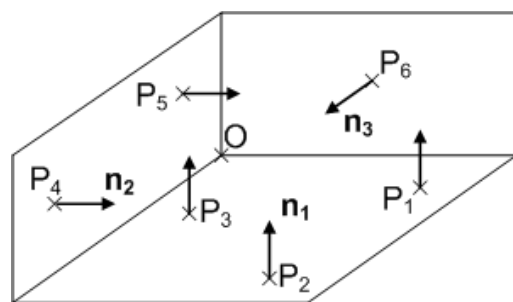


Figure 3 : classical 3-2-1 fixturing scheme

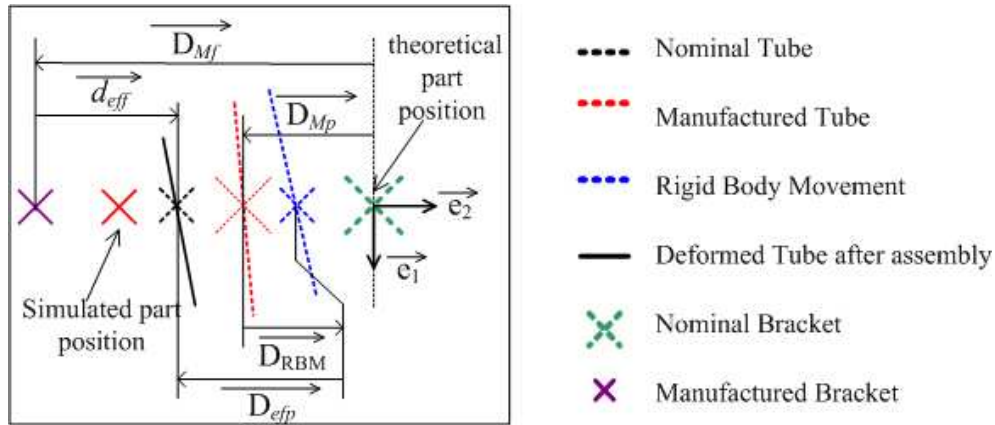


Figure 4: Assembly of an actual tube on actual brackets

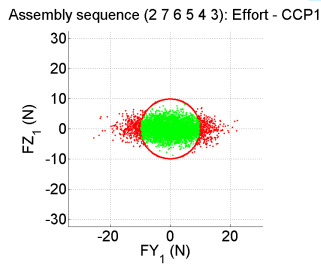


Figure 5a: synthetic graphical representation

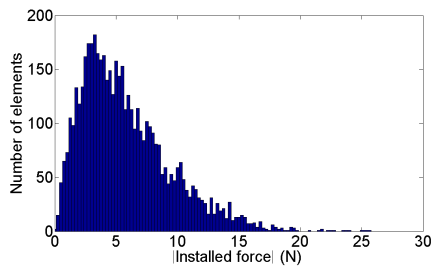


Figure 5b: histogram representation
Figure 5: representation of force distribution

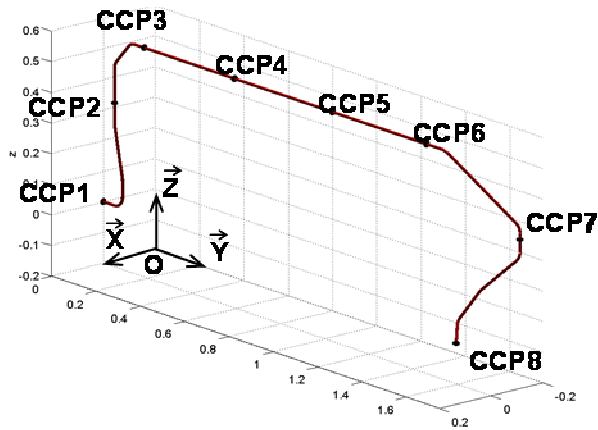


Figure 6: Case Study

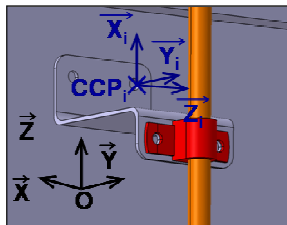


Figure 7: Bracket and structure frames

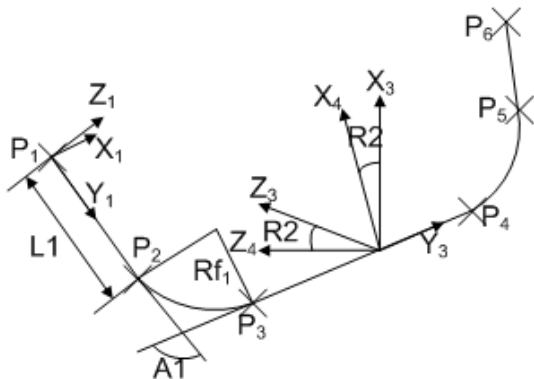


Figure 8: Manufacturing coordinates L, R, A

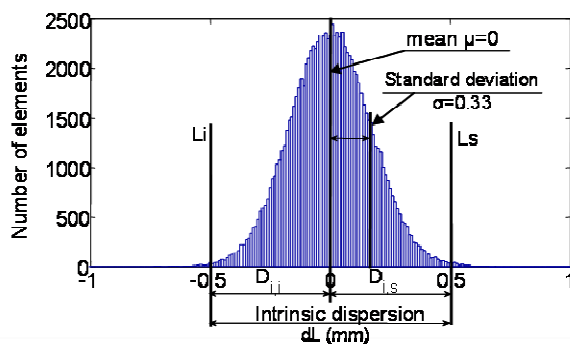


Figure 9: Intrinsic dL Bending process dispersion

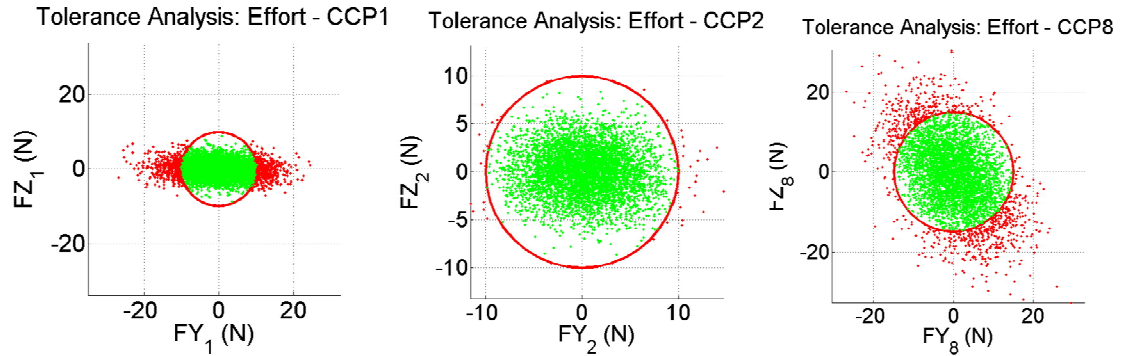


Figure 10: Local representations of the forces in CCP 1, CCP 2 and CCP 8 for Tolerance analysis

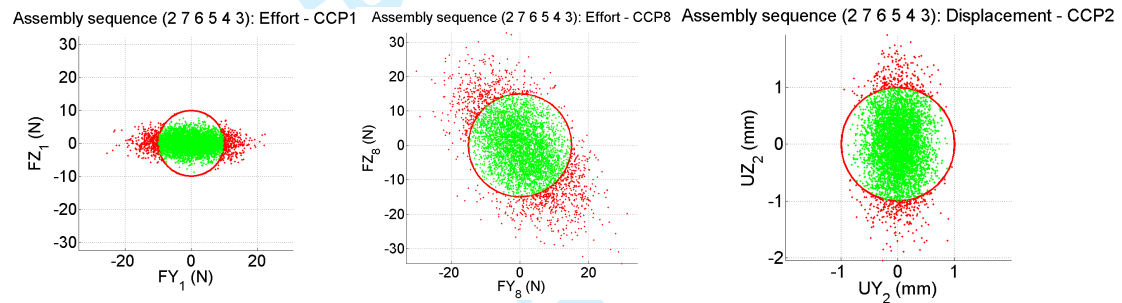


Figure 11: Local representations of the forces in CCP1 and CCP8 and displacement in CCP2 for assembly sequence 2 6 7 5 4 3

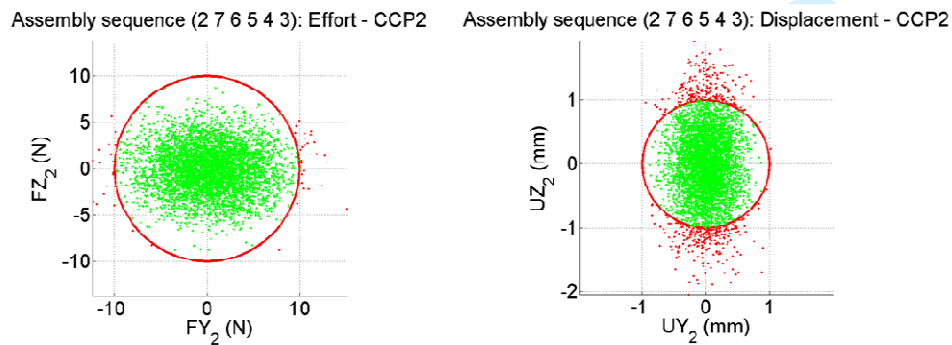


Figure 12: Local representations of the forces and displacements in CCP2 for assembly sequence 2 6 7 5 4 3